GUIDE

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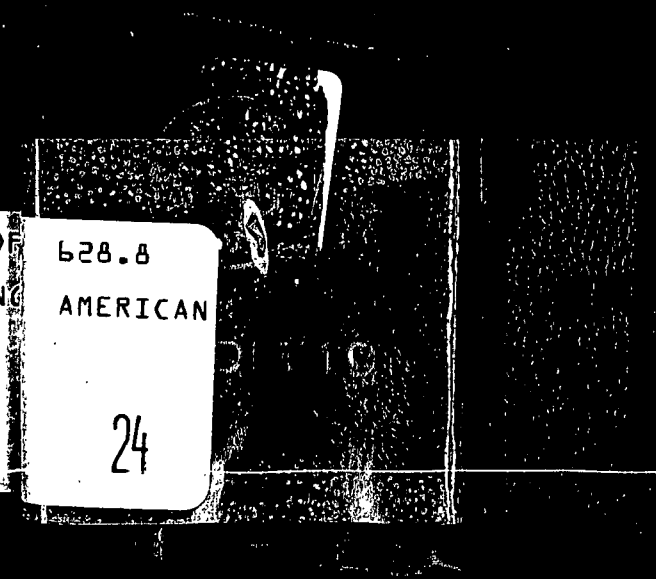

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HEATING VENTILATING AIR CONDITIONING GUIDE 1946

HEATING VENTILATING AIR CONDITIONING GUIDE

1946

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 CO-OPERATING INSTITUTIONS—AND THE PRACTICE OF THE MEMBERS AND FRIENDS OF THE SOCIETY

TOGETHER WITH A

Manufacturers' Catalog Data Section

CONTAINING ESSENTIAL AND RELIABLE INFORMATION CONCERNING MODERN EQUIPMENT

ALSO

The Roll of Membership of the Society

WITH

Complete Indexes

TO TECHNICAL AND CATALOG DATA SECTIONS

Vol. 24

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PREFACE TO THE 24th EDITION

THE twenty-fourth edition of THE HEATING, VENTILATING, AIR CONDITIONING GUIDE contains an enlarged technical data section and a new grouping of subjects to provide a more logical progression from fundamental principles to special application. THE GUIDE serves a dual function as both textbook and handbook. As text, it represents a condensation of the pertinent experimental data in this field. The desire for accuracy precludes the use of many of the approximate but convenient tables of the ordinary handbook.

Terminology, Chapter one, covering engineering terms used in heating, ventilating, and air conditioning, has been revised to obtain better conformance with the terms of basic science rather than the less accurate forms which are frequently adopted by engineers in a specific field. Progress in standardization of letter symbols has been continued; leading to uniformity of symbols used throughout the text as far as possible. Separate lists of symbols have been placed at the end of those chapters in which large numbers of symbols are used.

The most accurate information obtainable on properties of moist air and of water in the range −160 to +212 F has been obtained from the cooperative research work of the A.S.H.V.E. and the Towne Scientific School of the University of Pennsylvania and is incorporated in Chapter 3, Thermodynamics. The new Mollier Diagram for Moist Air, prepared by John A. Goff, corresponds with the revised tables of Properties of Moist Air and of Water in Chapter 3, and, in its larger size, is improved in readability.

Chapter 4, Fluid Flow, is a new and very useful addition to the text as it presents the basic theory of fluid flow through pipes, nozzles, and orifices and gives recommendations for location of pressure taps for orifice and venturi meter applications. The explanation of the method of using the data in Chapter 6, Heat Transmission Coefficients of Building Materials, has been simplified and certain suggestions regarding the use of the tabulated data have been added.

Chapter 10, Air Contaminants, contains most of the material previously given in the former chapter on Air Pollution as well as the latest acceptable limits for toxic gases and dusts in working spaces. The rewritten chapter gives the viewpoint of the industrial hygiene engineer. Much useful information regarding the use of testing equipment is given in Chapter 11, Instruments and Measurements.

In Chapter 12, Physiological Principles and Chapter 13, Air Conditioning in the Prevention and Treatment of Disease it has been possible to add interesting information obtained from tests conducted in cooperation with the military services during the war.

Chapter 15, Cooling Load, has been improved by the addition of new data on solar heat transmission through glass obtained from a recent study made at the A.S.H.V.E. Research Laboratory. Revision has been made in the method of computing the cooling load in order to reflect present knowledge of the subject.

A new section on furnaces has been included in Chapter 18, Heating Boilers and Furnaces. This chapter now includes information on the characteristics, performance, and latest rating methods for these appliances. Chapter 21, Gravity Warm Air Systems, and Chapter 22, Mechanical Warm Air Systems show the progress made during the past year in the methods of designing and installing warm air systems.

To the pipe capacity tables in Chapter 23, Steam Heating Systems and Piping, have been added the capacities expressed in pounds of steam so that these capacities are now available in both EDR and pounds. Similarly a chart has been added to Chapter 24, Hot Water Heating Systems and Piping, showing pipe capacities in pounds of water per hour corresponding to various friction losses. This chart is supplied in addition to the familiar chart showing the capacities based on 20 deg temperature drop. Methods (with illustrative examples) of computing heat loss through pipe covering consisting of layers of different kinds of insulation have been added to Chapter 28, Pipe Insulation.

Chapter 38, Dehumidification by Sorbent Materials, has been amplified to include the latest accepted practice in use of sorbents. Chapter 39, Refrigeration, has been enlarged to permit a simple presentation of the different refrigeration cycles in addition to a brief treatment of the most commonly used refrigeration equipment arrangements.

New material from Society sponsored research on air flow in outlets has been added to Chapter 40, Air Distribution, to provide additional useful design information. As a result of studies of the best available data on air flow, made at the A.S.H.V.E. Research Laboratory, a new air friction chart has been prepared and this chart is now included in Chapter 41, Air Duct Design.

The economics of air conditioning are given in a new Chapter 44, Owning and Operating Costs, which shows the method of determining the actual cost of air conditioning to the owner.

In Chapter 51, Codes and Standards, the list of various codes affecting installation of air conditioning and heating equipment has been brought up-to-date through information obtained from the various state departments concerned. The chapter also contains a list of more than 75 codes and standards, relating to performance, rating, testing, selecting and installing equipment used in heating, ventilating, and air conditioning work, adopted by various societies and organizations.

All chapters not specifically mentioned in the foregoing have been carefully reviewed so that they conform to current engineering practice.

A complete cross-index of the technical section provides ready reference to any desired information in the text.

As in the past the Guide Publication Committee has had the cooperation and enthusiastic support of a great many Society members and interested organizations, as well as other authorities on the subjects treated in THE GUIDE. To those who have contributed to the success of the previous editions great credit is due for the present high standing of THE GUIDE throughout the world. In the preparation of the 1946 edition the following members and others have served faithfully and capably and therefore the Guide Publication Committee takes great pleasure in expressing its sincere appreciation of their services:

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An enlarged Catalog Data Section contains condensed information from 233 manufacturers of modern equipment and constitutes a valuable reference section for the engineer, architect, contractor, or student interested in all types of heating, ventilating, and air conditioning installations. A cross index is included so that information on any product can be located quickly by the user.

This 24th edition of THE GUIDE, like its predecessors, provides the latest available data compiled from authoritative sources for the use of the practicing engineer and reflects the progress of engineering in the Heating, Ventilating, and Air Conditioning Field.

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CODE *of* ETHICS *for* ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

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AIR CONDITIONING
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CHAPTER 1

Terminology

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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 −273.2 C or −459.8 F.

Absorbent: A material which has the ability to take up water vapor but which changes physically, chemically, or both, during the cycle.

Absorption: A process in which a fluid, liquid or gaseous, passes into the interstices of a porous substance and is held there by absorption or capillarity (G.P.)[a]

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 (second) (second) or 32.174 ft per (second) (second) which is the actual value of this acceleration at sea level and about 45 deg latitude.

Adiabatic: An adjective descriptive of a process in which no heat is added to or extracted from the system executing the process.

Adsorbent: A material which has the ability to hold water or other vapors on its internal surfaces without itself being permanently changed physically or chemically.

Adsorption: A term applied to the phenomena connected with the adherence of molecules of a foreign substance to the surface of a solid or a liquid (G.P.).

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 air-borne impurities such as dusts, gases, vapors, fumes and smokes. (Air cleaners include air washers, air filters, electrostatic precipitators and charcoal filters.)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors and toxic gases, most of which affect in greater or lesser degree human health or comfort. (*See Comfort Air Conditioning.*)

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 weighing 0.075 lb per cubic foot. This is the density at standard atmospheric pressure (29.921 in. Hg) of dry air at 69.41 F and of saturated air at 60.12 F.

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 due to the weight of the atmosphere. It is the pressure indicated by a barometer. *Standard Atmospheric Pressure* or *Standard Atmosphere* is the pressure of 76 cm of mercury having a density of 13.5951 grams per cubic centimeter, under standard gravity of 980.665 cm per (second) (second). It is equivalent to 14.696 lb per square inch or 29.921 in. of mercury at 32 F.

Baffle: A surface for deflecting gases or 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.

[a]From Glossary of Physics, by LeRoy Dougherty Weld (McGraw-Hill, 1937).

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: 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, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

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

British Thermal Unit: Classically the Btu is defined as the quantity of heat required to raise the temperature of 1 lb of water 1 Fahrenheit degree. By this definition the exact value depends upon the initial temperature of the water. Several values of the Btu are in more or less common use, each differing from the others by a slight amount. One of the more common of these is the *mean Btu* which is defined as 1/180 of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F at a constant atmospheric pressure of 14.696 lb per square inch absolute.

For most accurate work the *International Table* (I.T.) Btu is usually used. This is defined by the relation: 1 (I.T.) Btu per (pound) (Fahrenheit degree) = 1 (I.T.) calorie per (gram) (Centigrade degree). This value corresponds to the amount of heat required to raise the temperature of 1 lb of water 1 Fahrenheit degree at 58 F and also at 149 F. The *mean Btu* corresponds to 1.0008 (I.T.) Btu.

By-Pass: A pipe or duct, usually controlled by valve or damper, for conveying a fluid around an element of a system.

Calorie (Gram Calorie): Classically the calorie is defined as the quantity of heat required to raise the temperature of 1 gram of water 1 Centigrade degree. By this definition the exact value depends upon the initial temperature of the water. Several values of the calorie are in more or less common use, each differing from the others by a slight amount. Among these are the *15 C calorie* and the *17½ C calorie.* The *mean calorie, i.e.* 1/100 the quantity of heat required to raise the temperature of 1 gram of water from 0 C to 100 C, is also extensively used.

For the most accurate work the International Table (I.T.) calorie, defined in terms of the international electrical units, is usually used: 1 (I.T.) calorie = 1/860 international watt-hour = 3,600/860 international watt-seconds or international joules.

The kilocalorie = 1,000 cal.

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 43.)

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.

Comfort Air Conditioning: 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. (See *Air Conditioning.*)

Comfort Line: The effective temperature at which the largest percentage of adults feels comfortable.

Comfort Zone (*Average*): The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. (See Chapter 12.)

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 25.)

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

$$\text{db difference} = 10 \log_{10} \frac{P_2}{P_1}$$

In acoustics the threshold of hearing at 1,000 cycles per second has been standardized at 10^{-16} watts per square centimeter. If P_2 is the power in watts per square centimeter of a measured sound, then $10 \log_{10} \frac{P_2}{10^{-16}}$ 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.

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 exists 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 of a unit volume of a substance.

Dew Point: See *Temperature, Dew Point.*

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 condensation 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 10, p. 184.)

Enthalpy: A term applied to the Gibbs function, $h = u + pv$ and superseding *total heat, heat content,* and *heat of formation.* The increase of enthalpy of a system is equal

to the heat absorbed by the system during a process occurring at constant pressure when the only work done is that of compression or expansion (See also Chapter 3.)

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: Entropy is the ratio of the heat added to a substance to the absolute temperature at which it is added. The entropy associated with an isolated physical system has the characteristic property that, as the system spontaneously settles into a final, steady state, the entropy approaches a maximum. It may be regarded as a measure of the degree in which the energy of the system is unavailable.

If a system absorbs an infinitesimal amount of heat dQ during a reversible process, the entropy change of the system is equal to

$$dS = \frac{dQ}{T}$$

where T is the absolute temperature of the system.

Two of the functions of entropy may be mentioned to clarify this somewhat abstruse property: (1) During a reversible *adiabatic* change of state, entropy is constant; (2) during a reversible *isothermal* change of state, the heat absorbed is equal to the absolute temperature times the change in entropy.

The term *Specific Entropy* is sometimes applied to entropy per unit weight, the English unit being Btu per (Fahrenheit degree) (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 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 10, p. 185.)

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 10, p. 184.)

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. (*A.S.M.E.* 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: Thermal energy. Heat is a mode of motion. When a change in the quantity of heat in a body results in a change of temperature without change of state the heat is called *Sensible Heat.* When a change in the quantity of heat in a substance results in a change of state, *e.g.* from liquid to vapor, solid to liquid, etc., without change in temperature, the heat is called *Latent Heat.*

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 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, 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, 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.).

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 actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature. (See discussion in Chapter 3.)

Humidity, Constant Relative Line: Any line on the psychrometric chart representing a series of conditions which may be evaluated by one percentage of relative humidity; there are also constant dry-bulb lines, wet-bulb lines, effective temperature lines, vapor pressure lines, and lines showing other physical properties of air mixed with water vapor.

Humidity Ratio: In a mixture of water vapor and air, the weight of water vapor per pound of dry air. 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 square foot.

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). The unit EDR is becoming obsolete.

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. $m = \frac{F}{a}$. 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. $J = 778.3$ ft-lb per Btu $= 4.187 \times 10^7$ ergs per gram-calorie.

Medium, Heating: A substance such as water, steam, air or furnace gas used to convey heat from the boiler, furnace or other souce 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 $= 1.934 \times 10^{-2}$ lb per square inch.

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 ft^3), *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 (ft^3 per mol).

One-Pipe Supply Riser—(*Steam*): A pipe which carries steam vertically to a heating unit and which also carries the condensation from the heating unit. In an upfeed system steam and condensation 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 condensation 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.

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, 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, Dynamic: Same as Total 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 co-exist 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 difference, due to velocity, between total pressure and static pressure.

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.

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 wave length. Radiant energy of any wave length may, when absorbed, become thermal energy and result in an increase in the temperature of the absorbing body.

Radiation, Equivalent Direct (*EDR, Steam*): That amount of heating surface, expressed in square feet, which will deliver 240 Btu per hour, under the design operating conditions. (*EDR Hot Water*): That amount of heating surface, expressed in square feet, which will deliver 150 Btu per hour, under the design operating conditions. Thus, 1 sq ft of EDR does not imply 144 sq in. of heater surface, but means a heat delivery of 240 (or 150) Btu per hour for each EDR of a given radiator or convector.

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 single 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 minute, 12,000 Btu per hour, or 288,000 Btu per 24 hours.

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

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

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 the system are of practically equal length.

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, or Per Cent: The ratio of the weight of a given volume of water vapor to the weight of an equal volume of saturated water vapor 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.

Sorbent: Material which has the property of dehumidifying gases without the help of refrigeration. The term generally refers to materials which have a large capacity for moisture compared to their bulk or weight.

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

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 per cent relative humidity) for a given absolute humidity at constant pressure.

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, 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. (*A.S.M.E.* Power Test Codes, Series 1932, Instruments and Apparatus, Part 18.)

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)[b] (G.P.); (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 *Over-all 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.

Unit: As applied to heating, ventilating and air conditioning equipment this word means factory-built and assembled equipment with apparatus for accomplishing some specified function or combination of functions. (See Chapters 26 and 36.)

It is loosely applied to a great variety of equipment. Usually the function is included in the name, and hence come terms like Unit Heater, Unit Ventilator, Humidifying Unit, and Air Conditioning Unit.

Units are said to be *direct* or *room*, when intended for location, or located in, the treated space; *indirect* or *remote*, when outside or adjacent to the treated space. They are *ceiling* units when suspended from above, and *floor* when supported from below. Other descriptive words include *free delivery* when the unit is not intended to be attached to ducts or similar resistance-producing devices, and *pressure* when for use with such ducts. Complete description requires the use of several of these qualifying words or phases. (See Chapters 26 and 36.)

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. (*A.S.A.* definition.)

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation 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.

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[b]Names of authors who first stated law are given in parentheses).

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.)

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 the various rooms of the building through ducts.

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, Mechanical: A warm air heating system in which circulation of air is effected by a fan. Such a system may include air cleaning devices.

CHAPTER 2

Abbreviations and Symbols

Standard Abbreviations; Standard Symbols; Greek Alphabet; Conversion Equations; Graphical Symbols for Piping, Ductwork, Heating and Ventilating, Refrigerating; Identification of Piping by Color; Specific Heat Table

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. Additional abbreviations applying to individual chapters will be found at the end of Chapters 3, 4, 5, 7, 39, 41, and 43.

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, Réaumur	R
Diameter	diam

[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, be included in expressions for numerical temperatures but, wherever feasible, the abbreviations for *degree* be omitted; as 68 F.

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
Kilowatthour	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	oz
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

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

[3] Letter Symbols for Mechanics of Solid Bodies, Z10.3-1942, and Letter Symbols for Heat and Thermodynamics, Z10.4-1943 (*American Standards Association*).

symbols found necessary in the individual chapters will be found in a list at the end of Chapters 3, 4, 5, 7, 39, 41, and 43.

Acceleration, due to gravity	g
Acceleration, linear	a
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[4]: 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_a

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

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.)

Over-all coefficient of heat transfer, Thermal transmittance per unit area: heat transferred per (unit time) (unit area) (degree over-all)..... 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 of time..... w

Work (total)..... W

[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.

THE GREEK ALPHABET

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

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
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

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

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]

GRAPHICAL SYMBOLS FOR DRAWINGS — *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 — FOF
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
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

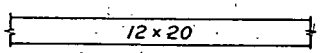
SPRINKLERS

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

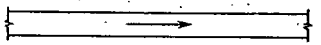
[6]Graphical Symbols for Use on Drawings in Mechanical Engineering, Z32.2-1941 (*American Standards Association*).

GRAPHICAL SYMBOLS FOR DRAWINGS — *Ductwork*

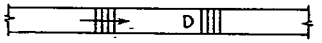
45. Duct (1st Figure, Width; 2nd, Depth)



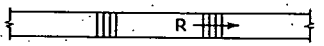
46. Direction of Flow



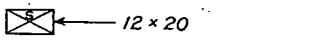
47. Inclined Drop in Respect to Air Flow



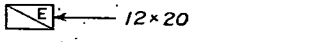
48. Inclined Rise in Respect to Air Flow



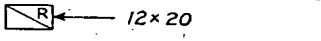
49. Supply Duct Section



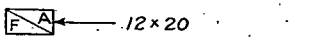
50. Exhaust Duct Section



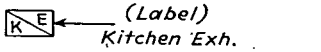
51. Recirculation Duct Section



52. Fresh Air Duct Section



53. Other Duct Sections



54. Register

R

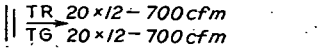
55. Grille

G

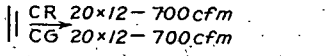
56. Supply Outlet

57. Exhaust Inlet

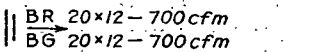
58. Top Register or Grille



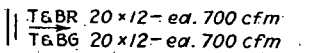
59. Center Register or Grille



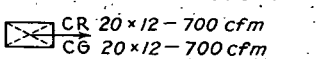
60. Bottom Register or Grille



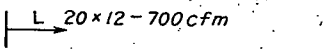
61. Top and Bottom Register or Grille



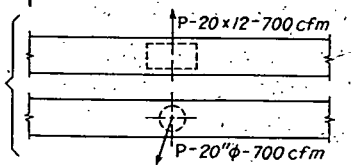
62. Ceiling Register or Grille



63. Louver Opening

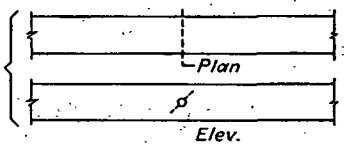


64. Adjustable Plaque

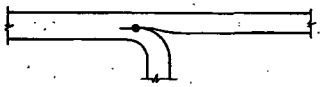


GRAPHICAL SYMBOLS FOR DRAWINGS — *Ductwork*

65. Volume Damper



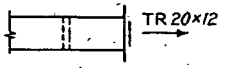
66. Deflecting Damper



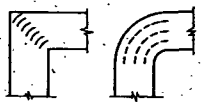
67. Deflecting Damper, Up

68. Deflecting Damper, Down

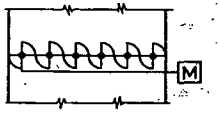
69. Adjustable Blank Off



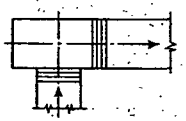
70. Turning Vanes



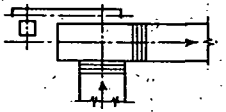
71. Automatic Dampers



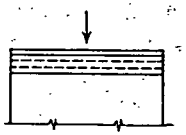
72. Canvas Connections



73. Fan and Motor With Guard

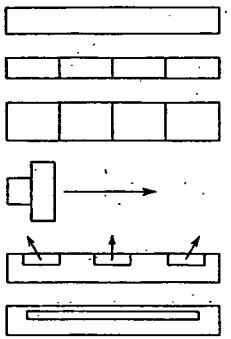


74. Intake Louvers and Screen



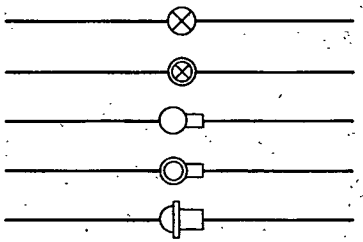
GRAPHICAL SYMBOLS FOR DRAWINGS — *Heating and Ventilating*

75. Heat Transfer Surface, Plan
76. Wall Radiator, Plan
77. Wall Radiator on Ceiling, Plan
78. Unit Heater (Propeller), Plan
79. Unit Heater (Centrifugal Fan), Plan
80. Unit Ventilator, Plan



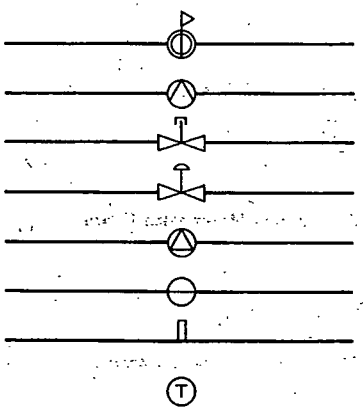
TRAPS

81. Thermostatic
82. Blast Thermostatic
83. Float and Thermostatic
84. Float
85. Boiler Return



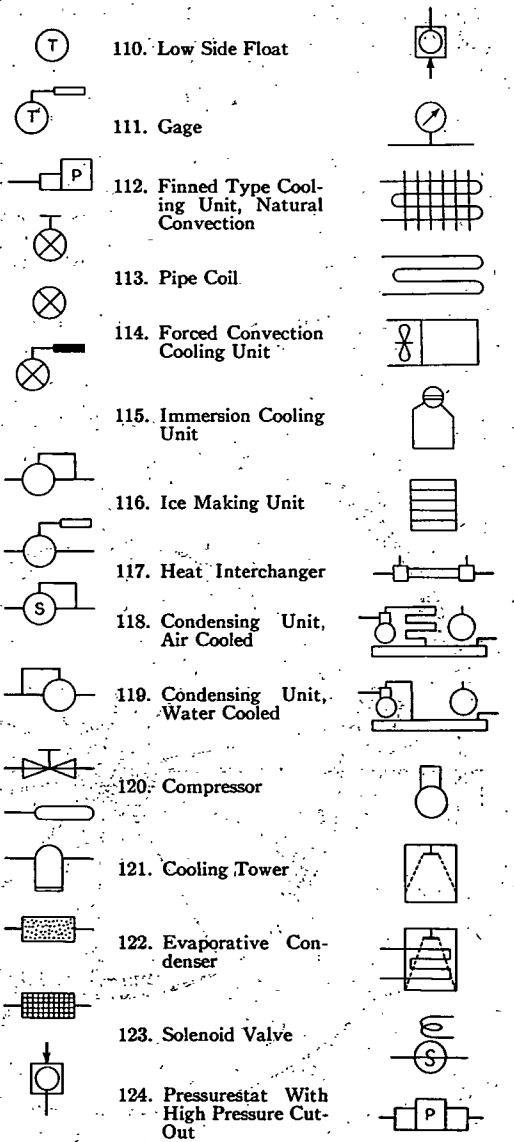
VALVES

86. Reducing Pressure
87. Air Line
88. Lock and Shield
89. Diaphragm
90. Air Eliminator
91. Strainer
92. Thermometer
93. Thermostat



GRAPHICAL SYMBOLS FOR DRAWINGS — *Refrigerating*

94. Thermostat (Self Contained)
95. Thermostat (Remote Bulb)
96. Pressurestat
97. Hand Expansion Valve
98. Automatic Expansion Valve
99. Thermostatic Expansion Valve
100. Evaporator Press. Regulating Valve, Throttling Type
101. Evaporator Press. Regulating Valve, Thermostatic Throttling Type
102. Evaporator Press. Regulating Valve, Snap-Action Valve
103. Compressor Suction Pressure Limiting Valve, Throttling Type
104. Hand Shut Off Valve
105. Thermal Bulb
106. Scale Trap
107. Dryer
108. Strainer
109. High Side Float
110. Low Side Float
111. Gage
112. Finned Type Cooling Unit, Natural Convection
113. Pipe Coil
114. Forced Convection Cooling Unit
115. Immersion Cooling Unit
116. Ice Making Unit
117. Heat Interchanger
118. Condensing Unit, Air Cooled
119. Condensing Unit, Water Cooled
120. Compressor
121. Cooling Tower
122. Evaporative Condenser
123. Solenoid Valve
124. Pressurestat With High Pressure Cut-Out



IDENTIFICATION OF PIPING SYSTEMS BY COLOR

The color scheme for identification of piping systems 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*.

All piping systems are classified according to the material carried in the pipes and colors are assigned as follows:

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

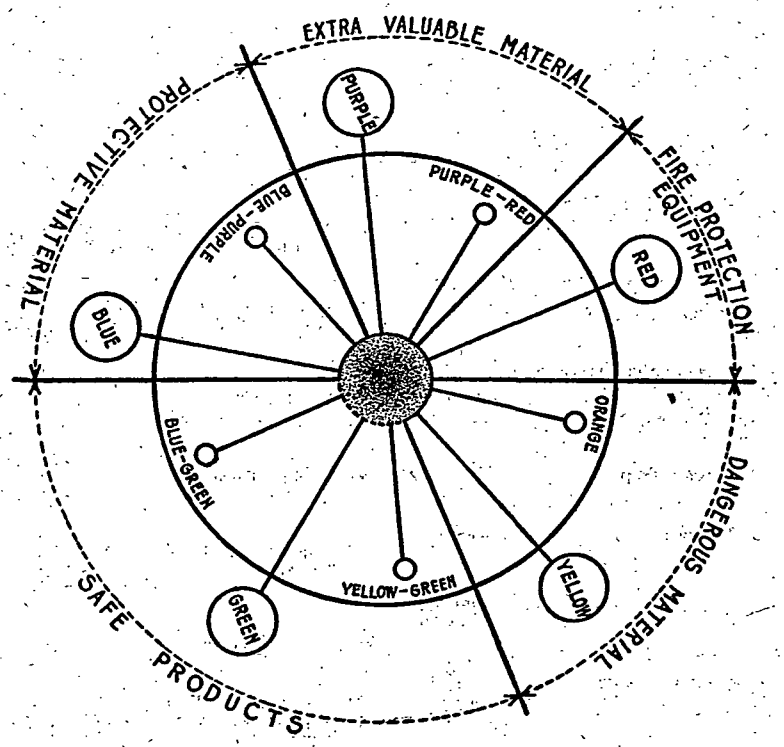


FIG. 1. MAIN CLASSIFICATION BY COLOR[a]

[a]From Scheme for Identification of Piping Systems, *Heating, Piping and Air Conditioning Contractors National Association*, Part V, Fourth Edition, p. 17. Used by permission.

SPECIFIC HEAT

TABLE 1. SPECIFIC HEAT OF SOLIDS[a]

Materials	Temperature F	Specific Heat	Authority
Alloys			
Brass, Red	32	0.0899	S
Brass, Yellow	32	0.0883	S
Bronze (80Cu, 20Sn)	57–208	0.0862	S
Monel Metal	68–2370	0.127	S
Aluminum	80–212	0.212	S
Asbestos	68–208	0.195	S
Brickwork	———	0.195	H
Carbon (Graphite)	104–1637	0.314	I
Coal	———	0.278	H
Coke	———	0.201	H
Concrete	———	0.270	H
Copper	64–212	0.0928	S
Fire Clay Brick	77–1832	0.258	I
Glass			
Crown	50–122	0.161	S
Flint	50–122	0.117	S
Gold	64	0.0312	S
Gypsum	———	0.259	H
Ice	32	0.487	S
Ice	−40	0.434	S
Iron, Pure	32	0.1043	S
Iron, Pure	32–600	0.127	M
Iron, Cast	68–212	0.1189	H
Iron, Wrought	59–212	0.1152	H
Lead	32	0.0297	S
Nickel	32	0.1032	S
Masonry	———	0.2159	H
Plaster	———	0.2	H
Platinum	58–212	0.0319	S
Rocks			
Gneiss	63–210	0.196	S
Granite	54–212	0.192	S
Limestone	59–212	0.216	S
Marble	32–212	0.21	S
Sandstone	———	0.22	S
Silver	32	0.0536	S
Steel	———	0.1175	H
Sulphur	240–320	0.220	S
Silica Brick	77–1832	0.263	I
Tin	77	0.0548	S
Woods (Average)	68	0.327	S
Zinc	32	0.0913	S

TABLE 2. SPECIFIC HEAT OF LIQUIDS

Liquid	Temperature F	Specific Heat	Authority
Alcohol, Ethyl	32	0.548	S
Alcohol, Methyl	59–68	0.601	S
Glycerine	59–122	0.576	S
Lead (Molten)	360	0.041	H
Mercury	68	0.03325	S
Petroleum	70–136	0.511	S
Sea Water			
Sp Gr 1.0043	64	0.980	S
Sp Gr 1.0463	64	0.903	S
Water	59	1.000	S

TABLE 3. SPECIFIC HEAT OF GASES AND VAPORS

Substance	Temperature F	Specific Heat at Constant Pressure	Ratio of Specific Heat C_p/C_v	Specific Heat at Constant Volume (Computed)	Authority
Air	32–392	0.2375	1.405	0.169	S
Ammonia	80–392	0.5356	1.277	0.419	S
Carbon Dioxide	52–417	0.2169	1.3003	0.1668	S
Carbon Monoxide	79–388	0.2426	1.395	0.1736	S
Coal Gas	68–1900	0.3145	———	———	S
Flue Gas	———	0.24 (Approx.)	———	———	H
Hydrogen	70–212	3.41	1.419	2.402	S
Nitrogen	32–392	0.2438	1.41	0.1729	S
Oxygen	55–404	0.2175	1.3977	0.155	S
Water Vapor	212	0.421	1.305	0.322	S
Water Vapor	356	0.51	———	———	S

[a]See also The Specific Heat of Thermal Insulating Materials, by Gordon B. Wilkes and Carl O. Wood (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 493).

NOTES: When one temperature is given the true specific heat is given, otherwise the value is the mean specific heat between the given limits.

AUTHORITIES: S—Smithsonian Physical Tables. 1933; I—International Critical Tables; H—Heating, Ventilation and Air Conditioning, by L. A. Harding and A. C. Willard; M—Engineers' Handbook, by Lionel S. Marks.

CHAPTER 3

Thermodynamics

Degree of Saturation; Mollier Diagram; Derived Properties; Typical Air Conditioning Processes, Heating, Cooling, Adiabatic Mixing; Wet-Bulb Temperatures Below 32 F; Dalton's Rule; Steady Flow Energy Equation, U. S. Standard Atmosphere

THE working substance of the air conditioning engineer is called *moist air*. In order to be able to apply the laws of conservation of energy and mass to the analysis of typical air conditioning processes, it is necessary to know the thermodynamic properties of moist air, particularly its enthalpy and volume. When the limitations imposed by the Second Law of Thermodynamics have to be considered, it is also necessary to know its entropy.

For the purpose of analysis, moist air may be regarded as a mixture of only two constituents, namely, dry air and water vapor. It has long been customary to predict the thermodynamic properties of the mixture from a knowledge of those of dry air and water vapor separately by means of Dalton's Rule. According to this rule: each constituent of a gas mixture occupies the whole volume of the mixture just as if no other constituent were present; it therefore exerts a *partial pressure* equal to the pressure it would exert if alone in the whole volume at the temperature of the mixture; the observed pressure of the mixture is the sum of these so-called partial pressures; the enthalpy of the mixture is the sum of separate contributions from the individual constituents as determined by their partial pressures, their weights, and the temperature of the mixture; and the entropy of the mixture is obtained in a similar manner.

Dalton's Rule has long been regarded erroneously as a fundamental law of nature. Actually it is not, and in many cases its predictions are quite unreliable. In the case of moist air at atmospheric pressure, it happens to give a close approximation to the truth; but as progress is made the need for greater accuracy than the rule can afford is felt even in this case. Fortunately most of the complications involved in following a correct procedure based on the predictions of statistical mechanics are met in preparing suitable tables of thermodynamic properties; and, once these tables have been prepared, their use in the analysis of typical air conditioning processes is actually simplified by abandonment of the rule together with its fictitious concepts of partial pressure, relative humidity, etc.

Thermodynamic Properties of Moist Air

Table 1, Thermodynamic Properties of Moist Air (Standard Atmospheric Pressure, 29.921 In. Hg), contains results of a cooperative investigation between the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Towne Scientific School, University of Pennsylvania. These results are to be considered by an International Joint Committee on Psychrometric Data as a possible starting point from which to reach agreement on standard properties of moist air. A detailed explanation of the data and methods used in constructing Table 1 is given in a paper [1] recently presented before the ASHVE upon the recommendation of its

Technical Advisory Committee on Psychrometry as a final report of the cooperative investigation.

In Table 1 there are 15 columns of figures, each column being headed by a suitable symbol. In the following sub-paragraphs are given brief explanations of the data in Table 1 under the appropriate column headings.

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

$$T = t + 459.69 \tag{1}$$

This particular Fahrenheit scale differs slightly from that derived from the International Centigrade Scale t(C) by the definition,

$$t(\text{F}) = 1.8t(\text{C}) + 32 \tag{2}$$

However, the maximum difference between the two Fahrenheit scales appears not to exceed 0.01 Fahrenheit deg in the range 32 to 212 F.

W_s = humidity ratio *at saturation*. By humidity ratio is meant the ratio, by weight, of water vapor to dry air, pounds of water vapor per pound of dry air. By saturation is meant the point where coexistence of the vapor phase (moist air) with a condensed phase (liquid or solid) is possible at the given temperature and pressure (standard atmospheric pressure in the case of Table 1). 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.

$v_{as} = v_s - v_a$, 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 = 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. It will be noticed that 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, though not exactly by definition, as follows: 1 Btu = 778.18 ft-lb.

$h_{as} = h_s - h_a$, 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_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) (F). It will be noticed that the specific entropy of dry air has been assigned the value zero at 0 F and standard atmospheric pressure.

$s_{as} = s_s - s_a$, 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) (F).

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

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. It will be noticed that, under this assignment, the specific enthalpy of liquid water at 32 F, standard atmospheric pressure, assumes the value 0.04 Btu/lb$_w$.

s_w = specific entropy of condensed water (liquid or solid) at standard atmospheric pressure. Btu per (pound of water) (F). The specific entropy of liquid water has been assigned the value zero at 32 F, saturation pressure. It will be noticed that, under this assignment, the specific entropy of liquid water at 32 F, standard atmospheric pressure, is also zero, though not exactly.

p_s = saturation pressure of pure water vapor, pounds per square inch or inches of Hg. Moist air can be saturated at any given values of temperature and pressure, though this requires that it have a definite humidity ratio W_s and that the coexisting condensed phase contain a definite, but very small quantity of dissolved air. On the other hand, pure water vapor (steam) cannot be saturated at any given values of temperature and pressure because its composition is invariable. It can, however, be saturated at any

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG)

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^3$	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/(Lb) (°F) s_w	Vap. Press In. Hg $p_s \times 10^3$	
−160	0.2120	7.520	0.000	7.520	−38.504	0.000	−38.504	−0.10300	0.00000	−0.10300	−222.00	−0.4907	0.1009	−160
−159	0.2394	7.545	0.000	7.545	−38.262	0.000	−38.262	−0.10219	0.00000	−0.10219	−221.68	−0.4896	0.1139	−159
−158	0.2703	7.571	0.000	7.571	−38.021	0.000	−38.021	−0.10139	0.00000	−0.10139	−221.36	−0.4885	0.1286	−158
−157	0.3049	7.596	0.000	7.596	−37.779	0.000	−37.779	−0.10059	0.00000	−0.10059	−221.04	−0.4874	0.1450	−157
−156	0.3435	7.622	0.000	7.622	−37.538	0.000	−37.538	−0.09980	0.00000	−0.09980	−220.72	−0.4864	0.1635	−156
−155	0.3869	7.647	0.000	7.647	−37.296	0.000	−37.296	−0.09901	0.00000	−0.09901	−220.40	−0.4853	0.1842	−155
−154	0.4354	7.673	0.000	7.673	−37.055	0.000	−37.055	−0.09822	0.00000	−0.09822	−220.07	−0.4843	0.2073	−154
−153	0.4897	7.699	0.000	7.699	−36.813	0.000	−36.813	−0.09743	0.00000	−0.09743	−219.75	−0.4832	0.2331	−153
−152	0.5502	7.724	0.000	7.724	−36.572	0.000	−36.572	−0.09664	0.00000	−0.09664	−219.42	−0.4822	0.2620	−152
−151	0.6178	7.750	0.000	7.750	−36.330	0.000	−36.330	−0.09586	0.00000	−0.09586	−219.10	−0.4811	0.2942	−151
−150	0.6932	7.775	0.000	7.775	−36.088	0.000	−36.088	−0.09508	0.00000	−0.09508	−218.77	−0.4800	0.3301	−150
−149	0.7772	7.801	0.000	7.801	−35.847	0.000	−35.847	−0.09430	0.00000	−0.09430	−218.44	−0.4789	0.3701	−149
−148	0.8709	7.826	0.000	7.826	−35.606	0.000	−35.606	−0.09352	0.00000	−0.09352	−218.11	−0.4779	0.4146	−148
−147	0.9750	7.851	0.000	7.851	−35.364	0.000	−35.364	−0.09274	0.00000	−0.09274	−217.78	−0.4768	0.4641	−147
−146	1.091	7.876	0.000	7.876	−35.123	0.000	−35.123	−0.09198	0.00000	−0.09198	−217.45	−0.4758	0.5194	−146
−145	1.219	7.902	0.000	7.902	−34.881	0.000	−34.881	−0.09121	0.00000	−0.09121	−217.12	−0.4747	0.5807	−145
−144	1.362	7.927	0.000	7.927	−34.640	0.000	−34.640	−0.09044	0.00000	−0.09044	−216.78	−0.4737	0.6488	−144
−143	1.521	7.953	0.000	7.953	−34.399	0.000	−34.399	−0.08967	0.00000	−0.08967	−216.45	−0.4726	0.7243	−143
−142	1.698	7.978	0.000	7.978	−34.157	0.000	−34.157	−0.08892	0.00000	−0.08892	−216.11	−0.4716	0.8082	−142
−141	1.893	8.004	0.000	8.004	−33.916	0.000	−33.916	−0.08816	0.00000	−0.08816	−215.78	−0.4705	0.9011	−141
−140	2.109	8.029	0.000	8.029	−33.674	0.000	−33.674	−0.08740	0.00000	−0.08740	−215.44	−0.4695	1.004	−140
−139	2.348	8.054	0.000	8.054	−33.433	0.000	−33.433	−0.08664	0.00000	−0.08664	−215.11	−0.4684	1.118	−139
−138	2.613	8.079	0.000	8.079	−33.192	0.000	−33.192	−0.08589	0.00000	−0.08589	−214.77	−0.4674	1.244	−138
−137	2.906	8.105	0.000	8.105	−32.951	0.000	−32.951	−0.08514	0.00000	−0.08514	−214.43	−0.4663	1.383	−137
−136	3.229	8.130	0.000	8.130	−32.709	0.000	−32.709	−0.08440	0.00000	−0.08440	−214.09	−0.4653	1.537	−136
−135	3.586	8.156	0.000	8.156	−32.468	0.000	−32.468	−0.08365	0.00000	−0.08365	−213.75	−0.4642	1.707	−135
−134	3.980	8.181	0.000	8.181	−32.226	0.000	−32.226	−0.08291	0.00000	−0.08291	−213.41	−0.4632	1.895	−134
−133	4.414	8.207	0.000	8.207	−31.985	0.000	−31.985	−0.08217	0.00000	−0.08217	−213.07	−0.4621	2.102	−133
−132	4.893	8.232	0.000	8.232	−31.744	0.000	−31.744	−0.08144	0.00000	−0.08144	−212.72	−0.4611	2.330	−132
−131	5.419	8.258	0.000	8.258	−31.503	0.000	−31.503	−0.08070	0.00000	−0.08070	−212.38	−0.4600	2.581	−131
−130	6.000	8.283	0.000	8.283	−31.262	0.000	−31.262	−0.07997	0.00000	−0.07997	−212.03	−0.4590	2.858	−130
−129	6.637	8.309	0.000	8.309	−31.021	0.000	−31.021	−0.07924	0.00000	−0.07924	−211.68	−0.4579	3.182	−129

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

TABLE 1. 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			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/(Lb) (°F) s_w	Vap. Press In. Hg $p_s \times 10^3$	
−128	0.7339	8.334	0.000	8.334	−30.780	0.000	−30.780	−0.07851	0.00000	−0.07851	−211.33	−0.4569	0.3492	−128
−127	0.8111	8.360	0.000	8.360	−30.539	0.000	−30.539	−0.07778	0.00000	−0.07778	−210.98	−0.4559	0.3863	−127
−126	0.8958	8.385	0.000	8.385	−30.298	0.000	−30.298	−0.07707	0.00000	−0.07707	−210.63	−0.4548	0.4267	−126
−125	0.9887	8.411	0.000	8.411	−30.057	0.000	−30.057	−0.07634	0.00000	−0.07634	−210.28	−0.4538	0.4710	−125
−124	1.091	8.436	0.000	8.436	−29.816	0.000	−29.816	−0.07562	0.00000	−0.07562	−209.93	−0.4527	0.5197	−124
−123	1.202	8.461	0.000	8.461	−29.575	0.000	−29.575	−0.07490	0.00000	−0.07490	−209.58	−0.4517	0.5730	−123
−122	1.325	8.486	0.000	8.486	−29.334	0.000	−29.334	−0.07419	0.00000	−0.07419	−209.23	−0.4506	0.6314	−122
−121	1.459	8.512	0.000	8.512	−29.093	0.000	−29.093	−0.07348	0.00000	−0.07348	−208.88	−0.4496	0.6953	−121
−120	1.606	8.537	0.000	8.537	−28.852	0.000	−28.852	−0.07277	0.00000	−0.07277	−208.52	−0.4485	0.7653	−120
−119	1.767	8.563	0.000	8.563	−28.611	0.000	−28.611	−0.07206	0.00000	−0.07206	−208.17	−0.4475	0.8419	−119
−118	1.942	8.588	0.000	8.588	−28.370	0.000	−28.370	−0.07135	0.00000	−0.07135	−207.81	−0.4464	0.9256	−118
−117	2.134	8.613	0.000	8.613	−28.129	0.000	−28.129	−0.07064	0.00000	−0.07064	−207.45	−0.4454	1.017	−117
−116	2.343	8.639	0.000	8.639	−27.889	0.000	−27.889	−0.06994	0.00000	−0.06994	−207.09	−0.4444	1.117	−116
−115	2.571	8.664	0.000	8.664	−27.648	0.000	−27.648	−0.06924	0.00000	−0.06924	−206.73	−0.4433	1.226	−115
−114	2.820	8.690	0.000	8.690	−27.407	0.000	−27.407	−0.06854	0.00000	−0.06854	−206.37	−0.4423	1.345	−114
−113	3.092	8.715	0.000	8.715	−27.166	0.000	−27.166	−0.06784	0.00000	−0.06784	−206.01	−0.4412	1.475	−113
−112	3.388	8.741	0.000	8.741	−26.926	0.000	−26.926	−0.06715	0.00000	−0.06715	−205.65	−0.4402	1.617	−112
−111	3.711	8.766	0.000	8.766	−26.685	0.000	−26.685	−0.06646	0.00000	−0.06646	−205.29	−0.4392	1.771	−111
−110	4.063	8.792	0.000	8.792	−26.444	0.000	−26.444	−0.06577	0.00000	−0.06577	−204.92	−0.4381	1.939	−110
−109	4.445	8.817	0.000	8.817	−26.204	0.000	−26.204	−0.06508	0.00000	−0.06508	−204.46	−0.4371	2.121	−109
−108	4.861	8.842	0.000	8.842	−25.963	0.001	−25.962	−0.06439	0.00000	−0.06439	−204.19	−0.4360	2.320	−108
−107	5.314	8.868	0.000	8.868	−25.772	0.001	−25.721	−0.06370	0.00000	−0.06370	−203.83	−0.4350	2.536	−107
−106	5.806	8.893	0.000	8.893	−25.481	0.001	−25.480	−0.06302	0.00000	−0.06302	−203.46	−0.4339	2.771	−106
−105	6.340	8.919	0.000	8.919	−25.240	0.001	−25.239	−0.06234	0.00000	−0.06234	−203.09	−0.4329	3.026	−105
−104	6.920	8.944	0.000	8.944	−25.000	0.001	−24.999	−0.06167	0.00000	−0.06167	−202.72	−0.4318	3.303	−104
−103	7.549	8.970	0.000	8.970	−24.759	0.001	−24.758	−0.06099	0.00000	−0.06099	−202.35	−0.4308	3.603	−103
−102	8.232	8.995	0.000	8.995	−24.518	0.001	−24.517	−0.06032	0.00000	−0.06032	−201.98	−0.4298	3.929	−102
−101	8.972	9.020	0.000	9.020	−24.278	0.001	−24.277	−0.05964	0.00000	−0.05964	−201.61	−0.4287	4.283	−101
−100	9.772	9.046	0.000	9.046	−24.037	0.001	−24.036	−0.05897	0.00000	−0.05897	−201.23	−0.4277	4.666	−100
−99	10.63	9.071	0.000	9.071	−23.797	0.001	−23.796	−0.05830	0.00000	−0.05830	−200.86	−0.4266	5.081	−99
−98	11.57	9.097	0.000	9.097	−23.556	0.001	−23.555	−0.05764	0.00000	−0.05764	−200.48	−0.4256	5.530	−98
−97	12.59	9.122	0.000	9.122	−23.316	0.001	−23.315	−0.05697	0.00000	−0.05697	−200.11	−0.4245	6.016	−97

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

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^6$	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/(Lb) (°F) s_w	Vap. Press In. Hg $p_s \times 10^4$	
−93	1.369	9.147	0.000	9.147	−23.075	0.001	−23.074	−0.05631	0.00000	−0.05631	−199.73	−0.4235	0.6542	−96
−95	1.489	9.173	0.000	9.173	−22.835	0.002	−22.833	−0.05565	0.00000	−0.05565	−199.35	−0.4225	0.7111	−95
−94	1.617	9.198	0.000	9.198	−22.594	0.002	−22.592	−0.05500	0.00000	−0.05500	−198.97	−0.4214	0.7725	−94
−93	1.756	9.224	0.000	9.224	−22.353	0.002	−22.341	−0.05434	0.00000	−0.05434	−198.59	−0.4204	0.8388	−93
−92	1.906	9.249	0.000	9.249	−22.113	0.002	−22.111	−0.05369	0.00001	−0.05368	−198.21	−0.4193	0.9105	−92
−91	2.068	9.275	0.000	9.275	−21.872	0.002	−21.870	−0.05303	0.00001	−0.05302	−197.83	−0.4183	0.9879	−91
−90	2.242	9.300	0.000	9.300	−21.631	0.002	−21.629	−0.05237	0.00001	−0.05236	−197.44	−0.4173	1.071	−90
−89	2.430	9.325	0.000	9.325	−21.391	0.003	−21.388	−0.05172	0.00001	−0.05171	−197.06	−0.4162	1.161	−89
−88	2.634	9.351	0.000	9.351	−21.150	0.003	−21.147	−0.05107	0.00001	−0.05106	−196.67	−0.4152	1.259	−88
−87	2.852	9.376	0.000	9.376	−20.909	0.003	−20.906	−0.05042	0.00001	−0.05041	−196.29	−0.4142	1.363	−87
−86	3.089	9.401	0.000	9.401	−20.669	0.003	−20.666	−0.04978	0.00001	−0.04977	−195.90	−0.4131	1.476	−86
−85	3.342	9.426	0.000	9.426	−20.428	0.003	−20.425	−0.04913	0.00001	−0.04912	−195.51	−0.4121	1.597	−85
−84	3.615	9.451	0.000	9.451	−20.188	0.004	−20.184	−0.04849	0.00001	−0.04848	−195.12	−0.4110	1.728	−84
−83	3.909	9.477	0.000	9.477	−19.947	0.004	−19.943	−0.04785	0.00001	−0.04784	−194.73	−0.4100	1.868	−83
−82	4.225	9.502	0.000	9.502	−19.706	0.004	−19.702	−0.04721	0.00001	−0.04720	−194.34	−0.4090	2.019	−82
−81	4.564	9.527	0.000	9.527	−19.466	0.005	−19.461	−0.04658	0.00001	−0.04657	−193.95	−0.4079	2.181	−81
−80	4.930	9.553	0.000	9.553	−19.225	0.005	−19.220	−0.04595	0.00001	−0.04594	−193.55	−0.4069	2.356	−80
−79	5.322	9.578	0.000	9.578	−18.984	0.005	−18.979	−0.04532	0.00002	−0.04530	−193.16	−0.4059	2.543	−79
−78	5.742	9.604	0.000	9.604	−18.744	0.006	−18.738	−0.04469	0.00002	−0.04467	−192.76	−0.4048	2.744	−78
−77	6.193	9.629	0.000	9.629	−18.503	0.006	−18.497	−0.04406	0.00002	−0.04404	−192.37	−0.4038	2.960	−77
−76	6.677	9.654	0.000	9.654	−18.263	0.007	−18.256	−0.04343	0.00002	−0.04341	−191.97	−0.4027	3.192	−76
−75	7.196	9.680	0.000	9.680	−18.022	0.007	−18.015	−0.04280	0.00002	−0.04278	−191.57	−0.4017	3.441	−75
−74	7.753	9.705	0.000	9.705	−17.782	0.008	−17.774	−0.04218	0.00003	−0.04215	−191.17	−0.4007	3.707	−74
−73	8.349	9.730	0.000	9.730	−17.541	0.008	−17.533	−0.04155	0.00003	−0.04152	−190.77	−0.3996	3.992	−73
−72	8.990	9.756	0.000	9.756	−17.301	0.009	−17.292	−0.04093	0.00003	−0.04090	−190.37	−0.3986	4.298	−72
−71	9.675	9.781	0.000	9.781	−17.060	0.010	−17.050	−0.04031	0.00003	−0.04028	−189.97	−0.3975	4.625	−71
−70	10.40	9.806	0.000	9.806	−16.820	0.011	−16.809	−0.03969	0.00003	−0.03966	−189.56	−0.3965	4.976	−70
−69	11.19	9.831	0.000	9.831	−16.579	0.011	−16.568	−0.03908	0.00004	−0.03904	−189.16	−0.3954	5.351	−69
−68	12.03	9.856	0.000	9.856	−16.339	0.012	−16.327	−0.03846	0.00004	−0.03842	−188.75	−0.3944	5.752	−68
−67	12.92	9.882	0.000	9.882	−16.098	0.013	−16.085	−0.03785	0.00004	−0.03781	−188.35	−0.3934	6.181	−67
−66	13.88	9.907	0.000	9.907	−15.858	0.014	−15.844	−0.03724	0.00004	−0.03720	−187.94	−0.3924	6.640	−66
−65	14.91	9.932	0.000	9.932	−15.617	0.015	−15.602	−0.03663	0.00005	−0.03658	−187.53	−0.3913	7.130	−65

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

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^6$	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/(Lb) (°F) s_w	Vap. Press In. Hg $p_s \times 10^4$	
−64	1.601	9.958	0.000	9.958	−15.377	0.016	−15.361	−0.03602	0.00005	−0.03597	−187.12	−0.3903	0.7654	−64
−63	1.718	9.983	0.000	9.983	−15.137	0.018	−15.119	−0.03541	0.00005	−0.03536	−186.71	−0.3893	0.8213	−63
−62	1.843	10.009	0.000	10.009	−14.896	0.019	−14.877	−0.03481	0.00006	−0.03475	−186.30	−0.3882	0.8810	−62
−61	1.976	10.034	0.000	10.034	−14.656	0.020	−14.636	−0.03420	0.00006	−0.03414	−185.89	−0.3872	0.9447	−61
−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
−59	2.269	10.085	0.000	10.085	−14.175	0.023	−14.152	−0.03300	0.00007	−0.03293	−185.06	−0.3851	1.0852	−59
−58	2.431	10.110	0.000	10.110	−13.935	0.025	−13.910	−0.03240	0.00007	−0.03233	−184.64	−0.3841	1.1624	−58
−57	2.603	10.135	0.000	10.135	−13.695	0.027	−13.668	−0.03180	0.00008	−0.03172	−184.23	−0.3830	1.2447	−57
−56	2.786	10.161	0.000	10.161	−13.454	0.029	−13.425	−0.03120	0.00008	−0.03112	−183.81	−0.3820	1.3324	−56
−55	2.982	10.186	0.000	10.186	−13.214	0.031	−13.183	−0.03061	0.00009	−0.03052	−183.39	−0.3810	1.4258	−55
−54	3.190	10.211	0.000	10.211	−12.974	0.033	−12.941	−0.03002	0.00009	−0.02993	−182.97	−0.3799	1.5253	−54
−53	3.411	10.237	0.000	10.237	−12.733	0.035	−12.698	−0.02943	0.0010	−0.02933	−182.55	−0.3789	1.6312	−53
−52	3.646	10.262	0.001	10.263	−12.493	0.038	−12.455	−0.02884	0.00011	−0.02873	−182.13	−0.3779	1.7438	−52
−51	3.897	10.288	0.001	10.289	−12.253	0.041	−12.212	−0.02825	0.00011	−0.02814	−181.71	−0.3769	1.8635	−51
−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
−49	4.446	10.338	0.001	10.339	−11.772	0.046	−11.726	−0.02707	0.00012	−0.02695	−180.87	−0.3748	2.1264	−49
−48	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
−47	5.066	10.389	0.001	10.390	−11.292	0.053	−11.239	−0.02590	0.00013	−0.02577	−180.02	−0.3728	2.4230	−47
−46	5.406	10.414	0.001	10.415	−11.051	0.056	−10.995	−0.02532	0.00014	−0.02518	−179.59	−0.3717	2.5854	−46
−45	5.766	10.440	0.001	10.441	−10.811	0.060	−10.751	−0.02474	0.00015	−0.02459	−179.16	−0.3707	2.7578	−45
−44	6.149	10.465	0.001	10.466	−10.571	0.064	−10.507	−0.02416	0.00016	−0.02400	−178.73	−0.3696	2.9408	−44
−43	6.555	10.490	0.001	10.491	−10.330	0.068	−10.262	−0.02358	0.00017	−0.02341	−178.30	−0.3686	3.1349	−43
−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
−41	7.441	10.541	0.001	10.542	−9.850	0.078	−9.772	−0.02243	0.00020	−0.02223	−177.44	−0.3665	3.5591	−41
−40	7.925	10.566	0.001	10.567	−9.609	0.083	−9.526	−0.02186	0.00021	−0.02165	−177.01	−0.3655	3.7906	−40
−39	8.437	10.592	0.001	10.593	−9.369	0.089	−9.280	−0.02129	0.00022	−0.02107	−176.58	−0.3645	4.0359	−39
−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
−37	9.556	10.642	0.002	10.644	−8.889	0.100	−8.789	−0.02015	0.00025	−0.01990	−175.71	−0.3624	4.5711	−37
−36	10.16	10.668	0.002	10.670	−8.648	0.106	−8.542	−0.01958	0.00026	−0.01932	−175.27	−0.3614	4.8626	−36
−35	10.81	10.693	0.002	10.695	−8.408	0.113	−8.295	−0.01902	0.00028	−0.01874	−174.84	−0.3604	5.1713	−35
−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
−33	12.21	10.744	0.002	10.746	−7.927	0.128	−7.799	−0.01789	0.00032	−0.01757	−173.96	−0.3583	5.8437	−33

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

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO $W_s \times 10^4$	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/(Lb) (°F) s_w	Vap. Press In. Hg $p_s \times 10^3$	
−32	1.298	10.769	0.002	10.771	−7.687	0.136	−7.551	−0.01733	0.00034	−0.01699	−173.52	−0.3573	0.62093	−32
−31	1.378	10.794	0.002	10.796	−7.447	0.145	−7.302	−0.01677	0.00036	−0.01641	−173.08	−0.3563	0.65979	−31
−30	1.464	10.820	0.002	10.822	−7.207	0.154	−7.053	−0.01621	0.00038	−0.01583	−172.64	−0.3552	0.70046	−30
−29	1.554	10.845	0.003	10.848	−6.966	0.163	−6.803	−0.01565	0.00040	−0.01525	−172.20	−0.3542	0.74365	−29
−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	0.78928	−28
−27	1.750	10.896	0.003	10.899	−6.486	0.184	−6.302	−0.01453	0.00045	−0.01408	−171.31	−0.3521	0.83748	−27
−26	1.856	10.921	0.003	10.924	−6.246	0.196	−6.050	−0.01398	0.00048	−0.01350	−170.86	−0.3511	0.88838	−26
−25	1.969	10.946	0.004	10.950	−6.005	0.207	−5.798	−0.01342	0.00051	−0.01291	−170.42	−0.3500	0.94212	−25
−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
−23	2.212	10.997	0.004	11.001	−5.525	0.232	−5.293	−0.01232	0.00057	−0.01175	−169.52	−0.3480	1.0587	−23
−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
−21	2.483	11.048	0.004	11.052	−5.044	0.261	−4.783	−0.01122	0.00064	−0.01058	−168.62	−0.3459	1.1885	−21
−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
−19	2.785	11.098	0.005	11.103	−4.564	0.293	−4.271	−0.01012	0.00072	−0.00940	−167.72	−0.3439	1.3327	−19
−18	2.948	11.124	0.005	11.129	−4.324	0.310	−4.014	−0.00958	0.00076	−0.00882	−167.26	−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	0.0006	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	0.412	−2.711	−0.00689	0.00099	−0.00590	−164.98	−0.3377	1.8677	−13
−12	4.125	11.275	0.008	11.283	−2.882	0.436	−2.446	−0.00636	0.00104	−0.00532	−164.52	−0.3367	1.9740	−12
−11	4.359	11.301	0.008	11.309	−2.642	0.461	−2.181	−0.00582	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	0.00115	−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	−1.922	0.543	−1.379	−0.00422	0.00128	−0.00294	−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	0.00142	−0.00174	−161.74	−0.3305	2.7377	−6
−5	6.040	11.452	0.011	11.463	−1.201	0.639	−0.562	−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	0.00165	0.00008	−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	−0.00052	0.00183	0.00131	−159.40	−0.3254	3.5720	−1

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

TABLE 1. 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			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/(Lb) (°F) s_w	Vap. Press In. Hg $p_s \times 10^3$	
0	0.7872	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	0.8295	11.604	0.015	11.619	0.240	0.880	1.120	0.00052	0.00202	0.00254	−158.46	−0.3234	3.9686	1
2	0.8739	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	0.9204	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	0.9692	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	1.020	11.705	0.019	11.724	1.201	1.085	2.286	0.00260	0.00246	0.00506	−156.57	−0.3193	4.8779	5
6	1.074	11.730	0.020	11.750	1.441	1.142	2.583	0.00312	0.00258	0.00570	−156.09	−0.3182	5.1339	6
7	1.130	11.756	0.021	11.777	1.681	1.202	2.883	0.00364	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	5.6832	8
9	1.251	11.806	0.024	11.830	2.162	1.332	3.494	0.00467	0.00299	0.00766	−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	6.2858	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	6.6085	11
12	1.454	11.882	0.028	11.910	2.882	1.550	4.432	0.00620	0.00346	0.00966	−153.21	−0.3121	6.9462	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	7.2997	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	1.772	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	0.01382	−150.29	−0.3059	9.3267	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	9.7889	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	10.272	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	2.606	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	0.01273	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	0.01322	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	0.01372	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	14.996	28
29	3.297	12.312	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	0.075	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	4.242	12.169	0.01666	0.00904	0.02570	1.05	0.0020	18.778	33
34	4.107	12.438	0.082	12.520	8.167	4.418	12.585	0.01715	0.00940	0.02655	2.06	0.0041	19.546	34

[a]Compiled by John A. Goff and S. Gratch.
*Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 1. 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			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/(Lb) (°F) s_w	Vap. Press In. Hg p_s	
35	4.275	12.464	0.085	12.549	8.407	4.601	13.008	0.01764	0.00977	0.02741	3.06	0.00061	0.20342	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	0.21166	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	0.22020	37
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	0.22904	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	0.23819	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	0.24767	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	0.25748	41
42	5.638	12.641	0.114	12.755	10.088	6.084	16.172	0.02101	0.01275	0.03376	10.09	0.0202	0.26763	42
43	5.860	12.666	0.119	12.785	10.329	6.328	16.657	0.02149	0.01323	0.03472	11.10	0.0222	0.27813	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	0.28899	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	0.30023	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	0.31185	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	0.32386	47
48	7.100	12.792	0.146	12.938	11.530	7.681	19.211	0.02387	0.01591	0.03978	16.11	0.0321	0.33629	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	13.032	12.250	8.612	20.862	0.02528	0.01774	0.04302	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	0.45176	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	14.172	11.61	25.78	0.02902	0.02357	0.05259	27.12	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	28.12	0.0555	0.52159	60
61	11.49	13.121	0.242	13.363	14.653	12.50	27.15	0.02994	0.02527	0.05521	29.12	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	30.12	0.0594	0.55994	62
63	12.35	13.172	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	0.03132	0.02803	0.05935	32.12	0.0632	0.60073	64
65	13.26	13.222	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	32.42	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.

TABLE 1. 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			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/(Lb) (°F) 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	0.0746	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	0.0765	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	0.0784	0.79112	72
73	1.757	13.424	0.377	13.801	17.537	19.20	36.74	0.03540	0.03800	0.07340	41.11	0.0803	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	0.96665	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	2.555	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	22.587	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	22.827	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	0.04686	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	52.59	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	54.32	79.31	0.04900	0.1020	0.1510	72.05	0.1367	2.1786	104

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

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

FAHR. TEMP. t(F)	HUMIDITY RATIO W_s x 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/(Lb) (°F) 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	25.472	57.95	83.42	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	0.1118	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	0.1226	0.1742	78.03	0.1472	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	0.6333	14.409	1.460	15.869	26.915	70.27	97.18	0.05239	0.1302	0.1826	80.03	0.1508	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	0.1963	83.02	0.1560	2.9962	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	0.1562	0.2111	86.02	0.1612	3.2603	118
119	0.7897	14.586	1.842	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	125.98	0.05656	0.1763	0.2329	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	14.712	2.174	16.886	29.804	103.0	132.8	0.05739	0.1872	0.2446	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	30.526	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	0.1783	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	31.489	128.8	160.3	0.06026	0.2314	0.2917	99.00	0.1834	4.6495	131
132	1.189	14.915	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
135	1.308	14.990	3.132	18.122	32.452	146.4	178.9	0.06189	0.2614	0.3233	103.00	0.1902	5.1676	135
136	1.350	15.016	3.237	18.253	32.692	151.2	183.9	0.06229	0.2695	0.3318	104.00	0.1918	5.3046	136
137	1.393	15.041	3.348	18.389	32.933	156.1	189.0	0.06270	0.2778	0.3405	105.00	0.1935	5.4446	137
138	1.439	15.066	3.462	18.528	33.174	161.2	194.4	0.06310	0.2865	0.3496	106.00	0.1952	5.5878	138
139	1.485	15.091	3.580	18.671	33.414	166.5	199.9	0.06350	0.2954	0.3589	107.00	0.1969	5.7342	139

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

TABLE 1. THERMODYNAMIC PROPERTIES OF MOIST AIR[a] (STANDARD ATMOSPHERIC PRESSURE, 29.921 IN. HG) (Continued)

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/(Lb) (°F) s_w	Vap. Press In. Hg p_s	
140	0.1534	15.117	3.702	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	3.829	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	3.961	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	4.239	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	4.386	19.629	34.859	202.5	237.4	0.06589	0.3559	0.4218	112.99	0.2068	6.6828	145
146	0.1862	15.268	4.539	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	4.698	19.991	35.340	216.4	251.7	0.06669	0.3790	0.4457	114.99	0.2101	7.0273	147
148	0.1989	15.319	4.862	20.181	35.581	223.7	259.3	0.06708	0.3912	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	247.3	283.6	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	5.996	21.466	37.026	273.9	310.9	0.06945	0.4743	0.5438	121.99	0.2216	8.3541	154
155	0.2514	15.496	6.213	21.709	37.267	283.5	320.8	0.06984	0.4901	0.5599	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	0.5768	123.99	0.2248	8.7701	156
157	0.2693	15.546	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	0.7142	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	136.01	0.2441	11.652	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
170	0.4327	15.874	10.938	26.812	40.882	490.6	531.5	0.07565	0.8273	0.9030	138.01	0.2473	12.203	170
171	0.4500	15.900	11.391	27.291	41.123	510.4	551.5	0.07603	0.8592	0.9352	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	0.4875	15.950	12.376	28.326	41.605	553.3	594.9	0.07680	0.9281	1.0049	141.01	0.2521	13.069	173
174	0.5078	15.975	12.911	28.886	41.846	576.5	618.3	0.07718	0.9654	1.0426	142.02	0.2537	13.369	174

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

TABLE 1. 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 v_a	v_{as}	v	ENTHALPY BTU/LB DRY AIR h_a	h_{as}	h	ENTROPY BTU PER (°F) (LB DRY AIR) s_a	s_{as}	s	CONDENSED WATER Enthalpy Btu/Lb h_w	Entropy Btu/(Lb) (°F) s_w	Vap. Press In. Hg p_s	FAHR. TEMP. t (F)
175	0.5292	16.001	13.475	29.476	42.087	601.1	643.2	0.07756	1.005	1.083	143.02	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	15.386	31.462	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	783.0	827.4	0.07984	1.295	1.376	149.03	0.2647	15.636	181
182	0.7218	16.177	18.565	34.742	43.775	821.0	865.7	0.08021	1.357	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	152.04	0.2693	16.702	184
185	0.8363	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	17.446	186
187	0.9283	16.304	24.047	40.351	44.981	1059	1104	0.08208	1.731	1.813	155.05	0.2740	17.828	187
188	0.9802	16.329	25.427	41.756	45.222	1119	1164	0.08245	1.821	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	30.355	46.760	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	51.038	46.428	1513	1559	0.08431	2.444	2.528	161.08	0.2833	20.271	193
194	1.416	16.480	37.036	53.516	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	1.635	16.531	42.885	59.416	47.153	1871	1918	0.08542	3.002	3.087	164.09	0.2879	21.594	196
197	1.767	16.556	46.402	62.958	47.394	2022	2069	0.08579	3.238	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	2.295	16.632	60.510	77.142	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.

given temperature (below the critical temperature), though this requires that it have a definite pressure p_s and that the coexisting condensed phase have the same temperature and pressure. The values of saturation pressure listed in Table 1 have been computed from the formulae of Goff and Gratch [2].

Thermodynamic Properties of Water at Saturation

Table 2 offers revisions to existing steam table data [3] with extension downward to −160 F. These revisions and extension were a necessary preliminary to the construction of Table 1. A detailed explanation of the methods employed in constructing Table 2 is given in a paper [2] by John A. Goff and S. Gratch. As in Table 1 the temperature scale used as argument in Table 2 is the Fahrenheit scale defined in terms of absolute temperature T by Equation 1 whereas the Fahrenheit scale used as argument in existing steam tables is that derived from the International Centigrade scale by means of Equation 2. The symbols used as column headings in Table 2 are the same as those used in the steam tables and have the same meanings; therefore, a detailed explanation seems unnecessary.

DEGREE OF SATURATION

At given values of temperature and pressure the humidity ratio W of moist air can have any value between zero (dry air) and W_s (moist air at saturation). For convenience a parameter μ called alternatively *degree of saturation* or *percent saturation* is introduced through the definition,

$$W = \mu W_s \tag{3}$$

Obviously the degree of saturation μ can have any value from zero (dry air) to unity (moist air at saturation).

To a degree of approximation within the estimated uncertainty of the data in Table 1 at temperatures below about 150 F, the volume v of moist air per pound of dry air at any degree of saturation μ may be computed from the simple relation,

$$v = v_a + \mu v_{as} \tag{4}$$

To obtain comparable accuracy at temperatures above about 150 F it is necessary to add a correction term $\bar{v}$ as follows,

$$\bar{v} = \frac{\mu(1 - \mu)A}{1 + aW_s\,\mu} \tag{4a}$$

where a denotes the ratio of the apparent molecular weight of dry air (28.966) to the molecular weight of water (18.016), namely, 1.6078. In Table 3 are given, for each of several higher temperatures, the corresponding value of the coefficient A, the value of μ at which the correction term $\bar{v}$ attains its maximum value, and the maximum value of the correction term there attained.

At temperatures below about 150 F the enthalpy h of moist air per pound of dry air at any degree of saturation μ may be computed from the simple relation,

$$h = h_a + \mu h_{as} \tag{5}$$

To obtain comparable accuracy at temperatures above about 150 F it is

Table 2. Thermodynamic Properties of Water at Saturation[a]

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^7$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-8}$	Sat. Vapor $v_g \times 10^{-5}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−160	0.4949	1.008	0.01722	36.07	36.07	−222.05	1212.43	990.38	−0.4907	4.0456	3.5549	−160
−159	0.5592	1.138	0.01722	32.03	32.03	−221.73	1212.55	990.82	−0.4896	4.0325	3.5429	−159
−158	0.6312	1.285	0.01723	28.47	28.47	−221.41	1212.67	991.26	−0.4886	4.0196	3.5310	−158
−157	0.7121	1.450	0.01723	25.32	25.32	−221.09	1212.79	991.70	−0.4875	4.0067	3.5192	−157
−156	0.8026	1.634	0.01723	22.54	22.54	−220.76	1212.90	992.14	−0.4864	3.9939	3.5075	−156
−155	0.9040	1.840	0.01723	20.08	20.08	−220.44	1213.02	992.58	−0.4854	3.9812	3.4958	−155
−154	1.017	2.072	0.01723	17.90	17.90	−220.12	1213.15	993.03	−0.4843	3.9685	3.4842	−154
−153	1.144	2.329	0.01723	15.97	15.97	−219.79	1213.26	993.47	−0.4832	3.9559	3.4727	−153
−152	1.286	2.618	0.01723	14.26	14.26	−219.47	1213.38	993.91	−0.4822	3.9435	3.4613	−152
−151	1.444	2.939	0.01723	12.74	12.74	−219.14	1213.49	994.35	−0.4811	3.9311	3.4500	−151
−150	1.620	3.298	0.01723	11.39	11.39	−218.82	1213.62	994.80	−0.4801	3.9188	3.4387	−150
−149	1.816	3.698	0.01723	10.19	10.19	−218.49	1213.73	995.24	−0.4790	3.9065	3.4275	−149
−148	2.035	4.143	0.01724	9.123	9.123	−218.16	1213.84	995.68	−0.4780	3.8944	3.4164	−148
−147	2.278	4.639	0.01724	8.174	8.174	−217.83	1213.95	996.12	−0.4769	3.8823	3.4054	−147
−146	2.549	5.190	0.01724	7.330	7.330	−217.50	1214.06	996.56	−0.4758	3.8702	3.3944	−146
−145	2.850	5.803	0.01724	6.577	6.577	−217.17	1214.17	997.00	−0.4748	3.8583	3.3835	−145
−144	3.184	6.483	0.01724	5.905	5.905	−216.83	1214.28	997.45	−0.4737	3.8464	3.3727	−144
−143	3.556	7.240	0.01724	5.305	5.305	−216.50	1214.39	997.89	−0.4727	3.8346	3.3619	−143
−142	3.967	8.076	0.01724	4.770	4.770	−216.16	1214.49	998.33	−0.4716	3.8229	3.3513	−142
−141	4.423	9.005	0.01724	4.292	4.292	−215.83	1214.60	998.77	−0.4706	3.8112	3.3406	−141
−140	4.928	10.03	0.01724	3.864	3.864	−215.49	1214.70	999.21	−0.4695	3.7996	3.3301	−140
−139	5.487	11.17	0.01724	3.481	3.481	−215.16	1214.82	999.66	−0.4685	3.7881	3.3196	−139
−138	6.106	12.43	0.01724	3.138	3.138	−214.82	1214.92	1000.10	−0.4674	3.7766	3.3092	−138
−137	6.790	13.82	0.01724	2.831	2.831	−214.48	1215.02	1000.54	−0.4664	3.7653	3.2989	−137
−136	7.546	15.36	0.01725	2.555	2.555	−214.14	1215.12	1000.98	−0.4653	3.7540	3.2887	−136
−135	8.380	17.06	0.01725	2.308	2.308	−213.80	1215.22	1001.42	−0.4643	3.7428	3.2785	−135
−134	9.301	18.94	0.01725	2.086	2.086	−213.46	1215.32	1001.86	−0.4632	3.7315	3.2683	−134
−133	10.32	21.01	0.01725	1.886	1.886	−213.11	1215.42	1002.31	−0.4622	3.7205	3.2583	−133
−132	11.44	23.28	0.01725	1.707	1.707	−212.77	1215.52	1002.75	−0.4611	3.7093	3.2482	−132
−131	12.67	25.79	0.01725	1.545	1.545	−212.43	1215.62	1003.19	−0.4601	3.6984	3.2383	−131
−130	14.03	28.56	0.01725	1.400	1.400	−212.08	1215.71	1003.63	−0.4590	3.6874	3.2284	−130
−129	15.52	31.60	0.01725	1.269	1.269	−211.73	1215.80	1004.07	−0.4580	3.6766	3.2186	−129

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

Table 2. Thermodynamic Properties of Water at Saturation[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^6$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (Lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-7}$	Sat. Vapor $v_g \times 10^{-7}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−128	1.716	3.494	0.01726	11.51	11.51	−211.39	1215.91	1004.52	−0.4569	3.6658	3.2089	−128
−127	1.897	3.862	0.01726	10.45	10.45	−211.04	1216.00	1004.96	−0.4559	3.6551	3.1992	−127
−126	2.095	4.265	0.01726	9.489	9.489	−210.69	1216.09	1005.40	−0.4548	3.6444	3.1896	−126
−125	2.312	4.708	0.01726	8.622	8.622	−210.34	1216.18	1005.84	−0.4538	3.6338	3.1800	−125
−124	2.551	5.194	0.01726	7.839	7.839	−209.99	1216.27	1006.28	−0.4527	3.6232	3.1705	−124
−123	2.812	5.726	0.01726	7.131	7.131	−209.64	1216.37	1006.73	−0.4517	3.6127	3.1610	−123
−122	3.099	6.310	0.01726	6.491	6.491	−209.28	1216.45	1007.17	−0.4506	3.6022	3.1516	−122
−121	3.413	6.949	0.01726	5.911	5.911	−208.93	1216.54	1007.61	−0.4496	3.5919	3.1423	−121
−120	3.757	7.649	0.01726	5.386	5.386	−208.58	1216.63	1008.05	−0.4485	3.5815	3.1330	−120
−119	4.133	8.414	0.01726	4.911	4.911	−208.22	1216.71	1008.49	−0.4475	3.5713	3.1238	−119
−118	4.544	9.251	0.01727	4.480	4.480	−207.86	1216.80	1008.94	−0.4464	3.5611	3.1147	−118
−117	4.993	10.17	0.01727	4.088	4.088	−207.51	1216.89	1009.38	−0.4454	3.5510	3.1056	−117
−116	5.484	11.16	0.01727	3.733	3.733	−207.15	1216.97	1009.82	−0.4444	3.5409	3.0965	−116
−115	6.019	12.26	0.01727	3.411	3.411	−206.79	1217.05	1010.26	−0.4433	3.5308	3.0875	−115
−114	6.604	13.44	0.01727	3.118	3.118	−206.43	1217.13	1010.70	−0.4423	3.5209	3.0786	−114
−113	7.241	14.74	0.01727	2.852	2.852	−206.07	1217.21	1011.14	−0.4412	3.5109	3.0697	−113
−112	7.936	16.16	0.01727	2.610	2.610	−205.70	1217.29	1011.59	−0.4402	3.5011	3.0609	−112
−111	8.693	17.70	0.01727	2.389	2.389	−205.34	1217.37	1012.03	−0.4391	3.4912	3.0521	−111
−110	9.517	19.38	0.01728	2.189	2.189	−204.98	1217.45	1012.47	−0.4381	3.4815	3.0434	−110
−109	10.41	21.20	0.01728	2.006	2.006	−204.61	1217.52	1012.91	−0.4370	3.4718	3.0348	−109
−108	11.39	23.19	0.01728	1.839	1.839	−204.24	1217.60	1013.36	−0.4360	3.4621	3.0261	−108
−107	12.45	25.35	0.01728	1.687	1.687	−203.88	1217.68	1013.80	−0.4350	3.4526	3.0176	−107
−106	13.60	27.70	0.01728	1.549	1.549	−203.51	1217.75	1014.24	−0.4339	3.4430	3.0091	−106
−105	14.86	30.25	0.01728	1.422	1.422	−203.14	1217.82	1014.68	−0.4329	3.4335	3.0006	−105
−104	16.22	33.01	0.01728	1.307	1.307	−202.77	1217.89	1015.12	−0.4318	3.4240	2.9922	−104
−103	17.69	36.02	0.01728	1.201	1.201	−202.40	1217.96	1015.56	−0.4308	3.4146	2.9838	−103
−102	19.29	39.28	0.01728	1.104	1.104	−202.03	1218.04	1016.01	−0.4298	3.4053	2.9755	−102
−101	21.03	42.81	0.01729	1.016	1.016	−201.65	1218.10	1016.45	−0.4287	3.3960	2.9673	−101
−100	22.91	46.64	0.01729	0.9352	0.9352	−201.28	1218.17	1016.89	−0.4277	3.3868	2.9591	−100
−99	24.94	50.79	0.01729	0.8613	0.8613	−200.90	1218.23	1017.33	−0.4266	3.3775	2.9509	−99
−98	27.15	55.28	0.01729	0.7936	0.7936	−200.53	1218.30	1017.77	−0.4256	3.3684	2.9428	−98
−97	29.54	60.14	0.01729	0.7314	0.7314	−200.15	1218.37	1018.22	−0.4246	3.3593	2.9347	−97

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

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

FAHR. TEMP. t(F)	ABSOLUTE PRESSURE $p_s \times 10^3$		SPECIFIC VOLUME, CU FT PER LB			ENTHALPY, BTU PER LB			ENTROPY, BTU PER (LB) (°F)			FAHR. TEMP. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-6}$	Sat. Vapor $v_g \times 10^{-6}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−96	3.212	6.539	0.01729	6.745	6.745	−199.78	1218.44	1018.66	−0.4235	3.3502	2.9267	−96
−95	3.491	7.108	0.01729	6.223	6.223	−199.40	1218.50	1019.10	−0.4225	3.3412	2.9187	−95
−94	3.793	7.722	0.01729	5.743	5.743	−199.02	1218.56	1019.54	−0.4214	3.3322	2.9108	−94
−93	4.118	8.385	0.01730	5.303	5.303	−198.64	1218.62	1019.98	−0.4204	3.3233	2.9029	−93
−92	4.470	9.102	0.01730	4.899	4.899	−198.25	1218.68	1020.43	−0.4194	3.3145	2.8951	−92
−91	4.850	9.876	0.01730	4.528	4.528	−197.87	1218.74	1020.87	−0.4183	3.3056	2.8873	−91
−90	5.260	10.71	0.01730	4.186	4.186	−197.49	1218.80	1021.31	−0.4173	3.2969	2.8796	−90
−89	5.702	11.61	0.01730	3.872	3.872	−197.10	1218.85	1021.75	−0.4162	3.2880	2.8718	−89
−88	6.179	12.58	0.01730	3.583	3.583	−196.72	1218.92	1022.20	−0.4152	3.2794	2.8642	−88
−87	6.692	13.63	0.01730	3.317	3.317	−196.33	1218.97	1022.64	−0.4142	3.2708	2.8566	−87
−86	7.245	14.75	0.01730	3.072	3.072	−195.94	1210.02	1023.08	−0.4131	3.2621	2.8490	−86
−85	7.841	15.96	0.01730	2.846	2.846	−195.56	1219.08	1023.52	−0.4121	3.2536	2.8415	−85
−84	8.482	17.27	0.01731	2.638	2.638	−195.16	1219.12	1023.96	−0.4110	3.2450	2.8340	−84
−83	9.171	18.67	0.01731	2.446	2.446	−194.78	1219.18	1024.40	−0.4100	3.2366	2.8266	−83
−82	9.913	20.18	0.01731	2.270	2.270	−194.38	1219.23	1024.85	−0.4090	3.2282	2.8192	−82
−81	10.71	21.81	0.01731	2.106	2.106	−193.99	1219.28	1025.29	−0.4079	3.2197	2.8118	−81
−80	11.57	23.55	0.01731	1.955	1.955	−193.60	1219.33	1025.73	−0.4069	3.2114	2.8045	−80
−79	12.49	25.42	0.01731	1.816	1.816	−193.20	1219.37	1026.17	−0.4058	3.2030	2.7972	−79
−78	13.47	27.43	0.01731	1.687	1.687	−192.81	1219.43	1026.62	−0.4048	3.1948	2.7900	−78
−77	14.53	29.59	0.01732	1.568	1.568	−192.41	1219.47	1027.06	−0.4038	3.1866	2.7828	−77
−76	15.67	31.91	0.01732	1.458	1.458	−192.01	1219.51	1027.50	−0.4028	3.1784	2.7756	−76
−75	16.89	34.39	0.01732	1.356	1.356	−191.62	1219.56	1027.94	−0.4017	3.1702	2.7685	−75
−74	18.20	37.05	0.01732	1.262	1.262	−191.22	1219.60	1028.38	−0.4007	3.1622	2.7615	−74
−73	19.60	39.91	0.01732	1.175	1.175	−190.82	1219.65	1028.83	−0.3996	3.1540	2.7544	−73
−72	21.10	42.96	0.01732	1.094	1.094	−190.41	1219.68	1029.27	−0.3986	3.1460	2.7474	−72
−71	22.71	46.24	0.01732	1.020	1.020	−190.01	1219.72	1029.71	−0.3976	3.1381	2.7405	−71
−70	24.43	49.74	0.01732	0.9501	0.9501	−189.61	1219.76	1030.15	−0.3965	3.1301	2.7336	−70
−69	26.27	53.49	0.01732	0.8858	0.8858	−189.20	1219.80	1030.60	−0.3955	3.1222	2.7267	−69
−68	28.24	57.51	0.01733	0.8261	0.8261	−188.80	1219.84	1031.04	−0.3945	3.1143	2.7198	−68
−67	30.35	61.80	0.01733	0.7707	0.7707	−188.39	1219.87	1031.48	−0.3934	3.1064	2.7130	−67
−66	32.60	66.38	0.01733	0.7193	0.7193	−187.98	1219.90	1031.92	−0.3924	3.0987	2.7063	−66
−65	35.01	71.28	0.01733	0.6715	0.6715	−187.58	1219.94	1032.36	−0.3914	3.0910	2.6996	−65

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

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

FAHR. TEMP. t(F)	ABSOLUTE PRESSURE $p_s \times 10^3$		SPECIFIC VOLUME, CU FT PER LB			ENTHALPY, BTU PER LB			ENTROPY, BTU PER (LB) (°F)			FAHR. TEMP. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-6}$	Sat. Vapor $v_g \times 10^{-6}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−64	0.3758	0.7652	0.01733	6.272	6.272	−187.17	1219.98	1032.81	−0.3903	3.0832	2.6929	−64
−63	0.4033	0.8211	0.01733	5.859	5.859	−186.76	1220.01	1033.25	−0.3893	3.0755	2.6862	−63
−62	0.4326	0.8808	0.01734	5.476	5.476	−186.35	1220.04	1033.69	−0.3882	3.0678	2.6796	−62
−61	0.4639	0.9444	0.01734	5.120	5.120	−185.93	1220.06	1034.13	−0.3872	3.0602	2.6730	−61
−60	0.4972	1.012	0.01734	4.788	4.788	−185.52	1220.10	1034.58	−0.3862	3.0526	2.6664	−60
−59	0.5328	1.085	0.01734	4.479	4.479	−185.11	1220.13	1035.02	−0.3851	3.0450	2.6599	−59
−58	0.5708	1.162	0.01734	4.192	4.192	−184.69	1220.15	1035.46	−0.3841	3.0376	2.6535	−58
−57	0.6112	1.244	0.01734	3.925	3.925	−184.28	1220.18	1035.90	−0.3831	3.0301	2.6470	−57
−56	0.6543	1.332	0.01734	3.675	3.675	−183.86	1220.20	1036.34	−0.3820	3.0226	2.6406	−56
−55	0.7001	1.426	0.01734	3.443	3.443	−183.44	1220.23	1036.79	−0.3810	3.0152	2.6342	−55
−54	0.7489	1.525	0.01735	3.226	3.226	−183.02	1220.25	1037.23	−0.3800	3.0079	2.6279	−54
−53	0.8009	1.631	0.01735	3.024	3.024	−182.60	1220.27	1037.67	−0.3789	3.0005	2.6216	−53
−52	0.8562	1.743	0.01735	2.836	2.836	−182.18	1220.29	1038.11	−0.3779	2.9932	2.6153	−52
−51	0.9151	1.863	0.01735	2.660	2.660	−181.76	1220.31	1038.55	−0.3769	2.9860	2.6091	−51
−50	0.9776	1.990	0.01735	2.496	2.496	−181.34	1220.34	1039.00	−0.3758	2.9786	2.6028	−50
−49	1.044	2.126	0.01735	2.343	2.343	−180.92	1220.36	1039.44	−0.3748	2.9715	2.5967	−49
−48	1.115	2.270	0.01736	2.200	2.200	−180.49	1220.37	1039.88	−0.3738	2.9643	2.5905	−48
−47	1.190	2.422	0.01736	2.066	2.066	−180.06	1220.38	1040.32	−0.3727	2.9571	2.5844	−47
−46	1.270	2.585	0.01736	1.941	1.941	−179.64	1220.40	1040.76	−0.3717	2.9501	2.5784	−46
−45	1.354	2.757	0.01736	1.824	1.824	−179.21	1220.42	1041.21	−0.3707	2.9430	2.5723	−45
−44	1.444	2.940	0.01736	1.715	1.715	−178.78	1220.43	1041.65	−0.3696	2.9359	2.5663	−44
−43	1.539	3.134	0.01736	1.612	1.612	−178.35	1220.44	1042.09	−0.3686	2.9289	2.5603	−43
−42	1.641	3.340	0.01736	1.516	1.516	−177.92	1220.45	1042.53	−0.3676	2.9219	2.5543	−42
−41	1.748	3.559	0.01736	1.427	1.427	−177.49	1220.47	1042.98	−0.3665	2.9149	2.5484	−41
−40	1.861	3.790	0.01737	1.343	1.343	−177.06	1220.48	1043.42	−0.3655	2.9080	2.5425	−40
−39	1.982	4.035	0.01737	1.264	1.264	−176.63	1220.49	1043.86	−0.3645	2.9012	2.5367	−39
−38	2.110	4.295	0.01737	1.191	1.191	−176.19	1220.49	1044.30	−0.3634	2.8942	2.5308	−38
−37	2.245	4.570	0.01737	1.122	1.122	−175.76	1220.50	1044.74	−0.3624	2.8874	2.5250	−37
−36	2.388	4.862	0.01737	1.057	1.057	−175.32	1220.51	1045.19	−0.3614	2.8807	2.5193	−36
−35	2.540	5.170	0.01737	0.9961	0.9961	−174.88	1220.51	1045.63	−0.3604	2.8739	2.5135	−35
−34	2.700	5.497	0.01737	0.9391	0.9391	−174.45	1220.52	1046.07	−0.3593	2.8671	2.5078	−34
−33	2.870	5.843	0.01738	0.8857	0.8857	−174.01	1220.52	1046.51	−0.3583	2.8604	2.5021	−33

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

Table 2. Thermodynamic Properties of Water at Saturation[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure $p_s \times 10^3$		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-4}$	Sat. Vapor $v_g \times 10^{-4}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
−32	0.3049	0.6208	0.01738	8.355	8.355	−173.57	1220.52	1046.95	−0.3573	2.8538	2.4965	−32
−31	0.3239	0.6595	0.01738	7.883	7.883	−173.13	1220.53	1047.40	−0.3562	2.8470	2.4908	−31
−30	0.3440	0.7003	0.01738	7.441	7.441	−172.68	1220.52	1047.84	−0.3552	2.8405	2.4853	−30
−29	0.3652	0.7435	0.01738	7.025	7.025	−172.24	1220.52	1048.28	−0.3542	2.8339	2.4797	−29
−28	0.3876	0.7891	0.01738	6.634	6.634	−171.80	1220.52	1048.72	−0.3532	2.8274	2.4742	−28
−27	0.4113	0.8373	0.01738	6.267	6.267	−171.35	1220.51	1049.16	−0.3521	2.8207	2.4686	−27
−26	0.4363	0.8882	0.01738	5.921	5.921	−170.91	1220.51	1049.60	−0.3511	2.8143	2.4632	−26
−25	0.4627	0.9420	0.01739	5.596	5.596	−170.46	1220.51	1050.05	−0.3501	2.8078	2.4577	−25
−24	0.4905	0.9987	0.01739	5.290	5.290	−170.01	1220.50	1050.49	−0.3490	2.8013	2.4523	−24
−23	0.5199	1.059	0.01739	5.003	5.003	−169.56	1220.49	1050.93	−0.3480	2.7949	2.4469	−23
−22	0.5509	1.122	0.01739	4.732	4.732	−169.12	1220.49	1051.37	−0.3470	2.7885	2.4415	−22
−21	0.5836	1.188	0.01739	4.477	4.477	−168.66	1220.48	1051.82	−0.3460	2.7822	2.4362	−21
−20	0.6181	1.259	0.01739	4.237	4.237	−168.21	1220.47	1052.26	−0.3449	2.7757	2.4308	−20
−19	0.6545	1.333	0.01739	4.011	4.011	−167.76	1220.46	1052.70	−0.3439	2.7695	2.4256	−19
−18	0.6928	1.410	0.01740	3.797	3.797	−167.31	1220.45	1053.14	−0.3429	2.7632	2.4203	−18
−17	0.7332	1.493	0.01740	3.596	3.596	−166.85	1220.43	1053.58	−0.3418	2.7568	2.4150	−17
−16	0.7757	1.579	0.01740	3.407	3.407	−166.40	1220.42	1054.02	−0.3408	2.7506	2.4098	−16
−15	0.8204	1.670	0.01740	3.228	3.228	−165.94	1220.41	1054.47	−0.3398	2.7444	2.4046	−15
−14	0.8676	1.766	0.01740	3.060	3.060	−165.48	1220.39	1054.91	−0.3388	2.7383	2.3995	−14
−13	0.9172	1.867	0.01740	2.901	2.901	−165.03	1220.38	1055.35	−0.3377	2.7320	2.3943	−13
−12	0.9694	1.974	0.01740	2.750	2.750	−164.57	1220.36	1055.79	−0.3367	2.7259	2.3892	−12
−11	1.024	2.086	0.01740	2.609	2.609	−164.11	1220.34	1056.23	−0.3357	2.7198	2.3841	−11
−10	1.082	2.203	0.01741	2.475	2.475	−163.65	1220.32	1056.67	−0.3347	2.7138	2.3791	−10
−9	1.143	2.327	0.01741	2.349	2.349	−163.18	1220.30	1057.12	−0.3336	2.7076	2.3740	−9
−8	1.207	2.457	0.01741	2.229	2.229	−162.72	1220.28	1057.56	−0.3326	2.7016	2.3690	−8
−7	1.274	2.594	0.01741	2.116	2.116	−162.26	1220.26	1058.00	−0.3316	2.6956	2.3640	−7
−6	1.344	2.737	0.01741	2.010	2.010	−161.79	1220.23	1058.44	−0.3306	2.6896	2.3590	−6
−5	1.419	2.888	0.01741	1.909	1.909	−161.33	1220.21	1058.88	−0.3295	2.6836	2.3541	−5
−4	1.496	3.047	0.01742	1.814	1.814	−160.86	1220.18	1059.32	−0.3285	2.6777	2.3492	−4
−3	1.578	3.213	0.01742	1.723	1.723	−160.39	1220.15	1059.76	−0.3275	2.6718	2.3443	−3
−2	1.664	3.388	0.01742	1.638	1.638	−159.92	1220.13	1060.21	−0.3264	2.6658	2.3394	−2
−1	1.754	3.572	0.01742	1.557	1.557	−159.45	1220.10	1060.65	−0.3254	2.6600	2.3346	−1

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

Table 2. Thermodynamic Properties of Water at Saturation[a] (Continued)

Fahr. Temp. t(F)	Absolute Pressure p_s		Specific Volume, cu ft per lb			Enthalpy, Btu per lb			Entropy, Btu per (lb) (°F)			Fahr. Temp. t(F)
	Lb/Sq In.	In. Hg	Sat. Solid v_i	Evap. $v_{ig} \times 10^{-3}$	Sat. Vapor $v_g \times 10^{-3}$	Sat. Solid h_i	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid s_i	Evap. s_{ig}	Sat. Vapor s_g	
0	0.01849	0.03764	0.01742	14.81	14.81	−158.98	1220.07	1061.09	−0.3244	2.6541	2.3297	0
1	0.01948	0.03966	0.01742	14.08	14.08	−158.51	1220.04	1061.53	−0.3234	2.6483	2.3249	1
2	0.02052	0.04178	0.01742	13.40	13.40	−158.04	1220.01	1061.97	−0.3224	2.6425	2.3201	2
3	0.02616	0.04400	0.01743	12.75	12.75	−157.56	1219.97	1062.41	−0.3213	2.6367	2.3154	3
4	0.02276	0.04633	0.01743	12.14	12.14	−157.09	1219.94	1062.85	−0.3203	2.6309	2.3106	4
5	0.02396	0.04878	0.01743	11.55	11.55	−156.61	1219.90	1063.29	−0.3193	2.6252	2.3059	5
6	0.02521	0.05134	0.01743	11.00	11.00	−156.14	1219.88	1063.74	−0.3182	2.6194	2.3012	6
7	0.02653	0.05402	0.01743	10.48	10.48	−155.66	1219.84	1064.18	−0.3172	2.6138	2.2966	7
8	0.02791	0.05683	0.01743	9.979	9.979	−155.18	1219.80	1064.62	−0.3162	2.6081	2.2919	8
9	0.02936	0.05977	0.01744	9.507	9.507	−154.70	1219.76	1065.06	−0.3152	2.6025	2.2873	9
10	0.03087	0.06286	0.01744	9.060	9.060	−154.22	1219.72	1065.50	−0.3142	2.5969	2.2827	10
11	0.03246	0.06608	0.01744	8.636	8.636	−153.74	1219.68	1065.94	−0.3131	2.5912	2.2781	11
12	0.03412	0.06946	0.01744	8.234	8.234	−153.26	1219.64	1066.38	−0.3121	2.5857	2.2736	12
13	0.03585	0.07300	0.01744	7.851	7.851	−152.77	1219.59	1066.82	−0.3111	2.5801	2.2690	13
14	0.03767	0.07669	0.01744	7.489	7.489	−152.29	1219.55	1067.26	−0.3101	2.5746	2.2645	14
15	0.03957	0.08056	0.01744	7.144	7.144	−151.80	1219.50	1067.70	−0.3090	2.5690	2.2600	15
16	0.04156	0.08461	0.01745	6.817	6.817	−151.32	1219.46	1068.14	−0.3080	2.5635	2.2555	16
17	0.04363	0.08884	0.01745	6.505	6.505	−150.83	1219.41	1068.58	−0.3070	2.5581	2.2511	17
18	0.04581	0.09326	0.01745	6.210	6.210	−150.34	1219.36	1069.02	−0.3060	2.5526	2.2466	18
19	0.04808	0.09789	0.01745	5.929	5.929	−149.85	1219.31	1069.46	−0.3049	2.5471	2.2422	19
20	0.05045	0.1027	0.01745	5.662	5.662	−149.36	1219.26	1069.90	−0.3039	2.5417	2.2378	20
21	0.05293	0.1078	0.01745	5.408	5.408	−148.87	1219.21	1070.34	−0.3029	2.5364	2.2335	21
22	0.05552	0.1130	0.01746	5.166	5.166	−148.38	1219.16	1070.78	−0.3019	2.5310	2.2291	22
23	0.05823	0.1186	0.01746	4.936	4.936	−147.88	1219.10	1071.22	−0.3008	2.5256	2.2248	23
24	0.06105	0.1243	0.01746	4.717	4.717	−147.39	1219.05	1071.66	−0.2998	2.5203	2.2205	24
25	0.06400	0.1303	0.01746	4.509	4.509	−146.89	1218.98	1072.09	−0.2988	2.5150	2.2162	25
26	0.06708	0.1366	0.01746	4.311	4.311	−146.40	1218.93	1072.53	−0.2978	2.5097	2.2119	26
27	0.07030	0.1431	0.01746	4.122	4.122	−145.90	1218.87	1072.97	−0.2968	2.5045	2.2077	27
28	0.07365	0.1500	0.01746	3.943	3.943	−145.40	1218.81	1073.41	−0.2957	2.4991	2.2034	28
29	0.07715	0.1571	0.01747	3.771	3.771	−144.90	1218.75	1073.85	−0.2947	2.4939	2.1992	29
30	0.08080	0.1645	0.01747	3.608	3.608	−144.40	1218.69	1074.29	−0.2937	2.4887	2.1950	30
31	0.08461	0.1723	0.01747	3.453	3.453	−143.90	1218.63	1074.73	−0.2927	2.4835	2.1908	31
32	0.08858	0.1803	0.01747	3.305	3.305	−143.40	1218.56	1075.16	−0.2916	2.4783	2.1867	32

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

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

FAHR. TEMP. t(F)	ABSOLUTE PRESSURE p_s		SPECIFIC VOLUME, CU FT PER LB			ENTHALPY, BTU PER LB			ENTROPY, BTU PER (LB) (°F)			FAHR. TEMP. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	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	
*32	0.088586	0.18036	0.01602	3304.6	3304.6	0.00	1075.16	1075.16	0.00000	2.1867	2.1867	32*
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	0.00409	2.1755	2.1796	34
35	0.099908	0.20342	0.01602	2947.8	2947.8	3.02	1073.46	1076.48	0.00612	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	0.01602	2633.8	2633.8	6.03	1071.77	1077.80	0.01220	2.1535	2.1657	38
39	0.11699	0.23819	0.01602	2537.6	2537.6	7.04	1071.20	1078.24	0.01422	2.1480	2.1622	39
40	0.12164	0.24767	0.01602	2445.4	2445.4	8.04	1070.64	1078.68	0.01623	2.1426	2.1588	40
41	0.12646	0.25748	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	2272.0	2272.0	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	2112.3	2112.3	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	0.42003	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	1087.42	0.05553	2.0385	2.0940	60
61	0.26545	0.54047	0.01604	1167.2	1167.2	29.08	1058.78	1087.86	0.05746	2.0334	2.0909	61

[a]Compiled by John A. Goff and S. Gratch.
*Extrapolated to represent metastable equilibrium with undercooled liquid.

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

FAHR. TEMP. t(F)	ABSOLUTE PRESSURE p_s		SPECIFIC VOLUME, CU FT PER LB			ENTHALPY, BTU PER LB			ENTROPY, BTU PER (LB) (°F)			FAHR. TEMP. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	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	
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	0.60073	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	0.71432	0.01605	896.47	896.49	37.07	1054.27	1091.34	0.07269	1.9941	2.0668	69
70	0.36304	0.73916	0.01605	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	0.07646	1.9845	2.0610	71
72	0.38856	0.79113	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	788.38	788.40	41.07	1052.01	1093.08	0.08022	1.9749	2.0551	73
74	0.41564	0.84626	0.01606	763.73	763.75	42.06	1051.46	1093.52	0.08209	1.9701	2.0522	74
75	0.42979	0.87506	0.01606	739.95	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	673.54	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	1048.62	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	0.54112	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	0.57722	1.1752	0.01608	560.04	560.06	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	454.26	454.28	59.03	1041.84	1100.87	0.11339	1.8919	2.0053	91

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

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

FAHR. TEMP. t(F)	ABSOLUTE PRESSURE p_s		SPECIFIC VOLUME, CU FT PER LB			ENTHALPY, BTU PER LB			ENTROPY, BTU PER (LB) (°F)			FAHR. TEMP. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	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	
92	0.74340	1.5136	0.01611	441.10	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	0.01612	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	0.01612	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	1036.72	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	330.96	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	312.93	312.95	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
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	2.5226	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	265.24	265.26	78.00	1031.01	1109.01	0.14724	1.8098	1.9570	110
111	1.3128	2.6728	0.01617	258.14	258.16	79.00	1030.44	1109.44	0.14899	1.8056	1.9546	111
112	1.3510	2.7507	0.01617	251.25	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	1028.72	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	203.16	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

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

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

FAHR. TEMP. t(F)	ABSOLUTE PRESSURE p_s		SPECIFIC VOLUME, CU FT PER LB			ENTHALPY, BTU PER LB			ENTROPY, BTU PER (LB) (°F)			FAHR. TEMP. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	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	
122	1.7897	3.6439	0.01621	192.85	192.87	89.98	1024.12	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	153.36	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	1.6942	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	0.20182	1.6828	1.8846	142
143	3.1204	6.3532	0.01631	114.40	114.42	110.96	1011.94	1122.90	0.20348	1.6790	1.8825	143
144	3.2006	6.5164	0.01631	111.72	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	6.8536	0.01632	106.54	106.56	113.96	1010.18	1124.14	0.20845	1.6678	1.8763	146
147	3.4517	7.0277	0.01633	104.06	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	7.3872	0.01634	99.306	99.322	116.96	1008.42	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	94.799	94.815	118.96	1007.24	1126.20	0.21667	1.6493	1.8660	151

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

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Continued)

FAHR. TEMP. t(F)	ABSOLUTE PRESSURE p_s		SPECIFIC VOLUME, CU FT PER LB			ENTHALPY, BTU PER LB			ENTROPY, BTU PER (LB) (°F)			FAHR. TEMP. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	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	
152	3.9074	7.9556	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	4.2046	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	84.536	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	9.2042	0.01638	80.798	80.814	125.97	1003.11	1129.08	0.22807	1.6239	1.8520	158
159	4.6304	9.4276	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	75.546	75.562	128.97	1001.33	1130.30	0.23292	1.6133	1.8462	161
162	4.9732	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	63.382	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	6.2746	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	54.602	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
178	7.1849	14.629	0.01650	52.341	52.357	146.01	991.14	1137.15	0.25998	1.5543	1.8143	178
179	7.3469	14.959	0.01650	51.260	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	0.26312	1.5475	1.8106	180
181	7.6801	15.637	0.01651	49.173	49.190	149.02	989.32	1138.34	0.26468	1.5442	1.8089	181

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

TABLE 2. THERMODYNAMIC PROPERTIES OF WATER AT SATURATION[a] (Concluded)

FAHR. TEMP. t(F)	ABSOLUTE PRESSURE p_s		SPECIFIC VOLUME, CU FT PER LB			ENTHALPY, BTU PER LB			ENTROPY, BTU PER (LB) (°F)			FAHR. TEMP. t(F)
	Lb/Sq In.	In. Hg	Sat. Liquid v_f	Evap. v_{fg}	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	
182	7.8514	15.986	0.01652	48.168	48.185	150.02	988.72	1138.74	0.26625	1.5408	1.8071	182
183	8.0258	16.341	0.01652	47.187	47.204	151.02	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	1139.92	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	43.489	43.506	155.04	985.67	1140.71	0.27404	1.5242	1.7982	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	40.128	40.145	159.06	983.22	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	9.9563	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	37.069	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	1144.62	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	33.623	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	1.4789	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	26.532	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	29.413	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	28.326	28.343	177.16	972.06	1149.22	0.30765	1.4536	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.

necessary to add a correction term $\bar{h}$ as follows,

$$\bar{h} = \frac{\mu(1 - \mu)B}{1 + aW_s\,\mu} \tag{5a}$$

In Table 3 are listed values of the coefficient B and maximum values of the correction term $\bar{h}$, the latter occurring at the same values of μ as do those of the correction term $\bar{v}$.

Unfortunately, values of the entropy s of moist air per pound of dry air computed from the simple relation,

$$s = s_a + \mu s_{as} \tag{6}$$

do not approximate the true values as closely as might be desired except at temperatures considerably lower than 150 F. Only a relatively small portion of the error is contributed by the correction term

$$\bar{s} = \frac{\mu(1 - \mu)C}{1 + aW_s\,\mu} \tag{6a}$$

Table 3 lists values of the coefficient C and maximum values of the correction term $\bar{s}$, the latter occurring at the same values of μ as do those of the correction terms $\bar{v}$ and $\bar{h}$. The larger portion of the error is contributed by the so-called *mixing entropy*. It can be expressed as an additional correction term $\bar{s}$ as follows,

$$\bar{s} = 0.1579\,[(1 + \mu aW_s)\log_{10}(1 + \mu aW_s) - \mu aW_s \log_{10}(\mu) - \mu(1 + aW_s)\log_{10}(1 + aW_s)]. \tag{6b}$$

Maximum values of $\bar{s}$ and the values of μ at which they occur are given in Table 3. In Equation 6b $\log_{10}$ denotes logarithm to the base 10.

MOLLIER DIAGRAM

It is a fundamental proposition of thermodynamics that *when a fluid flows across a section fixed in space it convects with it an amount of energy equal to its enthalpy as determined by the pressure, temperature, and composition of the fluid at that section.* This fundamental proposition provides the correct procedure for applying the law of conservation of energy to the processes occurring most frequently in air conditioning practice. Thus if moist air is flowing through a duct it carries across any section of the duct energy of amount mh Btu per minute and water of amount mW pounds per minute, if m denotes the weight of dry air crossing the section per minute.

The above considerations suggest the importance of having accurate knowledge regarding the enthalpy of the fluid in question and the desirability of using enthalpy as one of the coordinates in graphical representation. The use of enthalpy h and humidity ratio W as coordinates in the case of moist air is due to Mollier [4]. A convenient modification of the Mollier diagram introduced by Goff is obtained by taking humidity ratio W as ordinate and reduced enthalpy (h-1000W) as abscissa. A Mollier Diagram modified in this way is enclosed in the envelope attached to the inside back cover and an abridgement of the Diagram is shown in Fig. 1.

The Mollier Diagram is a constant-pressure chart, the one provided with this book being drawn for standard atmospheric pressure from the data in Table 1. Along the axis of abscissae ($W = 0$, $\mu = 0$) are plotted

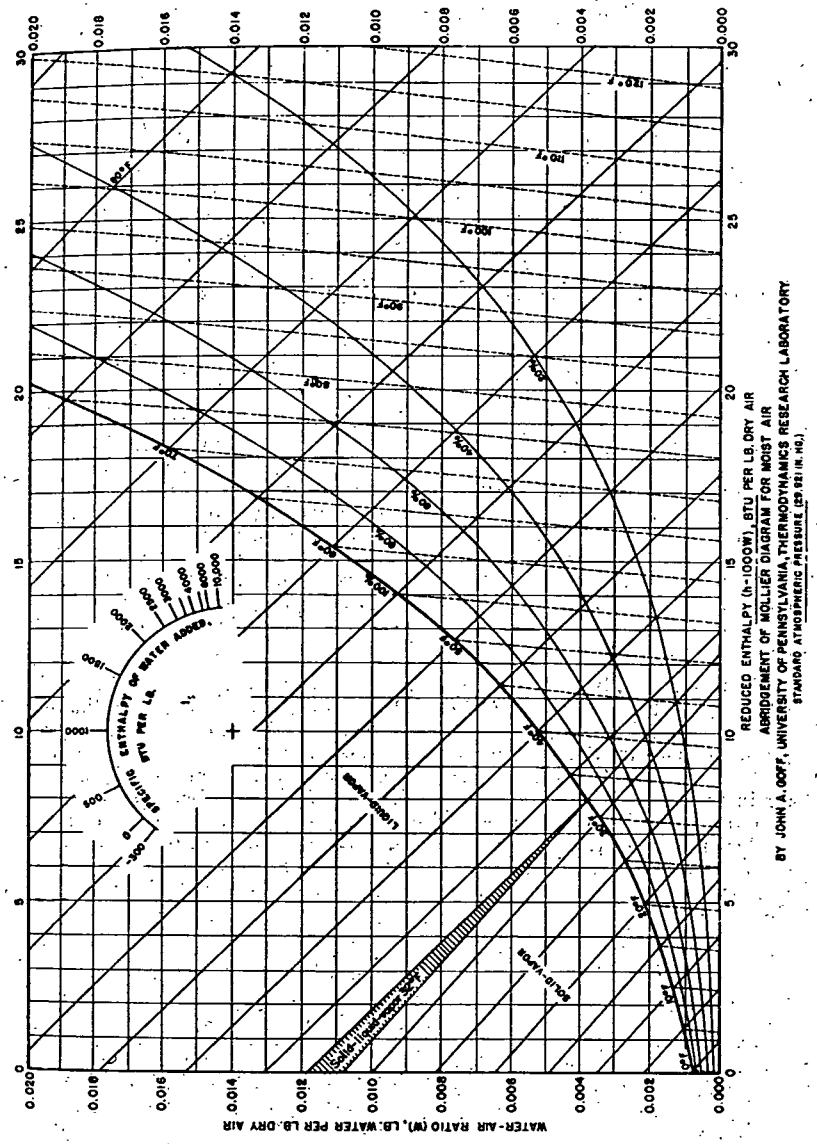


FIG. 1. ABRIDGEMENT OF MOLLIER DIAGRAM FOR MOIST AIR

TABLE 3. COEFFICIENTS A, B, C APPEARING IN EQUATIONS 4a, 5a, 6a, MAXIMUM VALUES OF CORRECTIONS DEFINED BY EQUATIONS 4a, 5a, 6a. DEGREE OF SATURATION AT WHICH THESE MAXIMA OCCUR, $\bar{\mu}_m$. MAXIMUM VALUE OF CORRECTION DEFINED BY EQUATION 6b. DEGREE OF SATURATION AT WHICH THIS MAXIMUM OCCURS, $\bar{\bar{\mu}}_m$.

(*Standard Atmospheric Pressure*)

t (F)	A (ft^3/lb_a)	B (Btu/lb_a)	C (Btu/F/lb_a)	$\bar{v}_{max}$ (ft^3/lb_a)	$\bar{h}_{max}$ (Btu/lb_a)	$\bar{s}_{max}$ (Btu/F/lb_a)	$\bar{\mu}_m$	$\bar{\bar{s}}_{max}$ (Btu/F/lb_a)	$\bar{\bar{\mu}}_m$
96	0.0018	0.0286	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

values of the specific enthalpy of dry air h_a at one-degree intervals of temperature. Values of humidity ratio at saturation W_s plotted against values of reduced enthalpy at saturation (h_s-1000W_s) determine the saturation curve (μ = 100 per cent). Lines of constant temperature connect points on the saturation curve with corresponding points on the dry-air axis and are inclined upward to the right. They are drawn straight in accordance with Equations 3 and 5 because the curvature contributed by the correction term 5a is inappreciable at all temperatures within the range of the chart. The portion of each isotherm lying between the dry-air axis and the saturation curve is divided into 10 equal parts by curves of constant per cent saturation. The per cent saturation of any point below the saturation curve is readily determined by linear interpolation along the isotherm through that point.

Each isotherm breaks at the saturation curve to incline upward to the left into the two-phase region above the saturation curve. The ordinate of a point in this region is the total weight of water in both the vapor phase (moist air) and the condensed phase (liquid or solid) per pound of dry air in both phases. Neglecting the very small amount of dissolved air in the condensed phase, it is the weight of water in both phases per pound of dry air in the vapor phase. The ordinate at the break in the isotherm through the point in question 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 phase per pound of dry air in the vapor phase.

It has been stated that the region above the saturation curve is the two-phase region. This is so except in the wedge with apex on the saturation curve at 32 F where three distinct phases, namely, solid, liquid, and vapor coexist. In fact, this wedge separates the liquid-vapor region above the wedge from the solid-vapor region below it. A point inside the wedge divides the horizontal line extending through it from one boundary to the other into two segments which are in the same ratio as are the weights of solid and liquid. The temperature is 32 F throughout the wedge.

The isotherms in the two-phase regions above the saturation curve have been extended downward to the right into the vapor-phase region

below the saturation curve as lines of constant *thermodynamic wet-bulb temperature*. The definition of thermodynamic wet-bulb temperature will be given later.

On the Mollier Diagram provided with THE 1946 GUIDE there has been drawn a *protractor* from which can be determined the direction in which the state point of a mixture of water and dry air will be moved *by simultaneous addition of energy and water without addition of dry air*. A particular direction is specified by the numerical value of the *ratio of energy to water added* which ratio is designated as q and called the *specific enthalpy of water added, Btu per pound*. The protractor is especially useful in locating the *condition line* of a cooling load or heating load problem.

DERIVED PROPERTIES

Thermodynamic Wet-bulb Temperature. For any state of moist air there exists a temperature t^* at which liquid (or solid) water may be evaporated into the air to bring it to saturation at exactly this same temperature. The humidity ratio of the air is increased from a given initial value W to the value W_s^* corresponding to saturation at the temperature t^*; the enthalpy of the air is increased from a given initial value h to the value h_s^* corresponding to saturation at the temperature t^*; the weight of water added per pound of dry air is $W_s^* - W$ and this adds energy of amount $(W_s^* - W)\, h_w^*$, where h_w^* denotes the specific enthalpy of the water as added at the temperature t^*; therefore, if the process is strictly adiabatic,

$$h + (W_s^* - W)h_w^* = h_s^* \qquad (7)$$

The solution of Equation 7 for given values of h and W is called *thermodynamic wet-bulb temperature.*

Example 1. Find the thermodynamic wet-bulb temperature of moist air at 80 F, 50 per cent saturation, atmospheric pressure.

Solution. The answer is obtained directly from the Mollier Diagram. From the data of Table 1, the enthalpy of the air is $h = 19.221 + 0.50 \times 24.47 = 31.46$ Btu/lb_a (Equation 5). To a first approximation this is the enthalpy at saturation at the thermodynamic wet-bulb temperature which is therefore approximately 67 F.

At 67 F the humidity ratio at saturation is 0.01424 lb_w/lb_a and the specific enthalpy of liquid water is 35.11 Btu/lb_w. The humidity ratio of the air is $W = 0.50 \times 0.02233 = 0.01117$ lb_w/lb_a (Equation 3). Therefore, to a second approximation, the enthalpy at saturation at the thermodynamic wet-bulb temperature is $h_s^* = 31.46 + (0.01424 - 0.01117) \times 35.11 = 31.57$ Btu/lb_a, Equation 7. Interpolation in Table 1 gives as final answer,

$$t^* = 66.94 \text{ F}$$

The psychrometer is an instrument consisting of two thermometers one of which has the bulb covered with a suitable wick that has been dipped in liquid water and thoroughly wetted by it. On placing the wet-bulb of the instrument in an air stream, the liquid begins to evaporate from the wick and it is usually assumed that such evaporation brings the air immediately adjacent to the wick to saturation. At first this air may reach saturation at a higher or lower temperature than that of the liquid on the wick; but in a relatively short time the temperature of the liquid will have changed to approach equality with that of the air touching the wick, even if this requires the liquid to freeze on the wick. Then the liquid (or solid) will continue for a time to evaporate into the air stream at such temperature as will bring a portion of the air stream to saturation

at this same temperature. This equilibrium temperature is called *wet-bulb temperature.*

It is clear that the readings of an actual wet-bulb thermometer cannot be regarded as values of a thermodynamic property of moist air; for these readings are importantly affected by a number of non-thermodynamic factors including design, construction, installation, and technique of using the instrument. Thus, unless the wet-bulb is effectively shielded against radiation from relatively warm surfaces the process will not be strictly adiabatic as tacitly assumed in writing Equation 7. Also, partial drying of the wick will prevent the air immediately adjacent to it from reaching complete saturation as assumed in Equation 7. A working theory developed by Arnold [5] enables the calculation of corrections to be applied to the readings of the actual instrument in order to make them agree with the values of temperature calculated from Equation 7. Fortunately, and indeed fortuitously, these corrections can be made small, but to emphasize the necessity of making them in accurate experimentation, the temperature defined by Equation 7 is called *thermodynamic wet-bulb temperature.*

Example 2. Find the degree of saturation of moist air at 90 F dry-bulb, 63 F thermodynamic wet-bulb, atmospheric pressure.

Solution. The answer is read directly from the Mollier Diagram. Inserting numerical data from Table 1 into Equation 7 gives

$$(21.625 + 34.31\mu) + (0.01235 - 0.03118\mu) \times 31.12 = 28.57$$

The solution of this equation is direct and the final answer is

$$\mu = 19.67 \text{ per cent}$$

Example 3. Find the temperature to which moist air initially saturated at 40 F and at standard atmospheric pressure must be heated in order to have a thermodynamic wet-bulb temperature of 60 F.

Solution. On the Mollier Diagram follow a horizontal line from the saturation curve at 40 F to its intersection with the 60 F thermodynamic wet-bulb line and read the corresponding temperature directly.

Inserting numerical data from Table 1 into Equation 7, this becomes

$$h_a + 0.005213h_{as}/W_s = 26.46 - (0.01108 - 0.005213) \times 28.12 = 26.295$$

At 85 F the lefthand member of this equation has the value 26.147; at 86 F its value is 26.389; by linear interpolation the answer is

$$t = 85.61 \text{ F}$$

Dew-Point Temperature. Corresponding to any given state of moist air there exists another state on the saturation curve having the same humidity ratio W and same pressure p as the given state. The temperature at this other state on the saturation curve is called the *dew-point temperature* of the given state. Obviously, if moist air is cooled at constant pressure and constant humidity ratio it will reach saturation when its temperature falls to a value equal to its dew-point temperature. This will usually be marked by the first appearance of a coexisting condensed phase. In one type of dew-point apparatus a continuous sample of air is passed over a mirror which can be cooled by external refrigeration and whose temperature can be accurately measured. The measured temperature at which the intensity of light reflected from the mirror is abruptly diminished by condensation is taken to be the dew-point temperature of the air sample. Examples 4 and 5 illustrate the relation between the dew-point, degree of saturation and dry-bulb temperature.

Example 4. Find the dew-point temperature of moist air at 80 F, 50 per cent saturation, atmospheric pressure.

Solution. On the Mollier Diagram follow a horizontal line from a given state point (80 F, 50 per cent) to the saturation curve and read the temperature at the intersection.

From the data in Table 1, the humidity ratio of the air is $W = 0.50 \times 0.02233 = 0.01117$ lb_w/lb_a. By interpolation this is found to be the humidity ratio at saturation at 60.22 F which is therefore the required answer.

Example 5. Find the degree of saturation of moist air at 90 F dry-bulb, 40 F (dew-point), atmospheric pressure.

Solution. On the Mollier Diagram follow a horizontal line from 40 F on the saturation curve to the 90 F isotherm (dry-bulb) and read the degree of saturation directly.

From the data in Table 1, the humidity ratio of the air must be $W = 0.005213$. But the humidity ratio at saturation at 90 F is 0.03118; hence the degree of saturation is

$$\mu = 0.005213/0.03118 = 16.72 \text{ per cent}$$

TYPICAL AIR CONDITIONING PROCESSES

The use of Table 1 and the Mollier Diagram in analyzing typical air conditioning processes is best explained by means of illustrative examples. In each of the following, it is to be understood that the process in question takes place at a constant pressure of 29.921 in. Hg, or standard atmospheric pressure.

Heating

The process of adding heat to moist air is represented by a horizontal line on the Mollier Diagram. The length of the line between the initial and final state points is the increase of reduced enthalpy; but, since the humidity ratio is constant, it is also the increase of enthalpy itself and therefore the quantity of heat added per pound of dry air.

Example 6. Air initially at 20 F, 80 per cent saturation is heated to 120 F. Find *the quantity of heat required to process 20,000 cfm of heated air.*

Solution. From the data in Table 1: the initial humidity ratio is $0.80 \times 0.002152 = 0.001722$ lb_w/lb_a; the initial enthalpy is $4.804 + 0.80 \times 2.302 = 6.646$ Btu/lb_a; the final degree of saturation is $0.001722/0.08149 = 2.113$ per cent; the final enthalpy is $28.841 + 0.02113 \times 90.70 = 30.757$ Btu/lb_a.

It may be supposed that the air is heated between two sections of a duct. The quantities of energy convected across the two sections per pound of dry air crossing them are the two enthalpies calculated. Conservation of energy requires that the difference between these two enthalpies be the quantity of heat added; thus,

$$_AQ_B = 30.757 - 6.646 = 24.111 \text{ Btu/lb}_a.$$

The final volume is $14.611 + 0.02113 \times 1.905 = 14.651$ cu ft/lb_a. Since 20,000 cfm of heated air is to be processed, the total quantity of heat required is

$$_AQ_B = 24.111 \times 20{,}000/14.651 = 32{,}914 \text{ Btu per minute.}$$

On the Mollier Diagram the process is represented by the horizontal line AB, Fig. 2, whose length is the quantity of heat added per pound of dry air. The reduced enthalpy at A is 4.92 while that at B is 29.03, both being read directly from the chart. Since humidity ratio is constant the difference between these reduced enthalpies is also the difference between the enthalpies themselves, namely, 24.11 Btu/lb_a.

Cooling

The process of cooling moist air is also represented by a horizontal line on the Mollier Diagram. The line may extend across the saturation curve into the two-phase region, nevertheless, the length of the line between

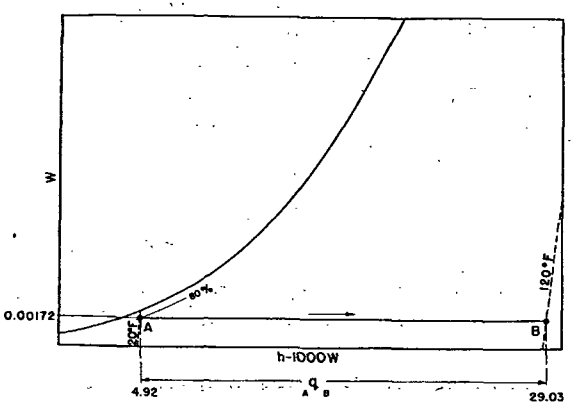


FIG. 2. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 6

the initial and final states is the quantity of heat removed, or refrigeration supplied, per pound of dry air. By following the final isotherm downward to the right to the saturation curve and reading the ordinate there, the weight of water vapor per pound of dry air in the vapor phase is determined. The difference between the initial humidity ratio and this ordinate is the weight of condensed phase per pound of dry air in the final state.

Example 7. Air at 95 F and 50 per cent saturation is cooled to 70 F. Find the refrigeration required to process 20,000 cfm of uncooled air.

Solution. From the data in Table 1: the initial humidity ratio is $0.50 \times 0.03673 = 0.01837$ lb_w/lb_a; the initial enthalpy is $22.827 + 0.50 \times 40.49 = 43.072$ Btu/lb_a; the humidity ratio at saturation at the final temperature is 0.01582 lb_w/lb_a; the quantity of liquid formed is $0.01837 - 0.01582 = 0.00255$ lb_w/lb_a; the enthalpy of the final two-phase mixture is $34.09 + 0.00255 \times 38.11 = 34.187$ Btu/lb_a.

It may be supposed that the air is cooled between two sections of a duct. The quantities of energy convected across the two sections per pound of dry air crossing them

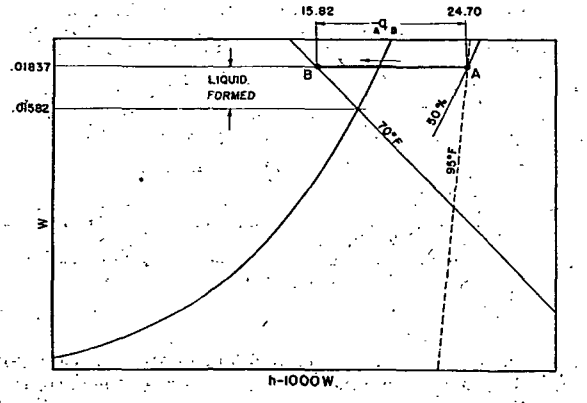


FIG. 3. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 7

are the two enthalpies calculated above. Conservation of energy requires that the difference between these two enthalpies be the quantity of heat removed, or refrigeration supplied, between the two sections. Therefore,

$$-{}_AQ_B = 43{,}072 - 34.187 = 8.885 \text{ Btu/lb}_a$$

The initial volume is $13.980 + 0.50 \times 0.822 = 14.391$ cu ft/lb_a. Since 20,000 cfm of air is to be processed, the total refrigeration required is

$$-{}_AQ_B = 8.885 \times 20{,}000 \div 14.391 = 12{,}348 \text{ Btu per minute}$$

On the Mollier Diagram the process is represented by the horizontal line AB, Fig. 3, whose length is the quantity of refrigeration required per pound of dry air.

Adiabatic Mixing of Two Air Streams

A typical air conditioning process requiring special analysis is the adiabatic mixing of two air streams. Referring to Fig. 4, let m_1, m_2, m_3 denote the weights of dry air convected across sections F_1, F_2, F_3, respectively, per minute. Then m_1W_1, m_2W_2, m_3W_3 and m_1h_1, m_2h_2, m_3h_3 will

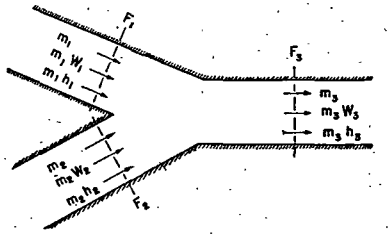


FIG. 4. ADIABATIC MIXING OF 2 AIR STREAMS

denote the weights of water and the quantities of energy similarly convected. If the mixing is adiabatic, it must be governed by the three equations,

$$\begin{aligned} m_1 + m_2 &= m_3 \\ m_1W_1 + m_2W_2 &= m_3W_3 \\ m_1h_1 + m_2h_2 &= m_3h_3 \end{aligned} \tag{8}$$

Elimination of m_3 gives,

$$\frac{h_2 - h_3}{h_3 - h_1} = \frac{W_2 - W_3}{W_3 - W_1} = \frac{m_1}{m_2} \tag{9}$$

according to which: *on the Mollier Diagram the state point of the resulting mixture lies on the straight line connecting the state points of the two streams being mixed and divides the line into two segments which are in the same ratio as are the weights of dry air in the two streams.*

Example 8. Outside Air at 0 F and 80 per cent saturation is to be mixed adiabatically with recirculated Inside Air at 70 F and 20 per cent saturation in the ratio of one pound of dry air in the former to seven in the latter. Find the temperature and degree of saturation of the resulting mixture.

Solution. The humidity ratio W_3 and the enthalpy h_3 of the resulting mixture must satisfy Equations 9, namely,

$$\frac{0.003164 - W_3}{W_3 - 0.000630} = \frac{20.270 - h_3}{h_3 - 0.668} = \frac{1}{7}$$

from which: $W_3 = 0.002847$, $h_3 = 17.820$. At the temperature of the resulting mixture, therefore,

$$h_a + 0.002847\, h_{as}/W_s = 17.820$$

At 61 F the lefthand member of this equation has the value 17.750; at 62 F its value is 17.991; by interpolation the temperature of the resulting mixture is 61.29 F where the humidity ratio at saturation, also by interpolation, is 0.01161; hence the degree of saturation of the resulting mixture is

$$\mu = 0.002847 \div 0.01161 = 24.52 \text{ per cent}$$

On the Mollier Diagram, Fig. 5, a straight line is drawn between point 1 (0 F, 80 per cent) and point 2 (70 F, 20 per cent); then point 3 is located on the line one-eighth of the distance from point 2 to point 1. The temperature and degree of saturation at point 3 are read directly.

Adiabatic Mixing with Injected Water

Another typical air conditioning process is that of injecting water into an air stream to mix with it adiabatically. Let $W_2 - W_1$, denote the increase in humidity ratio of the air; this is obviously the quantity of water injected per pound of dry air; it follows that the quantity of energy injected per pound of dry air is $(W_2 - W_1)h_w$, where h_w denotes the specific enthalpy of the water as injected; if the process is adiabatic this produces an equal increase in the enthalpy of the air, namely, $h_2 - h_1$; therefore,

$$h_2 - h_1 = h_w(W_2 - W_1) \qquad (10)$$

according to which: *the process of injecting water into an air stream to mix adiabatically with it is represented by a straight line on the Mollier Chart whose direction is fixed by the specific enthalpy of the water as injected.* The protractor drawn on the Mollier Diagram provided with this book provides a convenient means for determining this direction.

Example 9. It is desired to increase the humidity ratio of air at 70 F dry-bulb, without changing its temperature. Under what conditions may water be injected in order to accomplish the desired result:

Solution. At 70 F the increase of enthalpy per unit increase of humidity ratio is $h_{as}/W_s = 17.27 \div 0.01582 = 1092$ Btu per pound of water. This must be the specific enthalpy of the water added if the state point of the air is to be moved along the 70 F isotherm. Saturated steam at 668 F has this specific enthalpy (see Keenan-Keyes, Thermodynamic Properties of Steam).

On the Mollier Diagram, Fig. 6, it is seen that the 70 F isotherm is parallel to the line on the protractor for a specific enthalpy of 1092 Btu per pound.

Adiabatic Saturation

Any process by which the state point of moist air is moved to the saturation curve adiabatically may properly be called adiabatic saturation.

Example 10. Liquid water chilled to 35 F is evaporated into an air stream initially at 90 F and 50 per cent saturation. How much water must be evaporated to bring the air to saturation at what temperature.

Solution. The initial enthalpy of the air is $21.625 + 0.50 \times 34.31 = 38.780$ Btu/lb$_a$; the initial humidity ratio is $0.50 \times 0.03118 = 0.01559$ lb$_w$/lb$_a$; the specific enthalpy of the chilled water is 3.06 Btu/lb$_w$; therefore, the temperature at which the air reaches the saturation curve must be such that the enthalpy h_s and humidity ratio W_s at saturation satisfy the equation,

$$h_s - (W_s - 0.01559) \times 3.06 = 38.780$$

The solution is 75.19 F where the humidity ratio at saturation is 0.01894; consequently,

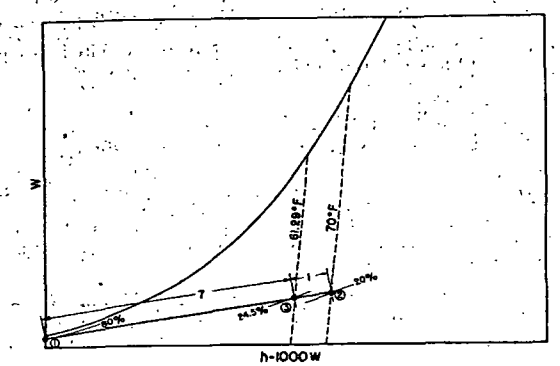


FIG. 5. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 8

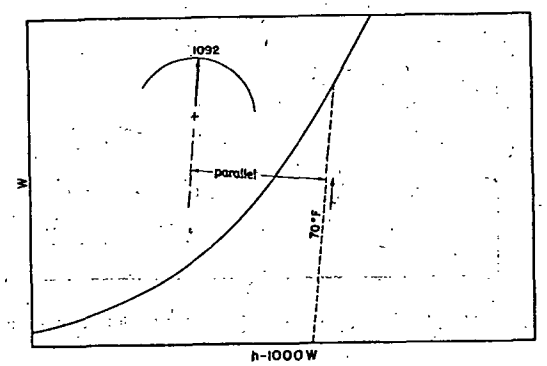


FIG. 6. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 9

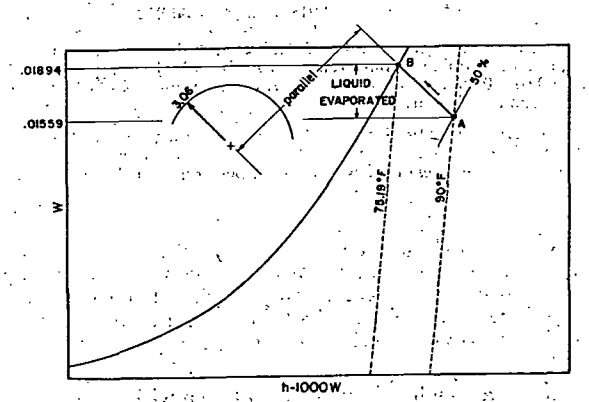


FIG. 7. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 10

the weight of water evaporated is 0.01894 − 0.01559 = 0.00335 lb per pound of dry air.

On the Mollier Diagram, Fig. 7, a line is drawn through the initial state point (90 F, 50 per cent saturation) in the direction given by the protractor for a specific enthalpy of 3.06 Btu/lb_w.

If air is saturated adiabatically with spray water which is recirculated, the water will ultimately assume a temperature such that the air is brought to saturation at exactly the same temperature; that is, the water will assume the thermodynamic wet-bulb temperature of the air.

Example 11. Air at 75 F and 60 per cent saturation is saturated adiabatically with recirculated spray water. Find the resulting temperature and the weight of water added per pound of dry air.

Solution. In view of the above remarks the solution of this example reduces to the determination of the thermodynamic wet-bulb temperature of the air. Its humidity

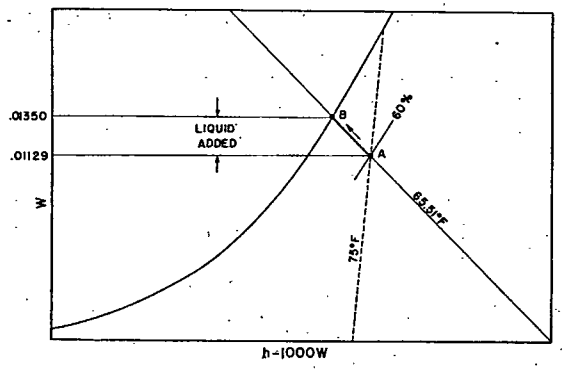


FIG. 8. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 11

ratio is 0.60 × 0.01882 = 0.01129; its enthalpy is 18.018 + 0.60 × 20.59 = 30.372; Equation 7 defining thermodynamic wet-bulb temperature becomes

$$h_s^* - (W_s^* - 0.01129)\, h_w^* = 30.372$$

At 65 F the value of the lefthand member is 29.995; at 66 F its value is 30.746; by interpolation the thermodynamic wet-bulb temperature is 65.51 F where the humidity ratio at saturation is 0.01350; consequently the weight of water added is 0.01350 − 0.01129 = 0.00221 lb per pound of dry air.

On the Mollier Diagram, Fig. 8, the process is represented by the line AB which is a segment of the 65.51 F thermodynamic wet-bulb line. The difference between the ordinates at B and at A is the weight of water added per pound of dry air.

Cooling Load

The problem of calculating the cooling load for an air conditioned space usually reduces to the determination of the quantity of *inside air* that must be withdrawn and the condition to which it must be brought by suitable processing so that its return to the conditioned space will have the net effect of removing given amounts of energy and water from the space.

Let M denote the weight of dry air withdrawn with *inside air* per hour. With it will be withdrawn energy of amount Mh_1 and water of amount

MW_1 per hour, where h_1 and W_1 denote the enthalpy and humidity ratio of the *inside air*, respectively. The weight of dry air returned with the *conditioned air* will necessarily be the same as that withdrawn with the *inside air*, but with it must be returned a smaller quantity of energy Mh and a smaller quantity of water MW. Let ΔQ and ΔW denote the given amounts of energy and water to be removed from the conditioned space per hour; then

$$Mh = Mh_1 - \Delta Q$$
$$MW = MW_1 - \Delta W$$

Eliminating M and letting q denote the ratio of energy removed to water removed, that is, $q = \Delta Q/\Delta W$,

$$\frac{h - h_1}{W - W_1} = q \qquad (11)$$

according to which: *all possible states for the Conditioned Air lie on a straight line on the Mollier Diagram passing through the state point of the inside air in the direction specified by the numerical value of the ratio q.* This line is called the *condition line* for the given problem. If the condition line crosses the saturation curve, the point of intersection is called the *apparatus dew-point* for the given problem.

The protractor on the Mollier Diagram facilitates the drawing of the condition line and the locating of the apparatus dew-point. For this purpose the numerical value of the ratio q is to be regarded as a value of the *specific enthalpy of water added, Btu per pound.*

Example 12. A condition of 80 F dry-bulb, and 67 F thermodynamic wet-bulb, is to be maintained in a clothing store, outside conditions being 95 F dry-bulb, and 75 F thermodynamic wet-bulb. The energy gain from normal heat transmission is estimated at 16,000 Btu per hour, that from solar radiation at 48,000 Btu per hour. The energy generated by lights, fans, etc. is estimated at 13,900 Btu per hour. The ventilation requirement is 30,000 cu ft per hour. The number of occupants is 50. Find the apparatus dew-point.

Solution. The properties of *inside air* and *outside air* are readily calculated from the data in Table 1, see especially Example 2.

		Inside Air	Outside Air
μ	=	0.5024	0.3848
h	=	31.514	38.408
W	=	0.01122	0.01413
v	=	.....	14.296

The weight of dry air entering with the *ventilating air* is 30,000/14.296 = 2098.5 lb per hour which brings with it energy of amount 2098.5 × 38.408 = 80.595 Btu per hour and water of amount 2098.5 × 0.01413 = 29.659 lb per hour.

The weight of dry air displaced from the store by the *ventilating air* is 2098.5 lb per hour which takes with it energy of amount 2098.5 × 31.514 = 66,132 Btu per hour and water of amount 2098.5 × 0.01122 = 23.541 lb per hour.

Each occupant may be regarded as a normal person standing at rest and therefore evaporating 0.198 lb of water per hour at about 79 F (Table 4, Chapter 12). From this source there is water of amount 50 × 0.198 = 9.90 lb per hour and energy of amount 9.90 × 1095.7 = 10.847 Btu per hour added to the conditioned space. In addition each occupant loses 225 Btu per hour by conduction, convection, and radiation, making a total for 50 persons of 11,300 Btu per hour.

The net energy gain is 16,000 + 48,000 + 13,900 + 80,595 − 66,132 + 10,847 + 11,300 = 114,510 Btu per hour. The net water gain is 29.659 − 23.541 + 9.90 = 16.018 lb per hour. Accordingly the direction of the condition line is fixed by the ratio, q = 114,510 ÷ 16.018 = 7148.8 Btu per pound of water.

On the Mollier Diagram, Fig. 9, the direction of the condition line is given by the protractor for a *specific enthalpy of water added* of 7148.8 Btu per pound. The line itself

passes through the state point of the Inside Air and intersects the saturation curve at the apparatus dew-point.

According to Equation 11 the enthalpy h_s and humidity ratio W_s at the apparatus dew-point must satisfy the equation

$$7148.8W_s - h_s = 7148.8 \times 0.01122 - 31.514 = 48.681$$

At 58 F the left-hand member has the value 48.513; at 59 F its value is 50.641; by interpolation the apparatus dew-point is 58.08 F.

It would be a mistake to assume that the refrigeration to be supplied is equal to the net energy to be removed; for in general water is to be removed simultaneously and unless this is removed as liquid at 32 F it will automatically take some energy with it. Thus, unless the water is removed as solid (ice) the refrigeration to be supplied will be somewhat less than the net energy to be removed.

Example 13. Referring to the cooling load problem of Example 12, suppose that the conditioning process consists of cooling a portion of the *inside air* to the apparatus dew-point temperature, separating out the liquid thus formed, and returning the resulting saturated mixture to the conditioned space. Find the quantity of *inside air* that must be processed in this manner and the corresponding quantity of refrigeration required.

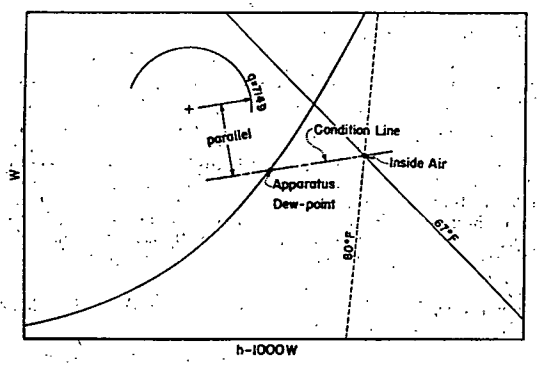


FIG. 9: ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 12

Solution. During the cooling operation the enthalpy of the Inside Air is reduced to the value,

$$h = 25.17 + (0.01122 - 0.01033) \times 26.20 = 25.193$$

where 25.17 and 0.01033 are the values of enthalpy and humidity ratio at saturation at the apparatus dew-point temperature and 26.20 is the specific enthalpy of liquid water at that temperature. It follows that the quantity of refrigeration required is 31.514 − 25.193 = 6.321 Btu per pound of dry air.

The *inside air* being processed leaves the store with an enthalpy of 31.514 and is returned with an enthalpy of 25.17; it therefore removes energy of amount 6.344 Btu per pound of dry air. This means that the weight of dry air involved in the process is 114,510/6.344 = 18,050 lb per hour and that the total refrigeration to be supplied is 18,050 × 6.321 = 114,090 Btu per hour, or 9.508 tons.

The quantity of liquid separated out during the conditioning process is 18,050 × (0.01122 − 0.01033) = 16.018 lb per hour as required. In leaving the apparatus it takes with it energy of amount 16.018 × 26.20 = 420 Btu per hour. This plus the refrigeration accounts for the total energy removal of 114,510 Btu per hour as required.

On the Mollier Diagram, Fig. 10, the cooling operation is represented by line AB whose length is the quantity of refrigeration per pound of dry air; the separation of the liquid

formed in the cooling operation is represented by line BC whose projection on the ordinate axis is the quantity of liquid so separated per pound of dry air. Point C is the apparatus dew-point and lies on the condition line as required.

In practice it may not be feasible to choose the apparatus dew-point as the point on the condition line to which to condition the *inside air* because to do so would require an excessive number of air changes in the given space. Or it may be that the condition line does not cross the saturation curve at all so that the apparatus dew-point as defined does not exist. Finally, it is rarely possible to obtain complete saturation in conventional air conditioning apparatus. Nevertheless the requirements of the cooling load problem can be exactly met if the conditioned air is brought to *any* point on the condition line of the problem.

Heating Load

The condition line is also useful in the analysis of heating load problems as may best be illustrated by means of an illustrative example.

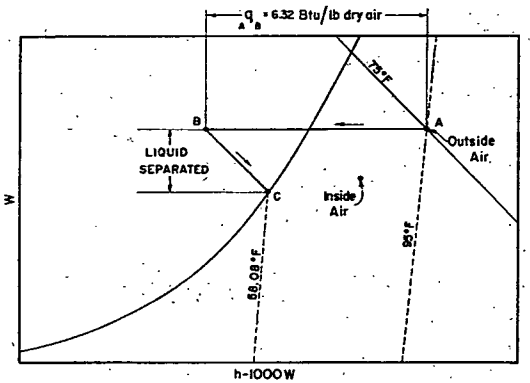


FIG. 10. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 13

Example 14. A certain space is to be maintained at 70 F and 50 per cent saturation with outside conditions at 0 F and 80 per cent saturation. The normal heat transmission through walls, partitions, floor, roof, glass and doors is estimated at 75,000 Btu per hour. Energy gained from lights and appliances is estimated at 15,000 Btu per hour. Energy and water gains from occupants are to be disregarded in the calculations. Double doors and windows are used so that infiltration is negligible. The ventilation requirement is 30,000 cu ft per hour of *outside air*.

The requirements of the problem are to be met in the following manner; preheat the *ventilating air*; mix it adiabatically with recirculated *inside air*; saturate the mixture adiabatically with recirculated spray water; heat the resulting mixture to 105 F and return it to the conditioned space as *supply air*.

Analysis. Every pound of dry air admitted to the system (air conditioned space plus air conditioning apparatus) with the *ventilating air* displaces a pound of dry air from the system with *inside air*. Since the *ventilating air* is not admitted directly to the space, then for every pound of dry air withdrawn with *inside air* there is a pound of dry air returned with *supply air*. This has to have the net effect of adding energy of amount 60,000 Btu per hour and water of amount *zero* pounds per hour. Thus the ratio q determining the direction of the condition line is infinite, which means that the condition line is horizontal as indicated by the protractor on the Mollier Diagram.

The properties of *inside air* are: $h = 25.451$, $W = 0.007910$. Since the state point of the *supply air* must be on the condition line at 105 F, its properties are: $h = 33.986$,

$W = 0.007910$. Therefore the weight of dry air withdrawn with *inside air* and returned with *supply air* is 60,000 ÷ (33.986 − 25.451) = 7029.9 lb per hour.

The properties of *outside air* are: $h = 0.668$, $W = 0.0006298$, $v = 11.590$. Therefore the weight of dry air introduced into the system with the *ventilating air* is 30,000 ÷ 11.590 = 2588.4 lb per hour. This *ventilating air* is to be mixed adiabatically with *inside air* containing 7029.9 − 2588.4 = 4441.5 lb of dry air per hour; therefore, the humidity ratio of the mixture must be (2588.4 × 0.0006298 + 4441.5 × 0.007910) ÷ 7029.9 = 0.005229.

The condition line crosses the saturation curve at 50.86 F where the enthalpy is 20.782 and the humidity ratio is 0.007910. This is the state point to be reached by adiabatic saturation of the mixture of *ventilating air* and *inside air* with recirculated spray water. Accordingly, the state point of the mixture must lie on the 50.86 F thermodynamic wet-bulb line so that its enthalpy must have the value,

$$h = 20.782 - (0.007910 - 0.005229) \times 18.97 = 20.731$$

This requires that the enthalpy of the preheated *ventilating air* have the value,

$$h = (7029.9 \times 20.731 - 4441.5 \times 25.451) \div 2588.4 = 12.632$$

Since the humidity ratio of the preheated *ventilating air* is known to be 0.0006298, its temperature is readily found to be 49.75 F.

The quantity of heat required for preheating the *ventilating air* is 2588.4 × (12.632 − 0.668) = 30,968 Btu per hour; that to be added to the *supply air* is 7029.9 × (33.986 − 20.782) = 92,823 Btu per hour; the energy added with the spray water is 7029.9 × 18.97 × (0.007910 − 0.005229) = 357 Btu per hour; that introduced into the system with the *ventilating air* is 2588.4 × 0.668 = 1729 Btu per hour; that carried out of the system with the *inside air* displaced by the *ventilating air* is 2588.4 × 25.451 = 65,877 Btu per hour; therefore, the net energy added to the system is 30,968 + 92,823 + 357 + 1729 − 65,877 = 60,000 Btu per hour as required.

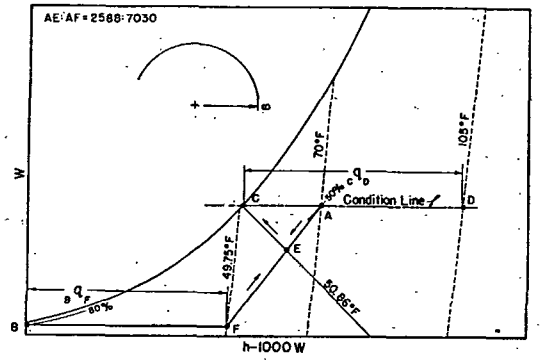


FIG. 11. ILLUSTRATION OF USE OF MOLLIER DIAGRAM IN SOLUTION OF EXAMPLE 14

On the Mollier Diagram, Fig. 11, point A is the state point of the *inside air*. The condition line is horizontal so that *point* D is the state point of the *supply air*. The condition line crosses the saturation curve at point C so that the state point of the mixture of preheated *ventilating air* and *inside air* before adiabatic saturation with recirculated spray water must lie somewhere on the thermodynamic wet-bulb line through C. The state point of the *ventilating air* is point B, hence that of the preheated *ventilating air* must lie somewhere on the horizontal line through B. Its exact location is determined graphically by finding the straight line AF which is cut by the thermodynamic wet-bulb line through C into two segments such that AE : AF = 2588.4 : 7029.9. The length of the line BF is the quantity of heat required for preheating the *ventilating air* per pound of dry air; the length of the line CD is the quantity of heat to be added to the *supply air*, per pound of dry air.

WET-BULB TEMPERATURES BELOW 32F

A condition in which the water evaporating from the wick of a wet-bulb thermometer remains liquid at 32 F or lower is one of metastable equilibrium and should therefore not be expected to occur in practice. The evidence that it does sometime occur appears to be indirect and inconclusive. Stable equilibrium requires that the water freeze at 32 F or lower and is the condition to be expected in practice. On the Mollier Diagram the lines of constant thermodynamic wet-bulb temperature have been drawn for stable equilibrium only. In other words it has been assumed that the water evaporating from the wick of the wet-bulb thermometer freezes when its temperature falls to 32 F or lower.

Example 15. Find the temperature at which dry air has a thermodynamic wet-bulb temperature of 32 F.

Solution. If it is assumed that the water evaporating from the wick of the wet-bulb thermometer remains liquid, the specific enthalpy of the dry air must have the value,

$$h_a = 11.758 - 0.04 \times 0.003788 = 11.758$$

corresponding to which the temperature is 48.95 F. On the other hand if it is assumed that the water freezes, the specific enthalpy of the dry air must have the value,

$$h_a = 11.758 + 143.36 \times 0.003788 = 12.301$$

corresponding to which the temperature is 51.21 F. The second assumption is the assumption of stable equilibrium and should be expected to represent the actual situation. The corresponding answer, namely, 51.21 F is the one given by the Mollier Diagram.

DALTON'S RULE

As stated in the introduction the thermodynamic properties of moist air have hitherto been obtained from those of dry air and water vapor separately by application of Dalton's Rule. Actual departures from the rule are due principally, but not entirely, to intermolecular forces; therefore, in order to apply the rule with any measure of consistency it is necessary to idealize the situation by assuming that the effects of such intermolecular forces are negligible and that both the dry air and the water vapor behave like perfect gases. Making this assumption, the volume v_T occupied by n_a mols of dry air at temperature T and pressure p_a is $v_T = n_a RT/p_a$ while that occupied by n_w mols of water vapor at the same temperature but at pressure p_w is $v_T = n_w RT/p_w$. According to Dalton's Rule, if the dry air and water vapor are mixed, each occupies the whole volume of the mixture at the temperature of the mixture and the pressure of the mixture is the sum of the individual pressures. Mathematically,

$$v_T = \frac{n_a RT}{p_a} = \frac{n_w RT}{p_w} = \frac{(n_a + n_w)RT}{p} \qquad (12)$$

It follows from these equations that the so-called "partial" pressure of each constituent is its mol-fraction times the observed pressure of the mixture; thus, for water vapor,

$$p_w = \frac{n_w}{n_a + n_w} p \qquad (13)$$

and similarly for dry air. Equation 13 may be regarded as the Dalton Rule definition of partial pressure in terms of the observable terms n_a, n_w, p.

The humidity ratio W is the mol ratio n_w/n_a times the ratio of molecular weights, namely, 18.016/28.966 = 0.6220; hence (13) can be written

$$W = 0.6220 \frac{p_w}{p - p_w} \tag{14}$$

Now, even if it is assumed that both the dry air and the water vapor behave like perfect gases, it does not follow that at saturation the "partial" pressure of the water vapor can be put equal to the saturation pressure of pure water at the temperature of the mixture because; (1) the coexisting liquid (or solid) phase is not pure water but contains a small amount of dissolved air, and (2) the coexisting liquid (or solid) phase has to support the observed pressure p and not just the saturation pressure p_s of pure water. These effects are calculable but are in general smaller than the effects of intermolecular forces which have already been ignored. Besides, the only legitimate reason for retaining Dalton's Rule is to gain simplicity; hence these effects should be disregarded also, and the humidity ratio at saturation estimated as follows,

$$W_s = 0.6220 \frac{p_s}{p - p_s} \tag{15}$$

In this chapter the ratio W/W_s has been called degree of saturation and denoted by the greek letter μ. The ratio p_w/p_s has long been called *relative humidity* and will be denoted by the Greek letter φ. Equations 14 and 15 can be combined to give

$$\mu = \varphi \frac{1 - p_s/p}{1 - \varphi p_s/p} \tag{16}$$

which can be inverted to give

$$\varphi = \frac{\mu}{1 - (1 - \mu)p_s/p} \tag{17}$$

Example 16. Find the relative humidity of moist air at 180 F, 20 per cent saturation.

Solution. Inserting numerical data from Table 1 into (17), the answer is

$$\varphi = \frac{0.20}{1 - 0.80 \times 15.294/29.921} = 0.3384$$

Example 17. Find the degree of saturation of moist air at 70 F, 50 per cent relative humidity.

Solution. Inserting numerical data from Table 1 into Equation 16, the answer is

$$\mu = 0.50 \frac{1 - 0.73915/29.921}{1 - 0.50 \times 0.73915/29.921} = 0.4937$$

The foregoing examples show that there is a substantial difference between degree of saturation and relative humidity, particularly at higher temperatures. Of course, they both have the value zero for dry air and the value unity for saturated moist air regardless of the temperature.

A Dalton Rule expression for the volume of moist air per pound of dry air obtainable directly from Equation 12 is

$$v = \left(\frac{R_a T}{p}\right) + \mu \left(\frac{W_s R_w T}{p}\right) \tag{18}$$

where

R_a = gas constant for dry air = 1545.31 ÷ 28.966 = 53.349 (ft/F).

R_w = gas constant for water vapor = 1545.31 ÷ 18.016 = 85.774 (ft/F).

This expression is of the form of Equation 4.

According to Dalton's Rule the enthalpy of moist air is the sum of separate contributions from the dry air and the water vapor; thus,

$$h = h_a + \mu(W_s h_w) \tag{19}$$

where, to be consistent, the specific enthalpies h_a and h_w should be allowed to vary with temperature only, not with pressure or composition. This expression is of the form of Equation 5.

Within the accuracy of Dalton's Rule the following empirical equations give suitable values of h_a and h_w:

$$\begin{aligned} h_a &= 0.240t \\ h_w &= 0.444t + 1061 \end{aligned} \tag{20}$$

Equation 7 defining thermodynamic wet-bulb temperature may be written in the form,

$$h - h' + (W_s^* - W)h_w^* = h_s^* - h'$$

If the quantity h' that has been subtracted from both sides is understood to be the enthalpy at the thermodynamic wet-bulb temperature t^* but at the humidity ratio W, then within the accuracy of Dalton's Rule

$$\begin{aligned} h - h' &= (0.240 + 0.444W)(t - t^*) \\ h_s^* - h' &= (1061 + 0.444t^*)(W_s^* - W) \\ h_w^* &= t^* - 32 \end{aligned}$$

With these approximations Equation 7 becomes

$$W_s^* - W = \frac{0.240 + 0.444W}{1093 - 0.556t^*}(t - t^*) \tag{21}$$

Carrier [6] has modified Equation 21 by introducing further approximations as follows,

$$\begin{aligned} W_s^* &= 0.6220p_s^*/(p - p_s^*) \\ W &= 0.6220p_w/(p - p_s^*) \\ 0.444\ W &= 0 \end{aligned}$$

the first of which is part of Dalton's Rule. The result is

$$p_w = p_s^* - \frac{p - p_s^*}{2830 - 1.44t^*}(t - t^*) \tag{22}$$

except that the numerical values of the constants in the denominator of the rightmost term are somewhat different than Carrier's.

Equation 22 permits direct calculation of the partial pressure p_w from observed values of pressure p, temperature t, and wet-bulb temperature t^*, assuming that information is available regarding the saturation pressure p_s. The ratio p_w/p_s is the so-called relative humidity.

Example 18. Find the relative humidity of moist air at 90 F dry-bulb, and 63 F (thermodynamic wet-bulb).

Solution. At 63 F the value of the saturation pressure is 0.58002 in. Hg. Therefore, at atmospheric pressure (29.921 in. Hg),

$$p_w = 0.58002 - 29.341 \times 27/2739 = 0.2908 \text{ in. Hg}$$

The relative humidity is

$$\varphi = 0.2908/1.4219 = 0.2045$$

the denominator being the value of saturation pressure at 90 F.

From Equation 16 may be computed the corresponding degree of saturation, the result being

$$\mu = 19.67 \text{ per cent}$$

in remarkably close agreement with the answer to Example 2.

STEADY FLOW ENERGY EQUATION

In steady flow, the energy convected by the fluid at any section is the sum of (*a*) kinetic energy due to velocity; (*b*) gravitational energy due to elevation; (*c*) enthalpy due to the condition of pressure, temperature and composition of the fluid.

Kinetic Energy

There are reasons to believe that the so-called *velocity pressure* h_v read by a Pitot tube is simply the kinetic energy per unit volume of the fluid immediately upstream from the tube, as application of Bernoulli's Equation suggests. Thus

$$V = 1097.3 \sqrt{\frac{h_v}{\rho}} \tag{23}$$

where

V = velocity, feet per minute.

h_v = velocity pressure, inches of water at 60 F.

ρ = density of fluid, pounds per cubic foot.

In the case of flow through a duct, the velocity pressure is found to vary considerably over the section and a traverse has to be made. The cross-sectional area of the duct is divided into a number of equal concentric areas, and measuring stations are located at centroidal points in each area along two perpendicular diameters. Usually the ultimate object is to determine an *average* velocity $\overline{V}$ from which the weight of fluid crossing the section per unit time can be obtained on multiplying by the cross-sectional area of the duct and by the density of the fluid. This is obtained by simply averaging the square roots of all measured velocity pressures as follows:

$$\overline{V} = \frac{1097.3}{\sqrt{\rho}} \left(h_v^{1/2} \right)_{av} \tag{24}$$

where

$\overline{V}$ = average velocity, feet per minute.

$(h_v^{1/2})_{av}$ = arithmetic average of the square roots of all measured velocity pressures, inches of water at 60 F.

But the item of present importance is the *average kinetic energy* convected with each pound of fluid. Consistently with the previous discussion, this can be shown to be

$$\overline{KE} = 0.006678\, v \frac{(h_v^{3/2})_{av}}{(h_v^{1/2})_{av}} \tag{25}$$

where

$\overline{KE}$ = average kinetic energy, Btu per pound.

v = specific volume, cubic feet per pound.

$(h_v^{3/2})_{av}$ = arithmetic average of the 3/2-powers of all measured velocity pressures, inches of water at 60 F.

If the velocity pressure were uniform over the section, Equations 24 and 25 could be combined to give

$$\overline{KE} = \left(\frac{\overline{V}}{13{,}430} \right)^2 \tag{26}$$

But, it is interesting to note that if the velocity varies *parabolically* from zero at the walls to maximum at the center as it does in the case of purely viscous flow in a circular duct, *then the average kinetic energy is twice that given by* Equation 26.

Example 19. If 2000 cfm of air flows through an 8 in. diameter circular duct, find the average kinetic energy per pound of air.

Solution. The cross-sectional area of the duct is 0.349 sq ft; hence the average flow velocity is 5730 fpm. If the velocity were uniform over the section, the average kinetic energy would be $(5730 \div 13{,}430)^2 = 0.182$ Btu per pound. But it is more likely that the actual distribution of velocity would approximate that characteristic of viscous flow; hence the average kinetic energy would be more nearly $2 \times 0.182 = 0.364$ Btu per pound.

Gravitational Energy

The potential energy due to elevation Z (feet) above any convenient datum is simply $Z \div 778.3$ Btu per pound of fluid. In the case of moist air,

$$\overline{PE} = \frac{\overline{Z}\,(1 + W)}{778.3} \tag{27}$$

where

$\overline{PE}$ = average potential energy, Btu per pound dry air.

$\overline{Z}$ = average elevation, feet.

W = humidity ratio, pound water per pound dry air.

Enthalpy

No further discussion of enthalpy is required. It may be well to emphasize, however, that enthalpies have been figured on the basis of one pound of dry air.

Heat and Shaft Work

Between any two sections 1 and 2 in an apparatus through which steady flow occurs, there may be heat *absorbed* from outside, ${}_1q_2$, Btu per pound of dry air, and shaft work *removed* to outside, ${}_1l_2$, Btu per pound of dry air. If heat is actually rejected to outside, ${}_1q_2$ is intrinsically negative; and if shaft work is actually put in from outside ${}_1l_2$, is intrinsically negative.

Steady-flow Energy Equation

A complete energy accounting takes the form of Equation 28 which is usually referred to as the steady-flow energy equation.

$${}_1q_2 = (h_2 + \overline{KE}_2 + \overline{PE}_2) - (h_1 + \overline{KE}_1 + \overline{PE}_1) + {}_1l_2 \tag{28}$$

where

$_1q_2$ = heat *added* from outside *between* sections 1 and 2, Btu per pound dry air.
h_2 = enthalpy of the mixture *at* section 2, Btu per pound dry air.
$\overline{KE}_2$ = average kinetic energy *at* section 2, Btu per pound dry air.
$\overline{PE}_2$ = average potential energy *at* section 2, Btu per pound dry air.
h_1 = enthalpy *at* section 1, Btu per pound dry air.
$\overline{KE}_1$ = average kinetic energy *at* section 1, Btu per pound dry air.
$\overline{PE}_1$ = average potential energy *at* section 1, Btu per pound dry air.
$_1l_2$ = shaft work withdrawn *between* sections 1 and 2, Btu per pound dry air.

In Equation 28 all quantities are *per pound of dry air*. If Equation 25 is used in computing average kinetic energy, the result will be in Btu per pound of dry air if v is taken as volume per pound of dry air. If Equation 26 is used, multiplication by $(1 + W)$ as in Equation 27 is required though this is a refinement seldom justified.

Thermodynamic properties of water at saturation are given in Table 2 for the range −160 to +212 F.

U. S. STANDARD ATMOSPHERE

The so-called U. S. Standard Atmosphere is an essential standard of reference in aeronautics and as such has become important to the air conditioning engineer who frequently has to simulate atmospheric conditions at high altitudes in connection with aeronautical research. In defining this standard it is first assumed that temperature T varies linearly with altitude Z above sea level, at any rate up to the lower limit of the isothermal layer at 35,332 ft. Thus,

$$T = T_o - 0.0019812\,Z \tag{29}$$

or

$$\frac{dT}{dZ} = -0.0019812 \text{ (degree Centigrade per foot)} \tag{30}$$

The second assumption is the validity of the perfect gas laws, namely,

$$Pv = RT \tag{31}$$

A horizontal disc of air having unit cross-sectional area (1 sq ft) and vertical thickness dZ (ft) weighs dZ/v (lb). This accounts for the difference of pressure dP (lb per sq ft) between the upper and lower faces of the disc; hence, using Equation 31

$$dZ = \frac{RT\,dP}{P} \tag{32}$$

Equations 30 and 32 can be combined to eliminate Z and then integrated to obtain the relation between pressure and temperature, namely,

$$\frac{T}{T_o} = \left(\frac{P}{P_o}\right)^{0.1903} \tag{33}$$

The values $T_o = 288\ K$ and $P_o = 29.921$ in. Hg are parts of the definition of the standard atmosphere.

Values of pressure and temperature are listed in Table 4 for altitudes in the standard atmosphere from −1,000 to 50,000 ft above sea level. Values for altitudes below the lower limit of the isothermal layer conform to Equations 29 and 33. For further explanation, reference (7) should be consulted.

TABLE 4. 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

LETTER SYMBOLS USED IN CHAPTER 3

μ = degree of saturation or per cent saturation.
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 3 for use in Equation 4a (obtained from Table 3).
B = coefficient to be used in Equation 5a (obtained from Table 3).
C = coefficient for use in Equation 6a (obtained from Table 3).
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_s^* = enthalpy of moist air at saturation at *thermodynamic wet-bulb temperature*, 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.
h_v = velocity pressure, inches of water at 60 F.
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^*, Btu per pound of dry air.
K = Kelvin degrees.
KE = kinetic energy, Btu per pound.
$\overline{KE}$ = average kinetic energy, Btu per pound.
l = shaft work withdrawn, Btu per pound of air.
$_1l_2$ = shaft work withdrawn between sections 1 and 2, Btu per pound of air.

LETTER SYMBOLS (Continued)

m = weight of dry air crossing any duct section, pounds per minute.

m_1, m_2, m_3 = weights of dry air convected across sections F_1, F_2, F_3 respectively, pounds per minute.

M = weight of dry air withdrawn with inside air, pounds per hour.

n_a = mols of dry air.

n_w = mols of water vapor.

p = total pressure of a mixture of air and water vapor, pounds per square inch or inches Hg.

p_a = partial pressure of 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.

P = atmospheric pressure, inches Hg.

P_o = standard atmospheric pressure by definition 29.921 in. Hg.

PE = potential energy, Btu per pound dry air.

$\overline{PE}$ = average potential energy, Btu per pound dry air.

q = ratio of energy added (or removed) to water added (or removed), Btu per pound. Also called specific enthalpy of *water added.*

${}_1q_2$ = heat added between sections 1 and 2, Btu per pound dry air.

${}_Aq_B$ = heat added between sections A and B per pound of dry air, Btu per pound.

${}_AQ_B$ = total heat added between sections A and B, Btu per minute.

ΔQ = energy to be removed from or added to conditioned space, Btu per hour.

R = universal gas constant.

R_a = gas constant for dry air.

R_w = gas constant for water vapor.

s = entropy of moist air per pound of dry air, Btu per (pound) (Fahrenheit degree).

$\bar{s}$ = correction to be added to entropy of moist air obtained from Equation 6.

$\bar{\bar{s}}$ = additional correction to be added to entropy because of "mixing entropy" (obtained from Table 3). Correction to be added to value of s obtained from Equation 6.

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^* = thermodynamic wet-bulb temperature, Fahrenheit degrees.

t(F) = temperature, Fahrenheit degrees.

T = absolute temperature, Fahrenheit degrees.

T_o = standard atmospheric temperature, by definition 288 Kelvin degrees.

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.

LETTER SYMBOLS (Continued)

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.

V = velocity, feet per minute.

$\overline{V}$ = average velocity, feet per minute.

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

ΔW = water to be removed from (or added to) conditioned space, pounds per hour.

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.

Z = elevation above any datum, feet.

$\overline{Z}$ = average elevation, feet.

REFERENCES

1—Thermodynamic Properties of Moist Air, by J. A. Goff and S. Gratch (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June 1945, p. 334).

2—Low Pressure Properties of Water in the Range, −160 to 212 F, by J. A. Goff and S. Gratch (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, February 1946).

3—Thermodynamic Properties of Steam, by J. H. Keenan and F. G. Keyes (John Wiley and Sons, Inc., New York, 1936).

4—Das i, x-Diagramm für Dampfluftgemische, by R. Mollier (ZVDI, 1929).

5—The Theory of the Psychrometer, by J. H. Arnold (*Physics*, Vol. 4, 1933).

6—Rational Psychrometric Formulae, by W. H. Carrier (*A.S.M.E. Transactions*, Vol. 33, 1911, p. 1005).

7—National Advisory Committee for Aeronautics, Technical Report No. 538, 1935.

CHAPTER 4

Fluid Flow

Theory; Pressure Loss in Circular and Non-Circular Pipes; Compressible Fluids; Nozzles and Orifices; Steam Flow Measurement; Metering Liquids; Nozzle Coefficients and Expansion Factors; Pitot Tube; Installation of Nozzles and Orifices.

THE flow of fluids is part of the branch of engineering science known as fluid mechanics, which 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 head equation based on energy considerations for 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:

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

where

V = velocity in feet per second.
g = acceleration due to gravity = 32.17 ft per (second) (second).
J = mechanical equivalent of heat = 778 foot pounds per Btu.
u = internal energy, in Btu per pound of fluid.
p = pressure in pounds per square foot.
v = specific volume, in cubic feet per pound.
E = mechanical work, in foot pounds per pound of fluid flowing.
q = heat transferred to the fluid, in Btu per pound of fluid flowing, and
z = elevation above some arbitraty datum, in 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

$$\frac{V_1^2}{2g} + Jh_1 + E + Jq + z_1 = \frac{V_2^2}{2g} + Jh_2 + z_2 \qquad (2)$$

The steady flow energy equation is applicable to a wide range of problems in the flow of fluids. Obviously it applies to flow through pipes, orifices, and nozzles, and to the flow through turbines and centrifugal pumps. Reciprocating engines and pumps are essentially intermittent, but the flow tends to become steady as the number of cylinders increases, and becomes practically uniform at the entrance and exit if the system includes receivers and pipes of sufficient size.

By substituting $\frac{p}{\rho}$ (where ρ is density in pounds per cubic foot) for

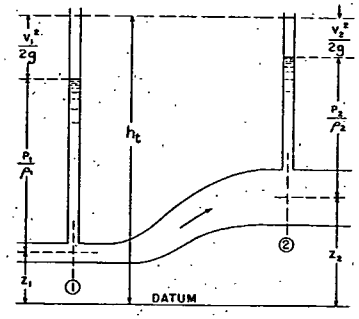


FIG. 1. RELATION OF VARIOUS FACTORS IN BERNOULLI EQUATION

its equivalent pv in Equation 1, the result, after rearranging, will be:

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho_1} + z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\rho_2} + z_2 + \left[J(u_2 - u_1) - E - Jq\right] \qquad (3)$$

In the case of flow through a pipe, no outside work is performed and, if the process is considered adiabatic, and if the change due to turbulence and friction is considered to be negligible, the bracketed expression in Equation 3 will disappear, leaving:

$$\frac{V_1^2}{2g} + \frac{p_1}{\rho_1} + z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\rho_2} + z_2 \qquad (4)$$

which is commonly called the Bernoulli equation, named after the Swiss mathematician and physician who first propounded the theory. $\frac{V^2}{2g}$ is

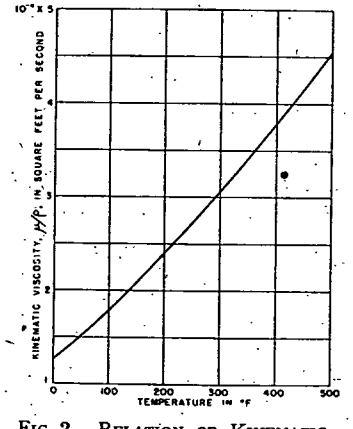


FIG. 2. RELATION OF KINEMATIC VISCOSITY TO TEMPERATURE OF AIR

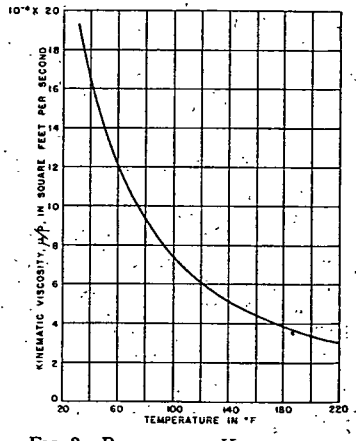


FIG. 3. RELATION OF KINEMATIC VISCOSITY TO TEMPERATURE OF WATER

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 head 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 head. 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 the *Bernoulli equation applies only to incompressible fluids*.

Pressure Loss in Circular Pipes

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

$$h_f = \frac{f\,l\,V^2}{2g\,d} \tag{5}$$

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 = 32.17 ft per (second) (second).
d = the internal diameter of the pipe, in feet, and,
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.

The factor f is a function of the Reynolds number

$$N_{Re} = \frac{d\,V\,\rho}{\mu} \tag{6}$$

where

N_{Re} = Reynolds number.
ρ = the density in pounds per cubic foot.
μ = the absolute viscosity in 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; it represents the formula

$$f = \frac{64}{N_{Re}} \tag{7}$$

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

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

$$V = \frac{\rho h_f}{4\mu l}(r^2 - L^2) \tag{8}$$

where

r = the radius of the pipe in feet.
L = distance perpendicularly from the axis of the pipe, in 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, there is an

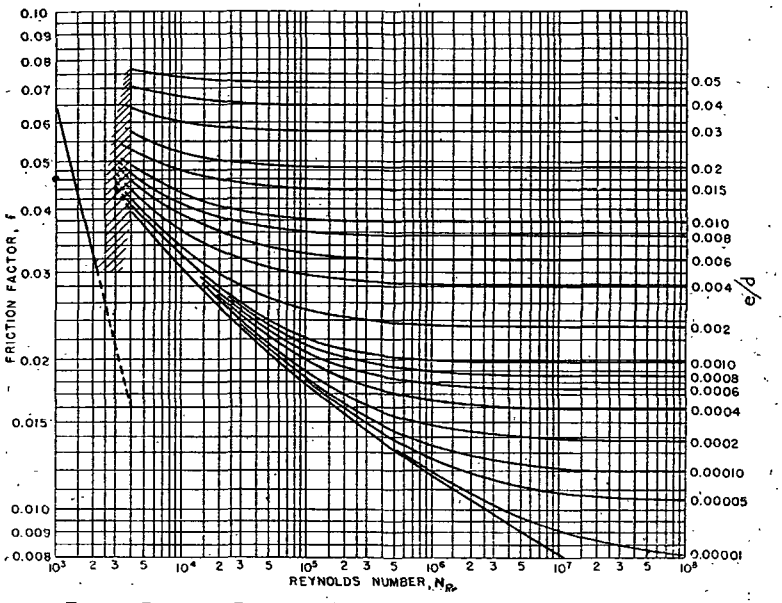


FIG. 4. RELATION BETWEEN FRICTION FACTOR AND REYNOLDS NUMBER
NOTE: The straight line at left shows values of Friction Factor for laminar flow.
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unstable region where the flow changes from laminar to turbulent, or vice versa. The actual value is impossible 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 the 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, which is a considerable

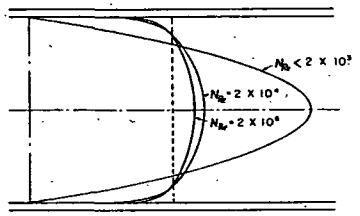


FIG. 5. COMPARISON OF VELOCITY PROFILES FOR 3 DIFFERENT REYNOLDS NUMBERS BUT FOR SAME AVERAGE VELOCITY

factor 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/d 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 to use in Equation 5.

Equation 5 is applicable to all liquids, and to gases when the pressure loss is less than 10 per cent of the initial pressure. When the loss in head is high, the formula to be used for gases is

$$\frac{p_1^2 - p_2^2}{p_1^2} = \frac{flV_1^2}{gd\ p_1 v_1} \tag{9}$$

which may be rearranged to give the loss in pressure

$$p_1 - p_2 = p_1\left[1 - \sqrt{1 - \frac{flV_1^2}{gd\ p_1 v_1}}\right] \tag{10}$$

Pressure Loss In Non-Circular Pipes

The formulas for flow in pipes are based upon the use of pipes of circular cross-section. The formulas may be used with conduits of other shapes, and in conduits not flowing full, when the flow is turbulent, by using the hydraulic radius, R_H, which is really a ratio:

$$R_H = \frac{\text{area of cross-section}}{\text{wetted perimeter of cross-section}} \tag{11}$$

TABLE 1. VALUES OF e/d FOR DIFFERENT KINDS OF PIPE

TYPE OF PIPE	e/d
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

For instance, in a square duct, I ft on a side, handling air, the hydraulic radius is ¼ or 0.25. If the same duct is handling water, flowing 9 in. deep, the hydraulic radius is 0.75/2.5 or 0.30. Note in this latter case that the wetted perimeter does not include the distance across the free surface.

In the case of a round pipe

$$R_H = \frac{\pi d^2/4}{\pi d} = \frac{d}{4} \text{ or } d = 4R_H \tag{12}$$

Substituting Equation 12 in Equation 6,

$$N_{Re} = \frac{4R_H V\rho}{\mu} \tag{13}$$

and in the flow Equation 5,

$$h_f = \frac{flV^2}{8gR_H} \tag{14}$$

and finally in the compressible fluid flow Equation 10,

$$p_1 - p_2 = p_1\left[1 - \sqrt{1 - \frac{flV_1^2}{4g\ R_H\ p_1 v_1}}\right] \tag{15}$$

Equations 13, 14, and 15 may be used to compute the flow in pipes and ducts of non-circular section and in any type of conduit not flowing full. *They should not be used when the flow is laminar.*

FLOW OF COMPRESSIBLE FLUIDS

The energy equation for the flow of compressible fluids, as represented by the gases, is derived from Equation 1. Assuming that no heat is transferred to the fluid, that no work is done, and that there is no difference in elevation, Equation 1 becomes

$$\frac{V_1^2}{2g} + Ju_1 + p_1 v_1 = \frac{V_2^2}{2g} + Ju_2 + p_2 v_2 \tag{16}$$

or, after rearrangement,

$$\frac{V_2^2 - V_1^2}{2g} = p_1 v_1 - p_2 v_2 + J(u_1 - u_2) \tag{17}$$

Since internal energy is dependent only on temperature,

$$u_1 - u_2 = c_v(T_1 - T_2) \tag{18}$$

where

c_v = the specific heat of the gas at constant volume.
T_1 and T_2 = the temperatures in Fahrenheit degrees at points 1 and 2, respectively.

Substituting Equation 18 in Equation 17,

$$\frac{V_2^2 - V_1^2}{2g} = p_1 v_1 - p_2 v_2 + Jc_v(T_1 - T_2) \tag{19}$$

Now

$$c_v = \frac{R}{J(k-1)} \tag{20}$$

where

R = the gas constant in the expression.

$$pv = RT \tag{21}$$

k = the ratio of the specific heat at constant pressure to the specific heat at constant volume.

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

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 per cent	1.28 to 1.32
Sulfur dioxide, ethylene, acetylene	1.24 to 1.26

Substituting Equations 20 and 21 in Equation 19, the energy equation becomes

$$\frac{V_2^2 - V_1^2}{2g} = \frac{k}{k-1}(p_1v_1 - p_2v_2) \tag{22}$$

While this is a convenient form of equation, it does not include all the necessary specifications. If the steady flow process is frictionless and reversible,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^k = \left(\frac{\rho_2}{\rho_1}\right)^k \tag{23}$$

By introducing this relation in Equation 22 it is possible to reduce that equation to:

$$\frac{V_2^2 - V_1^2}{2g} = \frac{k}{k-1}\frac{p_1}{\rho_1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right] \tag{24}$$

This form of the equation is applicable not only to flow in pipes, but also to flow through orifices and nozzles.

A significant factor in the flow of compressible fluids is the velocity of sound, V_{so}, which for present purposes will be considered as that velocity under the conditions at the entrance to the steady flow system. The velocity of sound is expressed as

$$V_{so} = \sqrt{\frac{gkp_1}{\rho_1}} \tag{25}$$

This formula may be developed rationally and agrees perfectly with experimental results. Substituting Equation 25 in Equation 24:

$$\frac{V_2^2 - V_1^2}{2} = \frac{V_{so}^2}{k-1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right] \tag{26}$$

or, by rearrangement,

$$\frac{p_2}{p_1} = \left[1 - \frac{k-1}{V_{so}^2}\left(\frac{V_2^2 - V_1^2}{2}\right)\right]^{\frac{k}{k-1}} \tag{27}$$

which permits the calculation of the ratio of pressures at entrance and exit of the steady flow device,—pipe, orifice, or nozzle.

FLOW THROUGH NOZZLE OR ORIFICE

Another useful expression, covering the energy change in an orifice or nozzle, may be derived from Equation 2. As with the flow through pipes, no outside work is done. Then, assuming that there is no difference in elevation, and since practically no heat is evolved or absorbed, *i.e.*, the process is adiabatic, E, z, and q of Equation 2 may be eliminated, and, by rearranging, the equation becomes

$$\frac{V_2^2}{2g} - \frac{V_1^2}{2g} = J(h_1 - h_2) \text{ foot pounds per second} \tag{28}$$

In any flow device,

$$\frac{V_1A_1}{v_1} = \frac{V_2A_2}{v_2} \text{ or, } V_1 = V_2\frac{A_2v_1}{A_1v_2} \tag{29}$$

in which A_1 or A_2 is the cross-sectional area of the flow at a particular point, expressed in square feet. With this substituted in Equation 28, and solving for V_2:

$$V_2 = \sqrt{\frac{2gJ(h_1 - h_2)}{1 - (A_2/A_1)^2(v_1/v_2)^2}} \tag{30}$$

Using this expression, it is possible to determine the velocity at any point in the flow through an orifice or nozzle. If the area at the point of entry is very large with respect to that at point 2, the denominator on the right side of Equation 30 will approach unity, and the equation will reduce to

$$V_2 = \sqrt{2gJ(h_1 - h_2)} \tag{31}$$

For this reason, the expression $\sqrt{\frac{1}{1 - (A_2/A_1)^2(v_1^2/v_2^2)}}$ is called the correction factor for the velocity of approach.

The velocity of approach factor may be further simplified if the difference in volume between points 1 and 2 is negligible. Under this condition, the velocity of approach factor becomes $\sqrt{\frac{1}{1 - (A_2/A_1)^2}}$. If

$$\frac{A_2}{A_1} = \frac{D_2^2}{D_1^2} = \beta^2 \tag{32}$$

the velocity of approach factor is $\sqrt{\frac{1}{1 - \beta^4}}$, in which form it is generally used in flow formulas. The quantity β is the ratio of the throat or orifice diameter to the pipe diameter.

The connection of the velocity of sound with the flow of fluids has already been noted. Its most important application is to the flow of gases through a converging tube or nozzle. If it is assumed that the inlet velocity of the fluid, V_1, is negligible, Equation 24 will reduce to

$$V_2 = \sqrt{\left(\frac{2gk}{k-1}\right)\frac{p_1}{v_1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]} \tag{33}$$

Let W represent the weight of gas flowing through the converging tube in a unit of time, and A_2 the area at the throat; then,

$$W = A_2V_2/v_2 \text{ or } V_2 = Wv_2/A_2 \tag{34}$$

Substituting this, as well as the relation $p_1v_1^k = p_2v_2^k$, in Equation 33 gives,

$$W = A_2\sqrt{\left(\frac{2gk}{k-1}\right)\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]} \tag{35}$$

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 formula:

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

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

Critical Pressure and Critical Flow

Actually, the broken part of the curve is not attained for the flow in the nozzle. If the ratio of p_2 to p_1 is decreased from unity, the weight

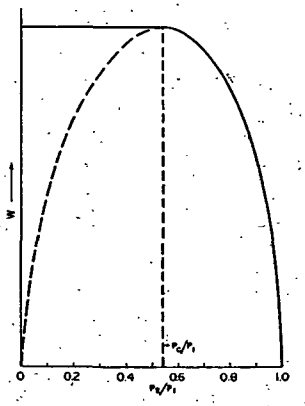


FIG. 6. RELATION OF FLOW OF GAS TO PRESSURE DROP IN A CONVERGING TUBE

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 from Equation 27 it is seen that p_c is approximately 53 per cent of p_1 when air is flowing.

To find the velocity at the critical pressure, let us assume that the upstream velocity V_1 in Equation 22 is so small as to be negligible. Using the subscript c to indicate conditions at the critical point, we have

$$\frac{V_c^2}{2g} = \left(\frac{k}{k-1}\right)(p_1v_1 - p_cv_c) \quad or,$$

$$V_c = \sqrt{\left(\frac{2gk}{k-1}\right)(p_1v_1 - p_cv_c)} \tag{37}$$

Substituting Equations 23 and 36, and rearranging, Equation 37 becomes

$$V_c = \sqrt{\left(\frac{2gk}{k+1}\right)p_1v_1} \tag{38}$$

and

$$V_c = \sqrt{\frac{gkp_c}{\rho_c}} \tag{39}$$

Comparing Equation 39 with Equation 25, it will be seen that the velocity at the throat is equal to the velocity of sound at the critical pressure.

Critical flow is attained only in converging tubes, in nozzles, and in orifices with a well-rounded approach. It does not occur in sharp-edged orifices or in nozzles having an expanding outlet section. The so-called critical flow prover uses this property of constant rate of flow above the critical pressure, and finds application as a flow regulator and a quantity-rate meter; in either case, the theoretical rate of flow may be computed from Equation 38, multiplying V_c by the area of the constriction to obtain the volume rate of flow.

In developing the working equations for orifices and nozzles, it is customary to start with

$$V_2^2 - V_1^2 = 2gh_f \tag{40}$$

This may be derived from the Bernoulli equation or from the relations of falling bodies. Now, since

$$A_1V_1 = A_2V_2 = Q_s \tag{41}$$

in which Q_s is the discharge rate in cubic feet per second,

$$\frac{Q_s^2}{A_2^2} - \frac{Q_s^2}{A_1^2} = 2gh_f \tag{42}$$

Transposing,

$$Q_s = \frac{A_1A_2}{\sqrt{A_1^2 - A_2^2}}\sqrt{2gh_f} \tag{43}$$

or

$$Q_s = A_2 \frac{1}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}} \sqrt{2gh_f} \qquad (44)$$

the central term on the right hand side of Equation 44 will be recognized as the *velocity of approach factor*, so that

$$Q_s = A_2 \frac{1}{\sqrt{1 - \beta^4}} \sqrt{2gh_f} \qquad (45)$$

Actual Flow Through Orifices and Nozzles

The actual rate of flow through an orifice, nozzle, or Venturi tube is rarely equal to the theoretical, and generally the actual rate is less than the theoretical. In the case of the nozzle and Venturi tube, this is due to losses from wall roughness, fluid friction, and turbulence during the expansion in the section following the throat. While wall roughness is not a factor in a sharp-edged orifice, fluid friction and turbulence are important, as is the fact that the discharge contracts to a degree variable with the ratio of outlet to inlet pressure after leaving the orifice, so that the limiting area is somewhat less than the opening in the orifice plate. Accordingly, Equation 45 must be modified by a correction factor, C. Usually, the velocity of approach factor is included with this correction factor, and, if

$$K = C \frac{1}{\sqrt{1 - \beta^4}}, \qquad (46)$$

$$Q_s = KA_2 \sqrt{2gh_f}. \qquad (47)$$

Multiplying by 3600 to convert from cubic feet per second to cubic feet per hour and converting area in square feet to diameter in inches, gives

$$Q_f = 3600\, K \frac{\pi D_2^2}{4 \times 144} \sqrt{2gh_f}$$

or

$$Q_f = 19.635\, KD_2^2 \sqrt{2gh_f} \qquad (48)$$

where

Q_f = rate of flow in cubic feet per hour.
D_2 = the diameter of the orifice or nozzle throat in inches.

Equation 48 is a general equation, expressing the flow of any fluid through an orifice or nozzle. Further use of it will be made as other types of flow are discussed.

The differential loss, h_f, is in terms of feet of the fluid flowing through the orifice or nozzle. In the case of a gas flowing, where it is customary to read the differential pressure in inches of water, feet of gas must be converted to inches of water. Since dry air at 32 F and 14.7 psi absolute pressure weighs 0.0807 lb per cubic foot, the weight of a cubic foot of any other kind of gas under the same conditions is 0.0807 G, where G is the specific gravity of the gas referred to air. Water weighs 62.37 lb per cubic foot at 60 F. Using also the relation of 12 in. in 1 ft,

$$h_f = \frac{h_w}{12} \times \frac{62.37}{0.0807G} \qquad (49)$$

in which h_w is the differential pressure in inches of water.

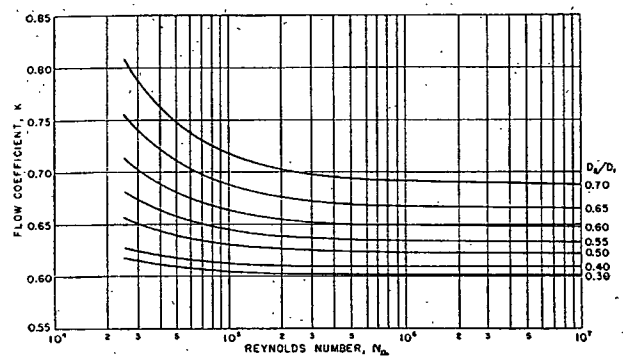


FIG. 7. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND FLANGE TAPS IN SMOOTH PIPE
NOTE: From Table 6, Bibliography [H].

Also, since the gas flowing is not necessarily at 32 F and 14.7 psi, it is necessary to apply Charles' and Boyle's laws to the density of the gas and therefore:

$$h_f = \frac{h_w}{12} \times \frac{62.37}{0.0807G} \times \frac{14.7}{P_f} \times \frac{T_f}{492} \qquad (50)$$

in which P_f and T_f are the absolute pressure and temperature of the flowing gas. Substituting this in Equation 48, and combining the constants,

$$Q_f = 218.44KD_2^2 \sqrt{\frac{h_w T_f}{P_f G}} \qquad (51)$$

Then, to correct the value of Q_f to any other standard conditions of pressure and temperature, using the gas laws,

$$Q_b = Q_f \times \frac{P_f}{P_b} \times \frac{T_b}{T_f} \qquad (52)$$

Equation 51 becomes

$$Q_b = 218.44KD_2^2 \frac{T_b}{P_b} \sqrt{\frac{P_f h_w}{T_f G}} \qquad (53)$$

Finally, since gases expand under the conditions of reduced pressure downstream from the orifice or nozzle, an expansion factor, Y, must be added. The final formula, then, is

$$Q_b = 218.44KYD_2^2 \frac{T_b}{P_b} \sqrt{\frac{P_f h_w}{T_f G}} \qquad (54)$$

In Equation 54, all the data must be observed at the time of measurement except K and Y. These must be obtained from charts, tables, or formulas, derived from or based on the results of a great many experiments, the results of which have been collected by a joint committee of the *American Gas Association* and the *American Society of Mechanical Engineers*[2,3]. The report of the two associations gives orifice coefficients as a function of the Reynolds number and of the ratio of orifice to pipe diameter, for pipes from 2 to 12 and 14 in. in diameter, and for four

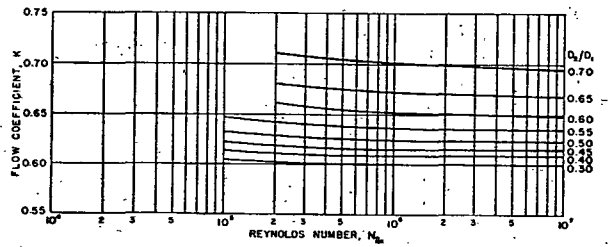


FIG. 8. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND RADIUS TAPS IN SMOOTH PIPE
NOTE: From Fig. 36d, Bibliography [K].

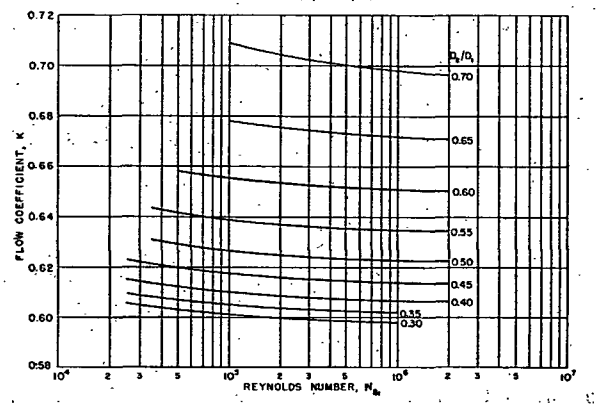


FIG. 9. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND VENA CONTRACTA TAPS, IN SMOOTH PIPE
NOTE: From Table 7, Bibliography [H].

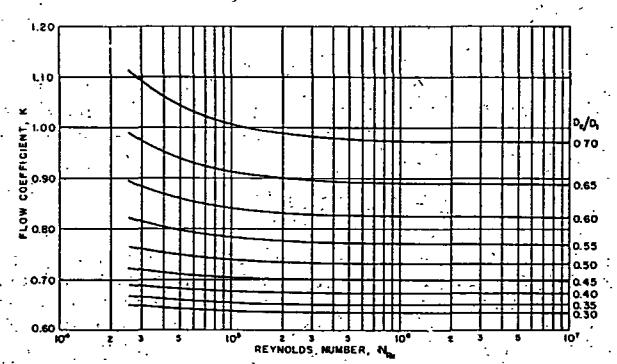


FIG. 10. FLOW COEFFICIENTS, K, FOR SQUARE-EDGED ORIFICE PLATES AND PIPE TAPS IN SMOOTH PIPE
NOTE: From Table 8, Bibliography [H].

different types of pressure taps in use in the United States. The coefficients are higher for the smaller pipe sizes. This is an effect of the turbulence produced by the roughness of the pipe surface, a given roughness being relatively greater with a small pipe than with a large one.

Space does not permit presenting all the coefficient data that are available. However, if the pipe is smooth drawn tubing, the effect of roughness is negligible, and the coefficients for the largest size of pipe apply also to smaller pipes. Figs. 7, 8, 9, and 10 show these coefficients, N_{Re}, being the Reynolds number referred to the diameter of the orifice or throat of the nozzle, in feet.

Pressure Taps–Location and Types

The different sets of pressure taps are called flange taps, radius taps,

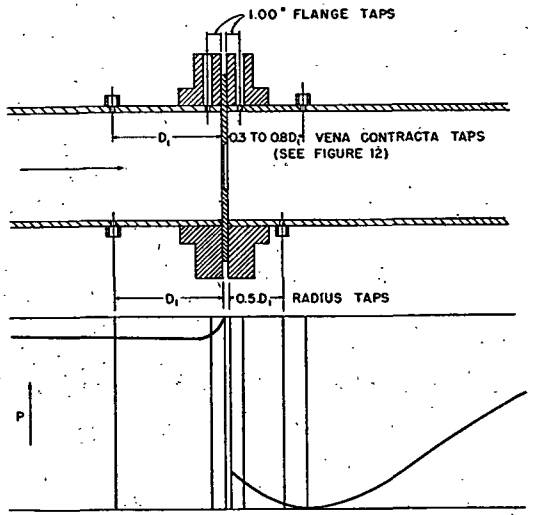


FIG. 11. RELATIVE LOCATION OF FLANGE, RADIUS AND VENA CONTRACTA TAPS

vena contracta taps, and pipe or full-flow taps. The relative locations of the first three of these are shown in Fig. 11, and the need for different coefficients for the different taps is indicated by the course of the change in pressure of the flowing fluid shown in the lower part of the figure. Pipe taps are located 2½ pipe diameters upstream and 8 pipe diameters downstream, both measured from the upstream face of the orifice plate, or in other words, before the orifice plate has had any effect on the flow and after the recovery in pressure has been completed.

Still another type of pressure tap is used in European practice,—corner taps. 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.

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, varies with the ratio of orifice to pipe diameter, and with rate of flow, as shown in Fig. 12; the tap is generally located in accordance with the mean curve in the figure.

Expansion Factor for Gases

The expansion factor, Y, for gases (for liquids, $Y = 1$) is found from the empirical formula

$$Y = 1 - (0.41 + 0.35\beta^4)\left(\frac{p_1 - p_2}{p_1}/k\right) \qquad (55)$$

This is applicable to flange, radius, and vena contracta taps. For pipe taps, the formula is

$$Y = 1 - \left[0.333 + 1.145\,(\beta^2 + 0.7\beta^5 + 12\beta^{13})\right]\left[\frac{p_1 - p_2}{p_1}/k\right] \qquad (56)$$

Values of Y for air, computed from these equations, are given in Fig. 13 and Fig. 14.

Computing Orifice Discharge

With this information it is possible to compute the discharge from an orifice if the Reynolds number is known. Here an odd complication is encountered—when the value of N_{Re} is computed, the rate of flow, which is the unknown quantity, must be used in the computation. However, it will be noted in Figs. 7, 8, 9, and 10 that the orifice coefficient does not change greatly as N_{Re} changes. If, then, an estimate is made of the velocity, using this in computing N_{Re}, and if the corresponding coefficient is used in Equation 54, a value for the rate of flow will be found. Using this velocity to compute a corrected value of N_{Re} and repeating the process, a more nearly correct value of Q_f is found. This cut-and-try method may be continued for several more cycles, but generally the first or second correction will be found sufficient.

Another method would be to use the value of K corresponding to $N_{Re} = \infty$, modifying this with a factor involving the rate of flow, determined from the temperature, and the differential and static pressures. This method is used by the *American Gas Association*[3].

STEAM FLOW MEASUREMENT

While steam may be considered as a gas, its measurement differs from that of the usual gases because of a number of factors. Equation 48 serves as the starting point. Since it is usual to measure the differential pressure in inches of water, it is necessary to convert h_f, the head in feet in terms of the flowing fluid, to h_{aw}, the actual head in inches of water, using the equation

$$h_f = \frac{h_{aw}}{12}\,\frac{\rho_w}{\rho} \qquad (57)$$

where

ρ_w = the density of water at 60 F (62.37 lb per cubic foot).
ρ = the density of the flowing fluid.

Substituting Equation 57 in Equation 48 gives

$$Q_f = 359.15\,KD_2^2\sqrt{\frac{h_{aw}}{\rho}} \qquad (58)$$

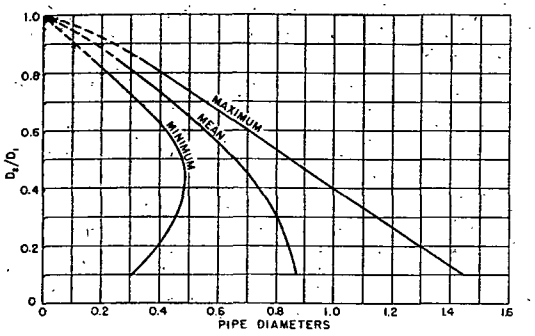


FIG. 12. LOCATION OF VENA CONTRACTA IN RELATION TO RATIO OF ORIFICE TO PIPE DIAMETER AND TO RATE OF FLOW

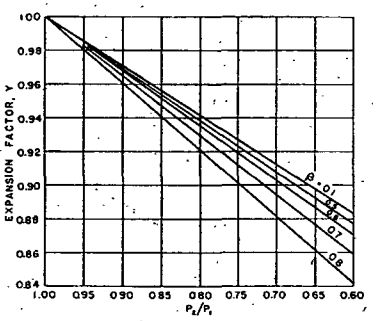


FIG. 13. EXPANSION FACTOR FOR AIR AND OTHER DIATOMIC GASES APPLICABLE TO FLANGE, RADIUS AND VENA CONTRACTA TAPS

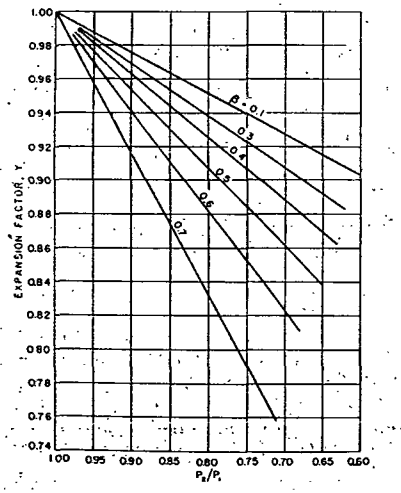


FIG. 14. EXPANSION FACTOR FOR AIR AND OTHER DIATOMIC GASES APPLICABLE TO PIPE TAPS

In steam measurement, a constant head of water is maintained over each leg of the manometer by means of condensing chambers, in order to keep the heat of the steam away from the meter. As the mercury level fluctuates, the difference in head as recorded on the chart, therefore, is not that due to mercury alone, but to mercury minus an equivalent head of water. To correct for this, the equation

$$h_{aw} = h_w \frac{12.557}{13.557} \tag{59}$$

is applied to Equation 58. The denominator in Equation 59 is the specific gravity of mercury; and the numerator is the difference in specific gravity between mercury and water. Equation 58, hence, becomes

$$Q_f = 345.65\, KD_2^2 \sqrt{\frac{h_w}{\rho}} \tag{60}$$

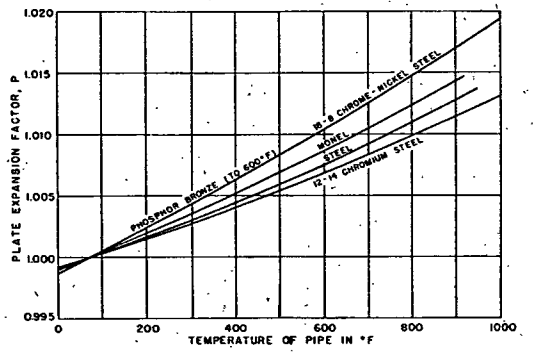


FIG. 15. VARIATION OF ORIFICE PLATE EXPANSION FACTOR, P, WITH TEMPERATURE AND MATERIAL

Steam is commonly measured in terms of weight, and since

$$W = \rho Q_f \tag{61}$$

where

W = the rate of flow in pounds per hour.

$$W = 345.65\, KD_2^2 \sqrt{h_w \rho} \tag{62}$$

The expansion factor Y, and the factor P, correcting for the expansion of the orifice plate with the temperature must then be applied, so that

$$W = 345.65\, KYPD_2^2 \sqrt{h_w \rho} \tag{63}$$

which is the *final form of the equation for the flow of steam through orifices.* Values of K may be obtained from Figs. 7, 8, 9, and 10, according to the pressure taps used. Y may be computed from Equations 55 or 56. Fig. 15 gives values of the correction factor P, according to the temperature and the material of the orifice plate. Values of the density, ρ, may be obtained from steam tables, such as Keenan and Keyes[4], which are widely used.

METERING LIQUIDS

Orifices are also used for metering liquids, and Equation 48 serves again as a starting point for developing the working formula. Again, too, it is necessary to convert h_f to h_{aw}, the actual head in inches of water, by substituting Equation 57. It is also necessary to correct for the weight of the fluid above the manometer, and since this may be other than water, it is better to use an equation of more general form than Equation 59:

$$h_{aw} = h_w \frac{13.557 - \frac{\rho_f}{\rho_w}}{13.557} \tag{64}$$

where

ρ_f = the density of the fluid over the mercury in the manometer.

ρ_w = the density of water at 60 F.

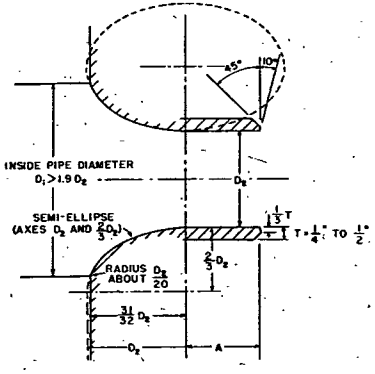


FIG. 16. SHAPE OF *ASME* LONG RADIUS NOZZLE WHEN RATIO OF THROAT TO PIPE DIAMETER IS 0.53 OR LESS

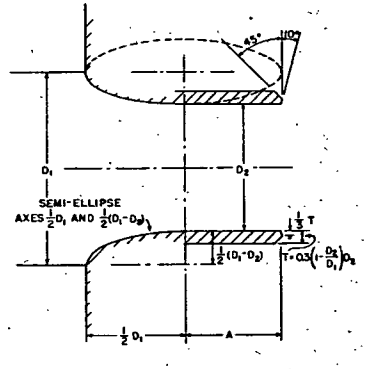


FIG. 17. SHAPE OF *ASME* LONG RADIUS NOZZLE WHEN RATIO OF THROAT TO PIPE DIAMETER IS 0.4 TO 0.7

Substituting Equations 57 and 64 in Equation 48 gives

$$Q_f = 44.764\, KD_2^2 \sqrt{h_w \left(\frac{1}{\rho_f} - 0.00118\right)} \tag{65}$$

Then, since liquids are generally measured in gallons instead of cubic feet, and since there are 7.4805 gal in 1 cu ft,

$$Q_w = 334.86\, KD_2^2 \sqrt{h_w \left(\frac{1}{\rho_f} - 0.00118\right)} \tag{66}$$

in which Q_w is the discharge or rate of flow, in gallons per hour.

Since liquids, for practical purposes, are incompressible, no expansion factor is necessary. If circumstances demand, the factor P for the expansion of the orifice may be applied. Also, if it is necessary to correct the volumetric discharge to a base temperature, application of the known expansion characteristics of the liquid will enable the conversion. Values

of K again are obtainable from Figs. 7, 8, 9, and 10, according to the type of pressure tap.

NOZZLE COEFFICIENTS AND EXPANSION FACTORS

Nozzles differ from orifices in that the flow is guided to the throat in such a way that contraction of the jet is suppressed, or, in other words, there is no vena contracta. Because of this fact, the coefficients are different from those of orifices, and very close to unity before the velocity of approach factor is added. Also, the expansion factor may be deduced rationally, rather than empirically, as with orifices.

Two shapes of nozzles that have been under investigation by the *A.S.M.E.* for some time are shown in Figs. 16 and 17. They are referred

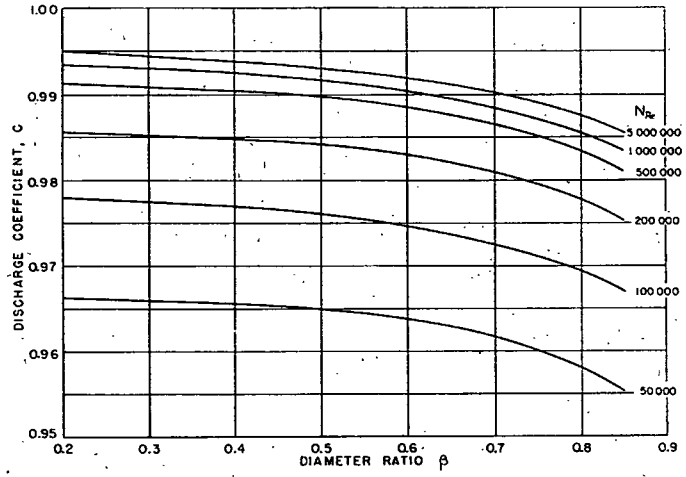


FIG. 18. RELATION OF NOZZLE DISCHARGE COEFFICIENT, C, FOR 2-INCH PIPE, TO DIAMETER RATIO AND REYNOLDS NUMBER

to as long-radius nozzles. Their contour is that of a semi-ellipse, and the contracting portion is followed by a cylindrical section of the same area as the throat. The shape shown in Fig. 16 is designed for use with ratios of throat to pipe diameter of 0.53 or less, that of Fig. 17 for ratios of 0.4 to 0.7. The most usual location of pressure taps is 1 pipe diameter upstream and ½ diameter downstream, both measured from the plane of the nozzle inlet. In addition, the *International Standards Association* has adopted still another shape of nozzle, which has a rather sharper approach than the *A.S.M.E.* nozzles, and which uses corner taps. Very little use of this nozzle has been made in this country.

The formulas already given for orifices apply equally to nozzles except for discharge coefficients, and for the expansion factor, when it is applied. Discharge coefficients for nozzles, as for orifices, vary with pipe size; they may either increase or decrease with decreasing size of pipes depending on the sharpness of the approach curvature of the nozzle. For the *A.S.M.E.* nozzles, they tend to decrease. Generally speaking, too, the

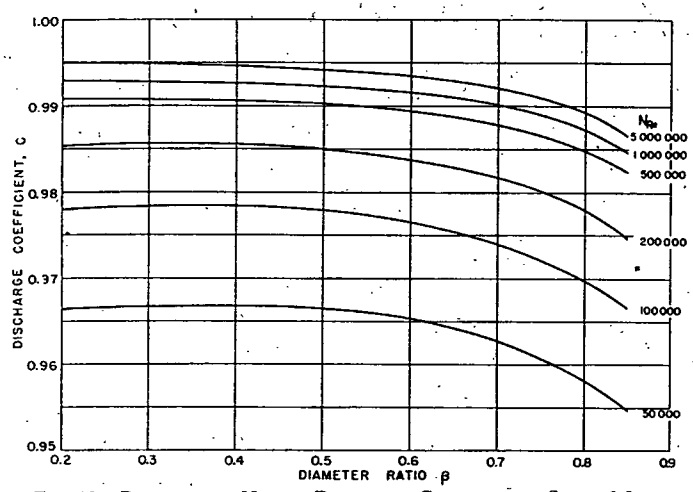


FIG. 19. RELATION OF NOZZLE DISCHARGE COEFFICIENT, C, FOR 6-INCH PIPE, TO DIAMETER RATIO AND REYNOLDS NUMBER

coefficient for a given nozzle shape is higher if the finish of the surface is smoother.

Discharge coefficients, C, for pipes 2, 6, and 10 in. in diameter are given in Figs. 18, 19, and 20, as correlated by Bean, Beitler and Sprenkle[5], as functions of the diameter ratio β and the Reynolds number N_{Re} referred to the diameter of the throat in feet.

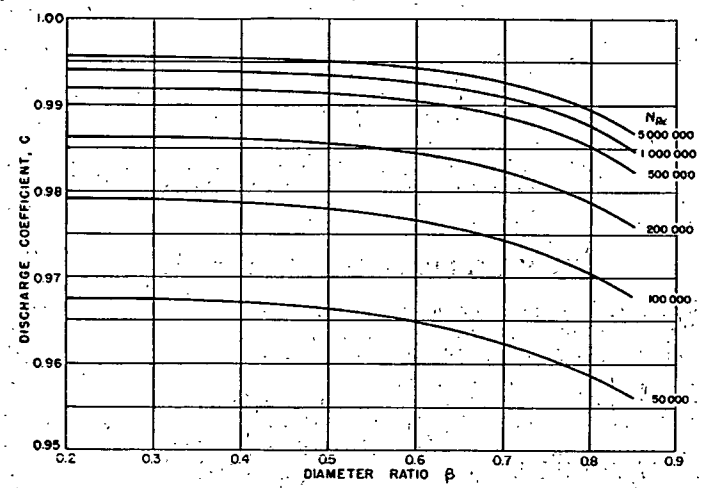


FIG. 20. RELATION OF NOZZLE DISCHARGE COEFFICIENT, C, FOR 10-INCH PIPE, TO DIAMETER RATIO AND REYNOLDS NUMBER

$$N_{Re} = \frac{D_2}{12}\,\frac{V_2\rho_1}{\mu_1} \qquad (67)$$

Coefficients from these curves must be multiplied by $\frac{1}{\sqrt{1-\beta^4}}$, the velocity of approach factor, in accordance with Equation 46, to obtain the value of K to use in the various equations.

The expansion factor for nozzles, designated by φ, is obtained from a rational formula, as already noted.

$$\varphi = \sqrt{\left(\frac{p_2}{p_1}\right)^{2/k}\left(\frac{k}{k-1}\right)\left(\frac{1-(p_2/p_1)^{k-1/k}}{1-p_2/p_1}\right)\left(\frac{1-\beta^4}{1-\beta^4(p_2/p_1)^{2/k}}\right)} \qquad (68)$$

This formula is plotted for $k = 1.40$ (air and other diatomic gases) and $k = 1.30$ (steam, carbon dioxide, natural gas) in Figs. 21 and 22, respectively.

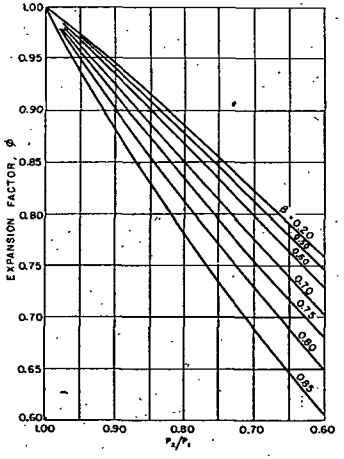


FIG. 21. RELATION OF EXPANSION FACTOR, φ, FOR NOZZLES TO DIAMETER RATIO AND PRESSURE LOSS FOR AIR AND OTHER DIATOMIC GASES

FLOW MEASUREMENT BY PITOT TUBE

There remains one other head type meter useful in ventilating work, the Pitot tube, named for the Frenchman who discovered the principle. The Pitot tube is essentially a bent tube with its open end pointed upstream, combined with another tube with its end pointed crosswise to the flow or downstream, or connected to openings crosswise to the flow. Used with flowing liquid, the liquid will rise in each tube, but higher in the one pointed upstream. Used with a flowing gas, and the two tubes connected by a U-tube containing water, the liquid level in the U-tube will be displaced, with the lower level on the side connected to the tube pointed upstream. The tube directed upstream receives the impact

pressure, which is the sum of the static and kinetic pressures, while the tube directed crosswise receives only the static pressure; the difference between the two, as read on the separate tubes or on the U-tube is, of course, the kinetic pressure. The velocity is expressed as

$$V = \sqrt{2gh_t} \qquad (69)$$

Application of Equation 57 serves to make the formula general, assuming that water is used in the manometer connecting the two tubes. Using this equation, and mutiplying by 60 to convert feet per second to feet per minute,

$$V_m = 1096.5\sqrt{\frac{h_{aw}}{\rho}} \qquad (70)$$

in which V_m is the rate of flow in feet per minute.

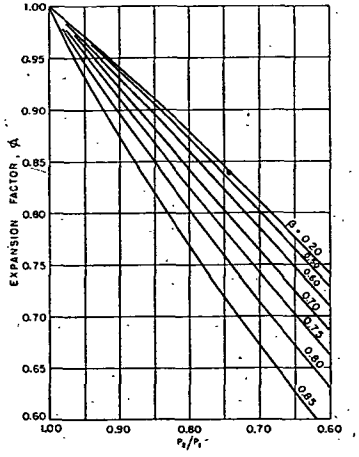


FIG. 22. RELATION OF EXPANSION FACTOR, φ, FOR NOZZLES TO DIAMETER RATIO AND PRESSURE LOSS FOR STEAM, CARBON DIOXIDE AND NATURAL GAS

It is often difficult to obtain the exact static pressure. In the usual construction of Pitot tubes, the static pressure openings are downstream from the impact pressure opening, and turbulence induced by the nose may affect the static pressure reading. If the static pressure openings point downstream in any degree, a suction effect is produced to falsify the reading. In instruments having the static pressure opening pointed downstream, the coefficient may be as low as 0.86. Consequently, for accurate work, Pitot tubes should be calibrated, and the pertinent coefficient should be applied to Equation 69. In a sense, this coefficient is advantageous, since it results in a higher differential reading, which, in turn, enables more accurate readings at low flows.

In using Pitot tubes, it is generally necessary to make a traverse of the pipe or duct to determine the course of the velocity pattern. In a pipe, for instance, one of the profiles shown in Fig. 5 would be obtained. Near a valve or fitting, however, the profile might be quite distorted, a

fact which would be revealed by the traverse. If the pipe or duct is divided into equal areas, and a determination of h_{aw} is made for each, the average velocity would be obtained by using the average of the square roots of h_{aw} in Equation 70.

INSTALLATION OF NOZZLES AND ORIFICES

A final note should be made of the installation of orifices and nozzles. Generally speaking, the orifice or nozzle, together with a holding arrangement, including pressure taps, is available from the manufacturer. In making the installation, the user should make certain that the flow

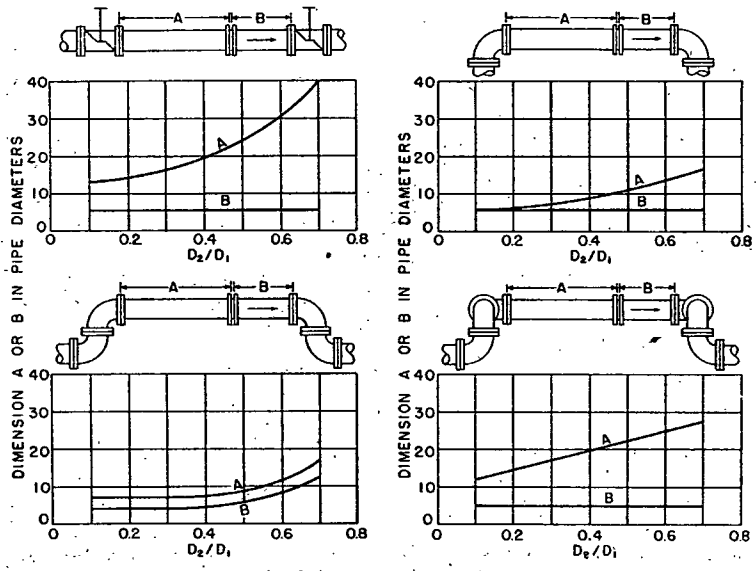


FIG. 23. MINIMUM CONDITIONS TO BE OBSERVED WHEN INSTALLING ORIFICES AND NOZZLES BETWEEN FITTINGS AND VALVES

approaching the nozzle or orifice is steady and evenly distributed, *i.e.*, with velocity profiles similar to those shown in Fig. 5. Fittings and valves, which tend to direct the flow to one side and which in some cases cause it to rotate as it advances, must be far enough upstream from the orifice or nozzle to permit the disturbed stream to straighten out to the even form before reaching the meter. Minimum conditions in the installation for avoiding trouble from fittings and valves are shown in Fig. 23. If necessary, straightening vanes may be used upstream from the orifice or nozzle at a distance of not less than 8 pipe diameters.

The information given in this chapter is of necessity an abridgment. It is strongly recommended that anyone contemplating accurate measurement should refer to the references and bibliography given at the end of this chapter.

LETTER SYMBOLS USED IN CHAPTER 4

β = ratio, throat or orifice diameter to pipe diameter.
μ = absolute viscosity in pounds per foot second.
μ/ρ = kinematic viscosity in square feet per second.
ρ = density of flowing fluid in pounds per cubic foot.
ρ_w = density of water at 60 F, (62.37 lb per cubic foot).
ρ_f = density of fluid over mercury in a manometer.
φ = expansion factor for nozzles.
A = cross-sectional area of flow, in 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 in feet.
d = internal diameter of pipe in feet.
E = mechanical work in foot pounds per pound of fluid flowing.
e = absolute roughness of pipe surface, in feet.
f = dimensionless friction coefficient.
g = acceleration due to gravity = 32.17 ft per (second) (second).
G = specific gravity of gas referred to air.
h = enthalpy, Btu per pound of fluid.
h_{aw} = loss of head in inches of water.
h_f = loss of head in feet of fluid.
h_t = total head in feet of fluid.
h_w = differential pressure in inches of water.
J = mechanical equivalent of heat = 778 foot pounds per Btu.
K = flow coefficient (correction factor), including velocity of approach, 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 in feet.
l = length of pipe in feet.
N_{Re} = Reynolds number.
P = correction factor for expansion of orifice plate with temperature.
p = pressure in pounds per square foot.
p_c = critical pressure.
P_b = standard pressure to which correction is to be made, pounds per square inch, absolute.
P_f = pressure of gas flowing, pounds per square inch, absolute.
Q_b = rate of flow in cubic feet per hour under standard conditions of pressure and temperature.
Q_f = rate of flow in cubic feet per hour.
Q_s = discharge rate in cubic feet per second.
Q_w = rate of flow in gallons per hour.
q = heat transferred to the fluid per pound of fluid flowing.
R = gas constant.
R_H = hydraulic radius = ratio of area of cross-section to wetted perimeter of cross-section.
r = radius of pipe in feet.
T = temperature, Fahrenheit degrees, absolute.
T_b = standard temperature to which correction is to be made, Fahrenheit degrees absolute.
T_f = temperature of gas flowing, Fahrenheit degrees, absolute.
u = internal energy, Btu per pound of fluid.
V = velocity in feet per second.
V_c = critical velocity, feet per second.
V_{so} = velocity of sound, feet per second.

V_m = velocity in feet per minute.
v = specific volume, cubic feet per pound.
W = weight of gas flowing, pounds per hour.
Y = expansion factor—correcting for expansion of gas under reduced downstream pressure.
z = elevation above some arbitrary datum, in feet.

REFERENCES

1—Friction Factors for Pipe Flow, by Lewis F. Moody (*A.S.M.E. Transactions, 66*, 1944, 671-678; Discussion, idem. *66*, 1944, 678-684).

2—History of Orifice Meters and the Calibration, Construction, and Operation of Orifices for Metering. Report of the Joint *A.G.A.—A.S.M.E.* Committee on Orifice Coefficients (*American Society of Mechanical Engineers*, 1935).

3—Gas Measurement Committee Report No. 2, Natural Gas Department (*American Gas Association*, 1935).

4—Thermodynamic Properties of Steam, by Joseph H. Keenan and Frederick G. Keyes (John Wiley & Sons, Inc., 1936).

5—Discharge Coefficients of Long Radius Flow Nozzles When Used with Pipe Wall Pressure Taps, by H. S. Bean, S. R. Beitler, and R. E. Sprenkle (*A.S.M.E. Transactions, 63*, 1941, 439-442; Discussion, idem. *63*, 1941, 442-445).

BIBLIOGRAPHY

[A] Principles of Thermodynamics, by G. A. Goodenough (Henry Holt & Co.).

[B] Principles of Engineering Thermodynamics, by Paul J. Kiefer and Milton C. Stuart (John Wiley & Sons, Inc., 1944).

[C] Fluid Mechanics, by Russell A. Dodge and Milton J. Thompson (McGraw-Hill Book Co., 1937).

[D] Fluid Mechanics, by R. C. Binder (Prentice-Hall, Inc., 1943).

[E] The Physics of Solids and Fluids, by P. P. Ewald, H. Pöschl and L. Prandtl (Blackie, 1936).

[F] A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler, Hydraulic Paper HYD-55-2 (*A.S.M.E. Transactions 55*, No. 10, 7-22, 1933; Discussion, idem., *55*, No. 10, 23-32, 1933).

[G] The Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering 55*, 1933, 497-501, 515).

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

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

[J] The Orifice Meter for Measurement of Flow of Gases and Liquids, by Allen D. MacLean (Pittsburgh Equitable Meter Co., 1938).

[K] Flow Measurement (*American Society of Mechanical Engineers*, 1940).

[L] Pitot Tube Practice, by Edward S. Cole (*A.S.M.E. Transactions 57*, 1935, 281-294; Discussion, idem. *58*, 1936, 146-156).

[M] Pitot Tubes in Large Pipes, by Edward S. Cole and E. Shaw Cole (*A.S.M.E. Transactions, 61*, 1939, 465-473; Discussion, idem. *61*, 1939, 473-475).

[N] Investigation of Errors of Pitot Tubes, by C. W. Hubbard (*A.S.M.E. Transactions, 61*, 477-497; Discussion, idem. *61*, 1939, 497-506).

[O] Piping Arrangements for Acceptable Flowmeter Accuracy, by R. E. Sprenkle (*A.S.M.E. Transactions, 67*, 345-357, 1945; Discussion, idem. *67*, 357-360, 1945).

CHAPTER 5

Fundamentals of Heat Transfer

Conduction, Convection, Radiation, Combined Convection and Radiation, Heat-Flow Resistance, Electrical Analogies, Practical Heat Transfer Problems, Unit Conductances for Convection Flow Systems, Radiation Factors or Emissivities, Solutions for Steady-State Conduction Problems

HEAT is that form of energy which is transferred from place to place 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 simple term and known as the resistance. Energy exchange associated with mass transfer from place to place (evaporation, condensation, etc.) due to concentration differences is treated elsewhere such as in the section on *cooling tower design* in Chapter 37. 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.

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

CONDUCTION, CONVECTION AND RADIATION

Thermal conduction in fluids is the term applied to the mechanism of heat transfer whereby the molecules of higher random kinetic energies transmit by direct molecular collision part of their energy to adjacent molecules of lower random kinetic energy. Since the temperature is proportional to the random kinetic energy of the molecules, thermal transfer will occur in the direction of decreasing temperature. The molecules oscillate about a mean position at fairly high velocities and frequencies, but there is no net material flow associated with the conduction mechanism. In the case of fluids, thermal conduction is significant in the region very close to a solid boundary or wall, for in this region the flow is laminar, parallel with the wall surface, and there are practically no cross currents in the direction of the heat transfer across the solid fluid boundary. In solid bodies the significant mechanism of heat transport is always thermal conduction and is ascribed to an energy transfer mechanism associated with the free electrons [1].

Contrasted to the thermal conduction mechanism, *thermal convection* involves energy transfer by eddy mixing and diffusion [2] in addition to conduction. This condition is pictured 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 slightly 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. But 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 induced by sources external to the heat transfer region, as for example a pump, the described solid to fluid heat transfer is termed *forced convection*. In contrast, if the fluid currents are internally generated, 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 heat is transferred as internal energy, *i.e.*, the random molecular kinetic energy associated with the material temperature. 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. Since radiant energy exhibits characteristic wave lengths, the

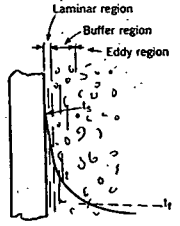


FIG. 1. THERMAL CONVECTION CONDITIONS

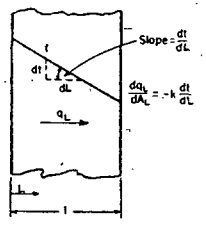


FIG. 2. THERMAL CONDUCTION IN A FLAT SLAB

solution of thermal radiation problems is in many respects similar to the solution of problems in the field of illumination.

The rate of thermal current flow (*i.e.*, 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 difference.

Thermal Conduction Equation

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

$$\frac{dq}{dA} = -k\frac{dt}{dL} \tag{1}$$

The minus sign on the right side of the equation is introduced to indicate positive current flow in the direction of decreasing temperature. Fig. 2 shows the physical significance of 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, when curved or spherical walls are considered, considerable complication is involved. Therefore, in this discussion the consistent units of conductivity expressed in *Btu per (hour) (square foot) (degrees Fahrenheit per one foot thickness)* are used throughout. *Conductivity values obtained from Chapter 6 or Table 1 in this chapter, which are expressed in inconsistent units, must therefore be converted for use in the calculations of this chapter by dividing by 12.* As an example, the conductivity of brick, expressed in inconsistent units as 5.0 in Table 2 of Chapter 6, becomes 0.42 when

TABLE 1. APPROXIMATE UNIT THERMAL CONDUCTIVITIES.[a]

Conductivity, k = Btu per (hr) (sq ft) (deg F 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 Heat Transmission, by W. H. McAdams (McGraw-Hill Co., 1942) for additional values.

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.*

Thermal Convection Equation

$$\frac{dq}{dA} = h_c\,(t_s - t_f) \tag{2}$$

This rate equation states that the thermal convection current per unit transfer area $(dq)/(dA)$, 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 convection conductance* (sometimes called the film coefficient for convection), h_c, Btu per (hour) (square foot) (degree Fahrenheit). These convection conditions are illustrated in Fig. 1.

The heat transmission by free or natural convection for objects surrounded by air can be conveniently expressed as in Equation 2a:

$$q_c = C\left(\frac{1}{D}\right)^{0.3}\left(\frac{1}{T_{av}}\right)^{0.181}(t_s - t_f)^{1.27} \tag{2a}$$

where

q_c = heat transmission by convection, Btu per (square foot) (hour).
C = a constant depending upon the surface shape.

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, degrees Fahrenheit absolute.

$t_s - t_f$ = temperature excess between wall surface and surrounding air, degrees Fahrenheit.

For horizontal cylinders, the value of $C = 1.016$ has been well established by various investigations. For vertical plates, the value of $C = 1.394$ 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.

TABLE 2. HEAT TRANSMISSION BY FREE CONVECTION FOR LARGE VERTICAL SURFACES

Expressed in Btu per square foot per hour

Temp Deg F	Temperature Difference between Body and Surrounding Still Air at 80 F													
	0	10	20	30	40	50	60	70	80	90	100	110	120	130
0	0	4.4	10.4	17.4	25.0	33.2	41.8	50.6	59.9	69.4	79.4	89.2	99.4	109.8
1	0.3	4.9	11.1	18.1	25.8	34.1	42.6	51.5	60.8	70.3	80.4	90.2	100.4	110.9
2	0.6	5.5	11.8	18.9	26.7	34.9	43.5	52.4	61.8	71.3	81.4	91.2	101.5	112.0
3	1.0	6.0	12.5	19.7	27.5	35.7	44.3	53.4	62.7	72.3	82.4	92.2	102.6	113.0
4	1.4	6.6	13.2	20.5	28.3	36.6	45.2	54.3	63.7	73.3	83.3	93.3	103.6	114.1
5	1.8	7.3	13.9	21.2	29.2	37.4	46.1	55.2	64.6	74.3	84.2	94.3	104.7	115.2
6	2.3	7.9	14.6	22.0	30.0	38.3	47.0	56.1	65.6	75.3	85.2	95.3	105.7	116.3
7	2.8	8.5	15.3	22.7	30.8	39.1	47.8	57.1	66.5	76.3	86.2	96.3	106.7	117.3
8	3.3	9.1	16.0	23.5	31.6	40.0	48.7	58.0	67.5	77.4	87.2	97.4	107.8	118.4
9	3.8	9.7	16.7	24.3	32.4	40.9	49.7	59.0	68.4	78.4	88.2	98.4	108.8	119.5

The heat transmission by free convection from vertical walls *24 in. or more in height* is given in Table 2 as calculated from Equation 2a for an ambient air temperature of 80 F. The values in Table 2 are not changed appreciably by a considerable change in air temperature for a given temperature excess. For instance, a change in air temperature from 80 to 40 F will increase the heat transmission given in Table 2 by only 1.3 per cent.

Table 2 can also be used for calculating the free convection rate of transmission for various commercial shapes such as pipes and ducts. These calculations are simplified by the use of the factors in Tables 3 and 4. Table 3 gives factors by which the values in Table 2 must be multiplied to obtain the free-convective transfer from various shapes *whose characteristic dimensions are 24 in. or over*, and Table 4 gives the factors to be used in conjunction with the factors in Table 3 for obtaining the free convection from Table 2 for pipes and ducts *whose characteristic dimensions are less than 24 in.*

For example, the free convection transfer from a 3 in. O.D. horizontal cylinder for a temperature difference of 40 F = 25.0 × 0.73 × 1.52 = 27.7 Btu per (square foot) (hour).

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:

$$q_c = h_c A (t_1 - t_2) \qquad (2b)$$

where

q_c = heat transmission by convection, Btu per hour.

A = surface area, square feet.

$t_1 - t_2$ = temperature difference between the surface and the fluid degrees Fahrenheit.

h_c = unit conductance, from Table 5, Btu per (square foot) (hour) (degree Fahrenheit temperature difference.)

TABLE 3. FREE CONVECTION FACTORS FOR VARIOUS SHAPES

Shapes	Factor
Horizontal cylinders 24 in. in diam. or over	0.73
Long vertical cylinders 24 in. in diam. or over	0.88
Vertical plates 24 in. in height or over	1.00
Horizontal plates warmer than air facing upward	1.28
Horizontal plates warmer than air facing downward	0.64
Horizontal plates cooler than air facing upward	0.64
Horizontal plates cooler than air facing downward	1.28

TABLE 4. FREE CONVECTION FACTORS FOR VARIOUS DIAMETER PIPES OR VARIOUS HEIGHT PLATES

Actual O.D., or height, in.	1	2	3	4	5	6	7	8
Factor	1.88	1.64	1.52	1.43	1.37	1.32	1.28	1.25
Actual O.D., or height, in.	9	10	12	14	16	18	20	22
Factor	1.22	1.19	1.15	1.11	1.09	1.06	1.04	1.02

Thermal Radiation Equation

The relation shown in Equation 3 is usually applicable to systems in which radiant exchange takes place between the surfaces of solids, as sche-

$$q_r = \sigma A_1 F_A F_E (T_1^4 - T_2^4) \qquad (3)$$

matically shown in Fig. 3. Gaseous and luminous radiation are not considered in this discussion. Equation 3 states that the net radiation current 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:

σ = the Stefan-Boltzmann radiation constant.

= 1730 × 10^{-12} Btu per (hour) (square foot) (degree Fahrenheit absolute temperature to the fourth power).

F_A = the configuration factor 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 is also dimensionless and ≤ 1. This factor accounts for the absorption and emission characteristics of the surfaces for the radiation which exists. Individual emissivities (ϵ) should be taken from Table 6 and applied, for either radiation or absorption, as follows:

a. For a small body in a large enclosure, use the emissivity of the small body only: $F_E = \epsilon_1$.

b. For rectangles or disks, either parallel or perpendicular and with a common side, use the product of the emissivities: $F_E = \epsilon_1 \times \epsilon_2$.

c. For large parallel planes, long concentric cylinders or large enclosed bodies, use both emissivities in the equation:

$$F_E = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}$$

The radiation under black-body conditions, or for an emissivity of 1.0, is given in Table 7[4] for cold surfaces as low as −39 F to warmer surfaces as high as 139 F. The emissivities of a number of surfaces ordinarily encountered in engineering practice are shown in Table 6. For radiation table at higher temperatures, and further discussion of radiation calculations, see Chapter 31.

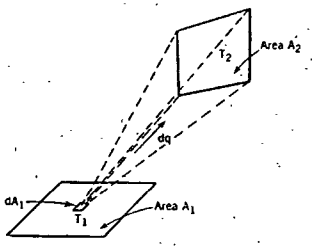


FIG. 3. RADIATION BETWEEN SURFACES

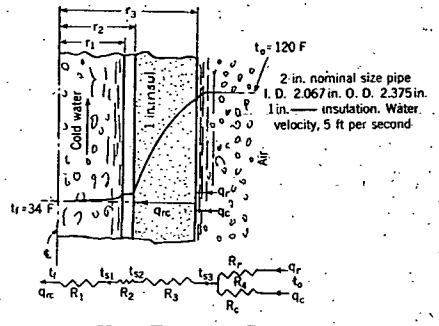


FIG. 4. HEAT TRANSFER CONDITIONS IN THE INSULATED COLD WATER LINE

NOMENCLATURE AND DIMENSIONS FOR TABLE 5

c_p = fluid unit heat capacity at constant pressure, Btu per (pound) (degree Fahrenheit).
D = cylinder diameter, feet.
$G = 3600\ V_s\rho$ = fluid mass velocity, pounds per (hour) (square foot of flow cross-section).
ρ = density, pounds per cubic foot.
h_c = unit conductance for thermal convection, Btu per (hour) (square foot) (degree Fahrenheit).
k = unit thermal conductivity of the fluid, Btu per (hour) (square foot) (degree Fahrenheit per one foot thickness).
R_H = hydraulic radius of the flow cross-section = flow cross-section area per wetted perimeter, feet.
s = fin spacing, feet.
t = average fluid film temperature, degree Fahrenheit.
$t_1 - t_2$ = temperature difference surface to main fluid, degree Fahrenheit.
V_s = fluid velocity, feet per second.
μ = fluid viscosity, pounds per (hour) (foot) = viscosity in centipoises × 2.42.

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

TABLE 5. APPROXIMATE UNIT CONDUCTANCES FOR THERMAL CONVECTION FOR SEVERAL FLOW SYSTEMS[a]

Expressed in Convenient Empirical Form

CASE	SYSTEM	UNIT CONDUCTANCE EQUATION[b]
	FORCED CONVECTION	
1.	Longitudinal flow in cylinders, turbulent region. Fluid being heated[c].	$\frac{h_c D}{k} = 0.0225 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{0.4}$ For $\left(\frac{DG}{\mu}\right) > 3000$
2.	For longitudinal air flow in cylinders case 1 reduces to[c].	$h_c = 0.0036\ G^{0.8}/D^{0.2}$ For $\left(\frac{DG}{\mu}\right) > 3000$
3.	For longitudinal water flow in cylinders case 1 reduces to[c].	$h_c = 0.00486\ (1 + 0.01t)\ \frac{G^{0.8}}{D^{0.2}}$ For $\left(\frac{DG}{\mu}\right) > 3000$
4.	Air flow normal to a single right circular cylinder.	$h_c = 0.45 \left(\frac{k}{D}\right) + 0.178\ G^{0.56} \left(\frac{k}{D}\right)^{0.44}$
5.	Air flow over staggered pipe banks.	$h_c = 0.061 \left(\frac{k}{D}\right)^{0.31} G^{0.69}$
6.	Air flow over single spheres.	$h_c = 0.040\ \frac{G^{0.53}}{D^{0.48}}$ $0 < t < 250$ F
7.	Air flow over plane surfaces.	$h_c = 1 + 0.22\ V_s$ For $V_s < 16$ fps or $h_c = 0.53\ V_s^{0.8}$ 16 fps $< V_s <$ 100 fps
8.	Air flow normal to finned cylinders.	$h_c = 6.2 \left(\frac{G}{3600}\right)^{0.6} \frac{s^{0.33}}{D^{0.52}}$ $0 < t < 250$ F
	FREE CONVECTION[d]	
9.	Single horizontal right circular cylinder in air.	$h_c = 0.23 \left(\frac{t_1 - t_2}{D}\right)^{0.25}$
10.	Vertical surfaces in air.	$h_c = 0.3\ (t_1 - t_2)^{0.25}$
11.	Top surface of horizontal plates to air.	$h_c = 0.4\ (t_1 - t_2)^{0.25}$
12.	Bottom surface of horizontal plates to air.	$h_c = 0.2\ (t_1 - t_2)^{0.25}$

[a] Heat Transmission, by W. H. McAdams.

[b] Fluid properties should be evaluated at the arithmetic mean fluid temperature, $t_f = (t_{surface} + t_{fluid})$ divided by 2.

[c] These expressions are applicable to longitudinal flow in other than right circular cylinders provided the hydraulic radius is employed as the conduit dimension parameter. For non-circular cross-sections $D = 4\ R_H$.

[d] For low rates of heat transfer by free convection the exponent decreases towards zero, and for higher rates increases towards 0.33. The following equations employing an exponent equal to 0.25 are applicable in the intermediate range.

practical cases it is desirable to treat convection and radiation as a single combined process, using a first-power equation:

$$q_{rc} = h_{rc} A (t_1 - t_2) \quad (4)$$

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

TABLE 6. RADIATION FACTORS OR EMISSIVITIES, ϵ,
For the determination of factor F_R in Equation 3

CLASS	SURFACES	FRACTION OF BLACK-BODY RADIATION		
		At 50–100 F	At 1000 F	Solar Radiation
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	0.3 to 0.5
6	Window glass	0.90 to 0.95	.....	Transparent
7	Bright aluminum paint; gilt or bronze paint	0.4 to 0.6	.....	0.3 to 0.5
8	Dull brass, copper, or aluminum; galvanized steel; polished iron	0.2 to 0.3	0.3 to 0.5	0.4 to 0.65
9	Polished brass, copper, monel metal	0.02 to 0.05	0.05 to 0.15	0.3 to 0.5
10	Highly polished aluminum, tin plate, nickel, chromium	0.02 to 0.04	0.05 to 0.10	0.10 to 0.40

radiation and convection, are given in Chapter 6, Table 1 and Fig. 3. Complete tables for the combined heat transfer of steam and hot water radiators, pipes, coverings, etc., will be found in the appropriate chapters.

When dealing with the effect of operating temperatures upon the combined heat transfer of a given piece of equipment (as for instance a steam radiator), another form of equation is frequently used:

$$q_{rc} = B A (t_1 - t_2)^n \quad (5)$$

Values of n in this equation usually range from 1.3 to 1.5 (see Chapter 25). The chief advantage of this equation is the convenience of representing heat transfer performance on logarithmic coordinates, and the factor B should be regarded as a simple constant of proportionality.

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

TABLE 7. HEAT TRANSMISSION BY RADIATION FOR BLACK-BODY CONDITIONS[a]
Expressed in Btu per (square foot) (hour)

TEMP DEG F	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	97.2	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

[a] *Example:* Radiation from walls of room at 32 F to surface at − 25 F for effective emissivity of 0.95 = (102 − 62.3).0.95 = 37.7 Btu per (square foot) (hour).

potential or temperature difference, and inversely proportional to the thermal resistance:

$$q_{rc} = \frac{t_1 - t_2}{R} \quad (6)$$

Following the electrical analogy, when there is a thermal current flowing through several *resistances in series*, the resistances are additive:

$$R_T = R_1 + R_2 + R_3 + \ldots + R_n \quad (7)$$

Similarly, conductance is the reciprocal of resistance, and for heat flow through several *resistances in parallel*, the conductances are additive:

$$C_T = \frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \ldots + \frac{1}{R_n} \quad (8)$$

Practical Heat Transfer Problems

The use of these simple relations for resistance and conductance simplifies many practical heat transfer problems. As discussed in Chapters 6, 7 and 28, 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 8 indicates the solutions 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. 4.

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 current by radiation and convection then passes through the insulating layer and the pipe wall by thermal conduction, and thence by convection 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. 4, the thermal current 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 degrees Fahrenheit per Btu, is the summation of the individual resistances:

$$R_T = R_1 + R_2 + R_3 + R_4 \tag{9}$$

where the resultant parallel resistance R_4 is obtained from:

$$\frac{1}{R_4} = \frac{1}{R_r} + \frac{1}{R_c}$$

Provided the individual resistances may be evaluated, the total resistance can be obtained from this relation. Then the heat transfer current for the length of pipe (N, ft) can be established by the relation:

$$q_{rc}\ (\text{Btu per hour}) = \frac{t_o - t_f}{R_T} \tag{10}$$

For a unit length of the pipe the heat transfer rate is:

$$\frac{q_{rc}}{N}\ (\text{Btu per hour foot}) = \frac{(t_o - t_f)}{R_T N} \tag{11}$$

The temperature drop, Δt, through an individual resistance may then be calculated from the relation:

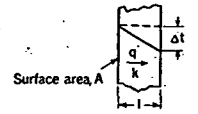
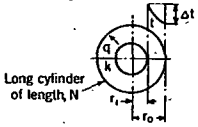
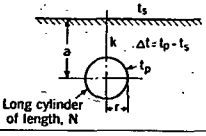
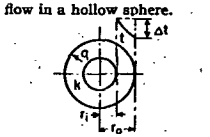
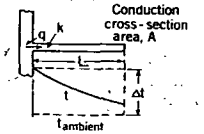
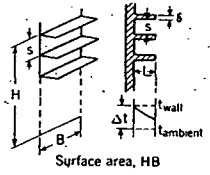
$$\Delta t = R\, q_{rc}$$

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 integration of the rate Equations 1, 2 and 3 to produce expressions of the form:

$$q = \frac{\Delta t}{R} \tag{12}$$

TABLE 8. 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. 	$R = \frac{\log_e \frac{2a}{r}}{2\pi kN}$; $R = \frac{\cosh^{-1}\frac{a}{r}}{2\pi kN}$ for $\frac{a}{r} \geq 3$ (See footnote *c*).
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 $ml > 2.3$, $\tanh m L \approx 1$ $m = \sqrt{h_s p/kA}$ 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. 	$R = \frac{(s+\delta)}{h_s\left(\frac{2}{m}\tanh\frac{m\,l+s}{}\right)HB}$ $m = \sqrt{\frac{h_s p}{kA}} = \sqrt{\frac{2\,h_s}{k\delta}}$ 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) (degree Fahrenheit for one foot thickness); units of h, Btu per (hour) (square foot) (degree Fahrenheit); units of area, A = square feet.

[b] The thermal conductivity, k, in these solutions should be taken at the average material temperature (see Table 5).

[c] $\text{Log}_e\, x = 2.303 \log_{10} x$.

[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.

where q is the heat transfer rate, and Δt is the potential drop or temperature difference through the resistance R. Table 8 lists such solutions for six different conduction systems. Table 2 in Chapter 6 and Table 1 of this chapter indicate the magnitudes of the thermal conductivities, k, to be employed in the expressions of Table 8, after dividing k by 12.

The solution applicable to the problem depicted in Fig. 4, for the calculation of R_2 and R_3, is case 2 in Table 8. 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:

$$R_2 = \frac{\log_e \frac{1.188}{1.033}}{2\pi \times 26 \times 1} = 8.5 \times 10^{-4} \text{ hr degree Fahrenheit per Btu.}$$

$$R_3 = \frac{\log_e \frac{2.188}{1.188}}{2\pi \times 0.025 \times 1} = 3.9 \text{ hr degree Fahrenheit per Btu.}$$

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_c, are dependent on the flow conditions prevailing at these surfaces, and on the thermal properties of the fluids. The unit conductances for thermal convection, h_c, Btu per (hour) (square foot) (degree Fahrenheit), have been determined by test for many flow systems. These data may be employed to predict the conductances for *similar flow systems*. Table 5 summarizes some empirical equations expressing such test results.

For the problem under consideration (Fig. 4) case 3 of Table 5 is applicable for the calculation of the cold water side convection resistance R_1. Corresponding to the water velocity of 5 fps, the mass velocity is:

$G = 5$ (ft per sec) $\times$ 62.4 (lb per cu ft) $\times$ 3600 (sec per hr) $= 11.2 \times 10^5$ lb per (hour) (square foot).

The inside diameter of the pipe D is $\frac{2.067}{12} = 0.1725$ ft.

The average water film temperature will be estimated as 36 F (mixed mean fluid temperature of 34 F). Then case 3, Table 5 yields:

$$h_c = 0.00486\,(1 + 0.36)\,\frac{(11.2 \times 10^5)^{0.8}}{(0.1725)^{0.2}} = 650 \text{ Btu per (hour) (square foot) (degree Fahrenheit).}$$

The transfer area on which this conductance is based is the inside tube area. Associated with 1 ft length of pipe there are:

$$\pi \times \frac{2.067}{12} \times 1 = 0.542 \text{ sq ft.}$$

Thus the resistance for 1 ft of tube length is:

$$R_1 = \frac{1}{h_c \pi D \times 1} = \frac{1}{650 \times 0.542} = 2.8 \times 10^{-3} \text{ hr degree Fahrenheit per Btu.}$$

Case 9, Table 5 is applicable for calculating the free thermal convection resistance, R_c, existing between the surrounding air and the insulation. The air temperature is given as 120 F. As an approximation a 20 deg

temperature difference between the air and the pipe surface will be assumed. Then case 9 yields:

$$D = \frac{4.375}{12} = 0.364 \text{ ft.}$$

$$h_c = 0.23 \left(\frac{20}{0.364}\right)^{0.25} = 0.63 \text{ Btu per (hour) (square foot) (degree Fahrenheit).} \quad (13)$$

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 mass velocity corresponds to:

$$G = 7 \times 0.07 \times 3600 = 1770 \text{ lb air per (hour) (square foot).}$$

A magnitude of $k = 0.014$ Btu per (hour) (square foot) (degree Fahrenheit) per one foot thickness applied to case 4 yields:

$$h_c = 0.45 \left(\frac{0.014}{0.364}\right) + 0.178\,(1770)^{0.56} \left(\frac{0.014}{0.364}\right)^{0.44}$$

$$= 0.017 + 2.8 = 2.8 \text{ Btu per (hour) (square foot) (degree Fahrenheit).}$$

This conductance is based on 1 sq ft of outside lagging area. Thus, since there are $\pi \times \frac{4.375}{12} = 1.14$ sq ft of outside lagging area associated with 1 ft length of pipe:

$$R_c = \frac{1}{2.8 \times 1.14} = 0.312 \text{ hr degree Fahrenheit per Btu.}$$

The radiation resistance, R_r, which acts in parallel with the convection resistance, R_c, for the transfer of heat to the surface of the insulation, may be calculated. For the purposes of this illustrative problem it will be assumed that the insulated pipe is exposed to (*sees*) surroundings, which exist at 120 F. Then the angle factor, F_A, is unity and for an estimated surface emissivity of 0.9 (see Table 6), $F_e = 0.9$. As a first approximation the insulation surface temperature will be estimated as 20 deg below the surroundings at 120 F. Then the radiation per degree of temperature difference, by Equation 3 (or more conveniently by Table 8) divided by the temperature difference will be:

$$h_r = \frac{(196 - 170)\,0.95}{20} = 1.17 \text{ Btu per (hour) (square foot) (degree Fahrenheit).}$$

The outside surface area of the insulation associated with 1 ft of pipe length was previously calculated as 1.14 sq ft. Thus:

$$R_r = \frac{1}{1.17 \times 1.14} = 0.75 \text{ hr degree Fahrenheit per Btu.}$$

The resultant resistance of R_c and R_r acting in parallel (see Fig. 4) can now be evaluated as:

$$\frac{1}{R_4} = \frac{1}{R_c} + \frac{1}{R_r} = \frac{1}{0.312} + \frac{1}{0.75} = 4.54 \text{ Btu per (hour) (degree Fahrenheit).}$$

$$R_4 = 0.22 \text{ hr degree Fahrenheit per Btu.}$$

The over-all resistance, R_T, surroundings to cold water, is the sum of $R_1 + R_2 + R_3 + R_4 = 4.1$ hr degree Fahrenheit 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:

$$\frac{q_{rc}}{N} = \frac{t_o - t_f}{R_T} = \frac{120 - 34}{4.1} = 21 \text{ Btu per (hour) (foot)}$$

or about 0.175 tons of refrigeration per 100 ft of pipe.

Since the calculation is based on a 1 ft pipe length:

$$q_{rc} = 21 \text{ Btu per hour.}$$

The temperature drops through the various resistances are now readily evaluated by Equation 12 as:

$t_o - t_{s3}$ air to insulation surface $= R_4 q_{rc} = 0.22 \times 21 = 4.6$ F.
$t_{s3} - t_{s2}$ through the insulation $= R_3 q_{rc} = 3.9 \times 21 = 82$ F.
$t_{s2} - t_{s1}$ through the pipe wall $= R_2 q_{rc} = 8.5 \times 10^{-4} \times 21 = 0.02$ F.
$t_{s1} - t_f$ pipe wall to cold water $= R_1 q_{rc} = 2.8 \times 10^{-3} \times 21 = 0.06$ F.

The solution was obtained on the assumption that the air temperature and the outside temperature differed by 20 F. In order to obtain a slightly better estimate of the rate of heat transfer the numerical solution should be repeated using the temperatures calculated from the previous listed temperature differences.

The foregoing problem serves to illustrate a general method of solving steady-state heat transfer problems. There are many problems which cannot be approximated by steady-state solutions. For instance, the problem of pipe line insulation in transient service; the behavior of automatically controlled thermoflow circuits; or the periodic absorption of solar energy by roof and wall structures during the day and nocturnal radiation to the *cold* sky at night. The transient heat transfer problem differs from the steady-state in that energy storage rates need to be considered. Thus thermal capacity in addition to resistance effects is significant. The vector sum of the thermal capacitance and resistance is the thermal impedance. It is not within the scope of this chapter to deal with many of these problems. There are, however, solutions available in graphical form for certain special cases. Also a general approximate method may be employed which is analogous to the treatment of capacity-resistance lumped parameter electrical circuits.

REFERENCES

1—The Metallic State, by H. Hume-Rothery (Oxford Press, 1931).

2—Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., 1937).

3—The Transmission of Heat by Radiation and Convection, by Griffith and Davis (*Special Report* No. 9, 1922, *Department of Scientific and Industrial Research*, His Majesty's Stationery Office, London, England).

4—Heat Insulation in Air Conditioning, by R. H. Heilman (*Industrial and Engineering Chemistry*, Vol. 28, July, 1936, p. 782).

CHAPTER 6

Heat Transmission Coefficients of Building Materials

Heat Transfer Symbols; Calculating Over-all Coefficients; Conductivity of Homogeneous Materials; Surface Conductance; Air Space Conductance; Practical Coefficients and Their Use; Computed Heat Transmission Coefficients; Roof Coefficients; Combined Ceiling and Roof Coefficients; Basement Floor, Basement Wall, and Concrete Slab Floor Coefficients, Condensation in Buildings

THE design of conditioning or heating systems for buildings requires a knowledge of the thermal properties of the walls enclosing the space. 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 *over-all coefficient of heat transmission*. 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 testing of all combinations of building materials is impracticable, the procedure and necessary data for calculation of the value of U are given in this chapter, together with tables of computed values for the more common constructions.

HEAT TRANSFER SYMBOLS

U = over-all coefficient of heat transmission (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 in construction and also to single materials, such as window glass, and includes the surface conductance on both sides.

k = thermal conductivity; the time rate of heat flow through a homogeneous material under steady conditions through unit area per unit temperature gradient in the direction perpendicular to the area. Its value is expressed in Btu per (hour) (square foot) (Fahrenheit degree per inch). 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 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 which may be either homogeneous or heterogeneous.

f = film or surface conductance; the time rate of heat flow between 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 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 and the character of the boundary surfaces. The relationships are not linear and accurate values must be obtained by test and not by computation.

R = thermal resistance. Its value is expressed in Fahrenheit degrees per (Btu) (hour) (square foot). It may represent any of the following and must therefore be properly described:

$\frac{1}{U}$ = over-all or air-to-air resistance

$\frac{1}{k}$ = resistance per unit thickness (resistivity)

$\frac{1}{C}$ = resistance of a material (surface-to-surface)

$\frac{1}{f}$ = film or surface resistance

$\frac{1}{a}$ = air space resistance

CALCULATING OVER-ALL 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. Then by definition,

$$U = \frac{1}{R_t} = \frac{1}{R_1 + R_2 + R_3 + \cdots + R_n} \quad (1)$$

where

R_1, R_2, etc. are the individual resistances of the wall components.

R_t = total resistance.

For a wall of a single homogeneous material of conductivity k and thickness x, with surface coefficients f_i and f_o

$$U = \frac{1}{R_t} = \frac{1}{\frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o}} \quad (2)$$

For a compound wall of three homogeneous materials in series, having conductivities k_1, k_2 and k_3 and thicknesses x_1, x_2 and x_3 respectively, and laid together without air spaces,

$$U = \frac{1}{R_t} = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o}} \quad (3)$$

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, and separated by an air space of conductance a

$$U = \frac{1}{R_t} = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o}} \quad (4)$$

In the case of 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 $\frac{1}{C}$ is therefore substituted for $\frac{x}{k}$ in Equations 2, 3 and 4.

CONDUCTIVITIES AND CONDUCTANCES

The method of calculating the over-all 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, which require modification of the

formula. In such cases calculated results should be checked by test measurements.

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.

Tables 1 and 2 give conductivities and conductances which are quite generally used in calculation and which have been selected from various sources. Wherever possible the properties of the material and test conditions are given. In selecting and applying heat transmission values to any construction, caution is necessary, because coefficients for the same material may differ because of variations which occur in test methods, in the materials themselves, or in the temperature of the material when tested.

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 air spaces and reflective types, are of a porous nature and consist of combinations of solid matter with small air cells. The thermal conductivity of these materials will vary with density, mean temperature, size of 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. It will be noted that for each there is an optimum density for lowest conductivity. Typical variation of conductivity with mean temperature is shown in Fig. 2.

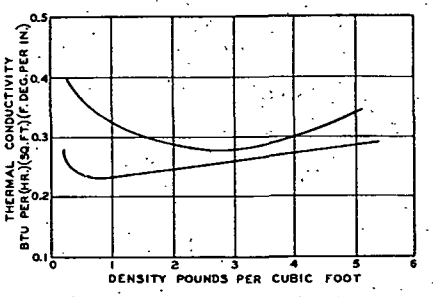


FIG. 1. TYPICAL VARIATION OF THERMAL CONDUCTIVITY WITH DENSITY—FOR FIBROUS MATERIAL

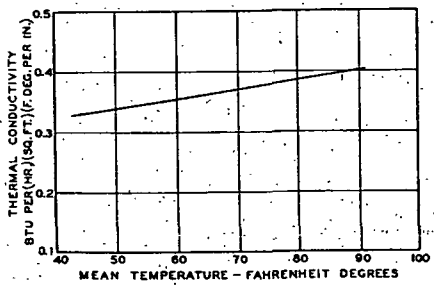


FIG. 2. TYPICAL VARIATION OF THERMAL CONDUCTIVITY WITH MEAN TEMPERATURE

TABLE 1. CONDUCTANCES (C) FOR SURFACES AND AIR SPACES

All conductance values expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference).

Section A. Surface Conductances for Still Air[a]

POSITION OF SURFACE	DIRECTION OF HEAT FLOW	SURFACE EMISSIVITY	
		$e = 0.83$	$e = 0.05$
Horizontal	Upward	1.95	1.16
Horizontal	Downward	1.21	0.44
Vertical		1.52[*]	0.74

Section B. Conductance of Vertical Spaces at Various Mean Temperatures[b]

MEAN TEMP FAHR DEG	CONDUCTANCES OF AIR SPACES FOR VARIOUS WIDTHS IN INCHES						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

Section C. Conductances and Resistances of Air Spaces Faced on One Surface with Reflective Insulation[c]

LOCATION AND POSITION OF AIR SPACE	DIRECTION OF HEAT FLOW	TEMP[d] DIFF FAHR DEG		CONDUCTANCE[e] (C)			RESISTANCE[e] $\left(\frac{1}{C}\right)$		
		Winter	Summer	No. of Air Spaces			No. of Air Spaces		
				1	2	3	1	2	3
Rafter Space (8 in.)									
Horizontal	Down	45			0.10	0.07		10.00	14.29
Horizontal	Up	45			0.27	0.17		3.70	5.88
Horizontal	Down		25		0.09	0.06		11.11	16.67
Horizontal	Up		25		0.24	0.16		4.17	6.25
30 deg slope	Down	45			0.15	0.10		6.67	10.00
30 deg slope	Up	45			0.25	0.17		4.00	5.88
30 deg slope	Down		25		0.13	0.09		7.69	11.11
30 deg slope	Up		25		0.23	0.14		4.35	7.14
Stud Space (3½ in.)									
Vertical[f]		30		0.34			2.94		
Vertical		40			0.23	0.13		4.35	7.69
Vertical[f]			15	0.32			3.13		
Vertical			20		0.18	0.11		5.56	9.09
Vertical[g]		30		0.46			2.17		

[a] Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 513).

[b] A.S.H.V.E. Research Report No. 825—Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol 35, 1929, p. 165).

[c] Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E. R. Queer (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940).

[d] Temperature difference is based on total space between plaster base and sheathing, flooring or roofing.

[e] These air space conductance and resistance values are based on one reflective surface (aluminum) having an emissivity of 0.05 facing each space and are based on total space between plaster base and sheathing, flooring or roofing. The rafter and stud spaces are divided into equal spaces.

[f] Stud space is lined on plaster base side with loose paper with aluminum on surface facing air space. The resistance of the small air space between the plaster base and paper was 0.43.

[g] Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 351).

[*] The recommended surface conductance for calculating heat losses for still air for non-reflective surfaces is 1.65 Btu. For a 15 mph wind velocity, the recommended value is 6.0 Btu. These coefficients were derived from Fig. 3 which was based on tests conducted at the University of Minnesota, and apply to vertical surfaces.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS

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)	MEAN TEMP (FAHR DEG)	CONDUCTIVITY OR CONDUCTANCE		RESISTANCE		AUTHORITY
				(k)	(C)	Per Inch Thickness $\left(\frac{1}{k}\right)$	For Thickness Listed $\left(\frac{1}{C}\right)$	
BUILDING BOARDS (NON-INSULATING)	Compressed cement and asbestos sheets	123	86	2.70	....	0.37	.....	(1)
	Corrugated asbestos board	20.4	110	0.48	....	2.08	.....	(2)
	Pressed asbestos mill board	60.5	86	0.84	....	1.19	.....	(3)
	Gypsum board—gypsum between layers of heavy paper	62.8	70	1.41	....	0.71	.....	(3)
	⅜ in. gypsum board	.....	.....	.....	3.73	.....	0.27	
	½ in. gypsum board	.....	.....	.....	2.82	.....	0.35	
	½ in. gypsum board	53.5	90	.....	2.60	.....	0.38	(1)
FRAME CONSTRUCTION COMBINATIONS	1 in. fir sheathing and building paper	.....	30	.....	0.86	.....	1.16	(4)
	1 in. fir sheathing, building paper and yellow pine lap siding	.....	20	.....	0.50	.....	2.00	(4)
	1 in. fir sheathing, building paper and stucco	.....	20	.....	0.82	.....	1.22	(4)
	Pine lap siding and building paper, siding 4 in. wide	.....	16	.....	0.85	.....	1.18	(4)
	Yellow pine lap siding	.....	....	.....	1.28	.....	0.78	(4)
MASONRY MATERIALS BRICK	Damp or wet	.....	....	5.0[*]	....	0.20	.....	(2)
	Common yellow clay brick[*]	.....	....	4.8	....	0.21	.....	(4)
	One tier yellow common clay brick, one tier face brick, approx. 8 in. thick[*]	.....	....	.....	0.77	.....	1.30	(4)
CLAY TILE, HOLLOW	2 in. Tile, ½ in. plaster both sides	120.0	110	.....	1.00	.....	1.00	(2)
	4 in. Tile, ½ in. plaster both sides	127.0	100	.....	0.60	.....	1.67	(2)
	6 in. Tile, ½ in. plaster both sides	124.3	105	.....	0.47	.....	2.13	(2)
	8 in. Tile, average of 8 types (Walls No. 59, 63, 64, 66, 67, 90, 91, 92[*])	.....	....	.....	0.82	.....	1.92	(4)
	12 in. Clay tile wall: 8 in. x 5 in. x 12 in. and 4 in. x 5 in. x 12 in.[*]	.....	....	.....	0.26	.....	3.84	(4)
CONCRETE	Sand and gravel aggregate, various ages and mixes	.....	....	11.35 to 16.86	....	0.09 to 0.06	.....	(5)
	Sand and gravel aggregate	142	75	12.6	....	0.08	.....	(4)
	Limestone aggregate	132	75	10.8	....	0.09	.....	(4)
	Cinder aggregate	97	75	4.9	....	0.22	.....	(4)
	Steam treated limestone slag aggregate[*]	74.6	75	2.27	....	0.44	.....	(4)
	Pumice (Mined in California) aggregate[*]	65.0	75	2.42	....	0.41	.....	(4)
	Expanded burned clay aggregate[*]	59.9	75	2.28	....	0.44	.....	(4)
	Burned clay aggregate[*]	67.1	75	2.86	....	0.35	.....	(4)
	Blast furnace slag aggregate	76.0	70	1.6	....	0.63	.....	(3)
	Expanded vermiculite aggregate	20	90	0.68	....	1.47	.....	(3)
	Expanded vermiculite aggregate	26.7	90	0.76	....	1.32	.....	(3)

AUTHORITIES:

[1] U. S. Bureau of Standards, tests based on samples submitted by manufacturers.

[2] A. C. Willard, L. C. Lichty and L. A. Harding, tests conducted at the University of Illinois.

[3] J. C. Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers.

[4] F. B. Rowley, et al, tests conducted at the University of Minnesota.

[5] A.S.H.V.E. Research Laboratory.

[6] E. A. Allcut, tests conducted at the University of Toronto.

[*] See Thermal Conductivity of Building Materials, by F. B. Rowley and A. B. Algren (University of Minnesota *Engineering Experiment Station Bulletin* No. 12).

[b] Heat Transmission Through Insulation as Affected by Orientation of Wall, by F. B. Rowley and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 331).

[c] The Effect of Convection in Ceiling Insulation, by G. B. Wilkes and L. R. Vianey (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 196).

[d] See A.S.H.V.E. RESEARCH REPORT No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).

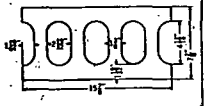
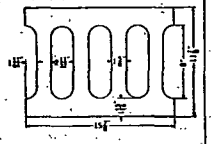
[e] See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

[f] See BMS13, U. S. Department of Commerce, National Bureau of Standards, Washington, D. C.

[g] Roofing, 0.15 in. thick (1.34 lb per square foot), covered with gravel (0.83 lb per square foot), combined thickness assumed 0.25.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS—Continued

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)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
MASONRY MATERIALS—(Continued)								
Concrete—(Continued)	Expanded vermiculite aggregate	35	90	0.86	....	1.16	.....	(3)
	Expanded vermiculite aggregate	50	90	1.10	....	0.91	.....	(3)
	Concrete plank	76	75	2.5	....	0.40	.....	(3)
	Cellular concrete	40.0	75	1.06	....	0.94	.....	(3)
	Cellular concrete	50.0	75	1.44	....	0.69	.....	(3)
	Cellular concrete	60.0	75	1.80	....	0.56	.....	(3)
	Cellular concrete	70.0	75	2.18	....	0.46	.....	(3)
8 In. Concrete Blocks 	8 in. three oval core, sand and gravel aggregate[a]	126.4	40	.....	0.90	.....	1.11	(4)
	8 in. three oval core, crushed limestone aggregate[a]	134.3	40	.....	0.86	.....	1.16	(4)
	8 in. three oval core, cinder aggregate[a]	86.2	40	.....	0.58	.....	1.73	(4)
	8 in. three oval core, burned clay aggregate[a]	67.7	40	.....	0.50	.....	2.00	(4)
	8 in. three oval core, expanded blast furnace slag aggregate[a]	.....	40	.....	0.49	.....	2.04	(4)
12 In. Concrete Blocks 	12 in. three oval core, sand and gravel aggregate[a]	124.9	40	.....	0.78	.....	1.28	(4)
	12 in. three oval core, cinder aggregate[a]	86.2	40	.....	0.53	.....	1.88	(4)
	12 in. three oval core, burned clay aggregate[a]	76.7	40	.....	0.47	.....	2.13	(4)
Gypsum	3 in. solid gypsum partition tile[a]	.....	....	2.41	....	0.42	.....	(4)
	3 in. three cell gypsum partition tile[a]	.....	....	.....	0.74	.....	1.35	(4)
	4 in. three cell gypsum partition tile[a]	.....	....	.....	0.60	.....	1.67	(4)
	87½ per cent gypsum, 12½ per cent wood chips	51.2	74	1.66	....	0.60	.....	(4)
	Gypsum plaster	.....	....	3.30	....	0.30	.....	
PLASTERING MATERIALS	Gypsum plaster, ⅝ in. thick	.....	73	.....	8.80	.....	0.11	(4)
	Cement plaster	.....	....	8.00	....	0.13	.....	(2)
	Wood, lath and plaster, total thickness ¾ in.	.....	70	.....	2.50	.....	0.40	(4)
	Gypsum plaster and expanded vermiculite, 4 to 1 mix	39.9	75	0.85	....	1.18	.....	(3)
	Insulating plaster 0.9 in. thick applied to ⅜ in. gypsum board	54.0	75	.....	1.07	.....	0.93	(3)
ROOFING	Asbestos shingles	65.0	75	.....	6.0	.....	0.17	(3)
	Asphalt, composition or prepared	70.0	75	.....	6.5	.....	0.15	(3)
	Asphalt shingles	70.0	75	.....	6.5	.....	0.15	(3)
	Built-up roofing, bitumen or felt, gravel or slag surfaced[a]	.....	....	1.33	....	0.75	.....	(2)
	Slate	.....	....	10.00	....	0.10	.....	
	Wood shingles	.....	....	.....	1.28	.....	0.78	
WOODS	Balsa	20.0	90	0.58	....	1.72	.....	(1)
	Balsa	8.8	90	0.38	....	2.63	.....	(1)
	Balsa	7.3	90	0.33	....	3.03	.....	(1)
	California redwood, 0 per cent moisture[a]	28.0	75	0.70	....	1.43	.....	(4)
	Cypress	28.7	86	0.67	....	1.49	.....	(1)
	Douglas fir, 0 per cent moisture[a]	34.0	75	0.67	....	1.49	.....	(4)
	Eastern hemlock, 0 per cent moisture[a]	30.0	75	0.76	....	1.32	.....	(4)
	Long leaf yellow pine, 0 per cent moisture[a]	40.0	75	0.86	....	1.16	.....	(4)
	Mahogany	34.3	86	0.90	....	1.11	.....	(1)

See notes on Page 117.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS—Continued

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)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
WOODS—(Continued)	Hard maple, 0 per cent moisture[a]	46.0	75	1.05	....	0.95	.....	(4)
	Maple	44.3	86	1.10	....	0.91	.....	(1)
	Maple, across grain	40.0	75	1.20	....	0.83	.....	(3)
	Norway pine, 0 per cent moisture[a]	32.0	75	0.74	....	1.35	.....	(4)
	Red cypress, 0 per cent moisture[a]	32.0	75	0.79	....	1.27	.....	(4)
	Red oak, 0 per cent moisture[a]	48.0	75	1.18	....	0.85	.....	(4)
	Short leaf yellow pine, 0 per cent moisture[a]	36.0	75	0.91	....	1.10	.....	(4)
	Soft elm, 0 per cent moisture[a]	34.0	75	0.88	....	1.14	.....	(4)
	Soft maple, 0 per cent moisture[a]	42.0	75	0.95	....	1.05	.....	(4)
	Sugar pine, 0 per cent moisture[a]	28.0	75	0.64	....	1.56	.....	(4)
	Virginia pine	34.3	86	0.96	....	1.04	.....	(1)
	West coast hemlock, 0 per cent moisture[a]	30.0	75	0.79	....	1.27	.....	(4)
	White pine	31.2	86	0.78	....	1.28	.....	(1)
	Yellow pine	.....	....	1.00	....	1.00	.....	(3)
	Sawdust, various	12.0	90	0.41	....	2.44	.....	(1)
	Shavings, various from planer	8.8	90	0.41	....	2.44	.....	(1)
	Shavings, from maple beech and birch (coarse)	13.2	90	0.36	....	2.78	.....	(1)
INSULATING MATERIALS Blanket and Bat Insulations	Chemically treated wood fibers held between layers of strong paper	3.62	70	0.25	....	4.00	.....	(3)
	Eel grass between strong paper	4.60	90	0.26	....	3.85	.....	(1)
	Eel grass between strong paper	3.40	90	0.25	....	4.00	.....	(1)
	Flax fibers between strong paper	4.90	90	0.28	....	3.57	.....	(1)
	Chemically treated hog hair between kraft paper	5.76	71	0.26	....	3.85	.....	(3)
	Chemically treated hog hair between kraft paper and asbestos paper	7.70	71	0.28	....	3.57	.....	(3)
	Hair felt between layers of paper	11.00	75	0.25	....	4.00	.....	(3)
	Kapok between burlap or paper	1.00	90	0.24	....	4.17	.....	(1)
	Stitched and creped expanding fibrous blanket	1.50	70	0.27	....	3.70	.....	(3)
	Paper and asbestos fiber with emulsified asphalt binder	4.2	94	0.28	....	3.57	.....	(1)
	Cotton insulating bat	0.875	72	0.24	....	4.17	.....	(3)
	Cotton fibers							
	Short Staple Linters, Fireproofed	6.25	90	0.25	....	4.00	.....	(1)
	Short Staple Linters, Fireproofed	4.50	90	0.24	....	4.17	.....	(1)
	Short Staple Linters, Fireproofed	2.45	90	0.24	....	4.17	.....	(1)
	Short Staple Linters, Fireproofed	1.60	90	0.26	....	3.85	.....	(1)
	Short Staple Linters, Fireproofed	0.85	90	0.29	....	3.45	.....	(1)
	Short Staple Linters, Fireproofed	0.65	90	0.30	....	3.33	.....	(1)
	Felted cattle hair	13.00	90	0.26	....	3.84	.....	(1)
	Felted cattle hair	11.00	90	0.26	....	3.84	.....	(1)
	Felted hair and asbestos	7.80	90	0.28	....	3.57	.....	(1)
	Ground paper between two layers, each ⅜ in. thick made up of two layers of kraft paper (sample ¾ in. thick)	12.1	75	.....	0.40	.....	2.50	(4)
Reflective	See Table 1, Section C	.....	....	.....	....	.....	.....	
Insulating Board	Made from sugar cane fiber	13.5	70	0.33	....	3.03	.....	(3)
	Made from corn stalks	15.00	71	0.33	....	3.03	.....	(3)
	Made from exploded wood fibers	17.90	78	0.32	....	3.12	.....	(4)
	Made from hard wood fibers	15.20	70	0.32	....	3.12	.....	(3)
	Made from wood fiber	15.90	72	0.33	....	3.03	.....	(3)
	Made from wood fiber	15.00	70	0.33	....	3.03	.....	(3)
	Made from wood fiber	.....	52	0.33	....	3.03	.....	(6)
	Made from wood fiber	8.50	72	0.29	....	3.45	.....	(3)
	Made from wood fiber	15.20	....	0.33	....	3.03	.....	(3)
	Made from wood fiber	16.90	90	0.34	....	2.94	.....	(1)

See notes on Page 117.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING AND INSULATING MATERIALS—Concluded

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)	Mean Temp (Fahr Deg)	Conductivity or Conductance (k)	Conductivity or Conductance (C)	Resistance: Per Inch Thickness $\left(\frac{1}{k}\right)$	Resistance: For Thickness Listed $\left(\frac{1}{C}\right)$	Authority
INSULATING MATERIALS—Continued								
Insulating Board—Continued	Made from licorice root	16.1	81	0.34	....	2.94	.....	(3)
	½ in. insulating boards without special finish[7] (eleven samples)	16.5 to 21.8	90	0.33 to 0.40	....	3.03 to 2.50	....	(1)
	1 in. insulating board[8]	13.2	....	0.34	....	2.94	.....	(4)
Loose Fill Type	Made from ceiba fibers	1.90	75	0.23	....	4.35	.....	(3)
	Made from ceiba fibers	1.60	75	0.24	....	4.17	.....	(3)
	Fibrous material made from dolomite and silica	1.50	75	0.27	....	3.70	.....	(3)
	Fibrous material made from slag	9.40	103	0.27	....	3.70	.....	(1)
	Redwood bark	3.00	90	0.31	....	3.22	.....	(1)
	Redwood bark	5.00	75	0.26	....	3.84	.....	(3)
	Glass wool fibers 0.0003 in. to 0.006 in. in diameter	1.50	75	0.27	....	3.70	.....	(3)
	Granular insulation made from combined silicate of lime and alumina	4.20	72	0.24	....	4.17	.....	(3)
	Expanded vermiculite	.....	....	0.48	....	2.08	.....	(1)
	Expanded vermiculite, particle size—3 + 14	6.2	....	0.32	....	3.12	.....	(3)
	Regranulated cork about ¼ in. particles	8.10	90	0.31	....	3.22	.....	(1)
	Hand applied granular mineral wool 2 in. to 6 in. thick; horizontal position[9]. No covering	6.05 to 7.13	....	0.30 to 0.33	....	3.33 to 3.03	....	(4)
	4 in. machine blown granular mineral wool, horizontal position[9]. No covering	5.74	....	0.30	....	3.33	.....	(4)
	Rock wool	10.0	90	0.27	....	3.70	.....	(1)
Slab Insulations	Corkboard, no added binder	14.0	90	0.34	....	2.94	.....	(1)
	Corkboard, no added binder	10.6	90	0.30	....	3.33	.....	(1)
	Corkboard, no added binder	7.0	90	0.27	....	3.70	.....	(1)
	Corkboard, no added binder	5.4	90	0.25	....	4.00	.....	(1)
	Corkboard[8]	8.7	....	0.29	....	3.45	.....	(4)
	Corkboard, asphaltic binder	14.5	90	0.32	....	3.12	.....	(1)
	Chemically treated hog hair with film of asphalt	10.0	75	0.28	....	3.57	.....	(3)
	Sugar cane fiber insulation blocks encased in asphalt membrane	13.8	70	0.30	....	3.33	.....	(3)
	Made from 85 per cent magnesia and 15 per cent asbestos	19.3	86	0.51	....	1.96	.....	(1)
	Made from shredded wood and cement	24.2	72	0.46	....	2.17	.....	(3)
	Made from shredded wood and cement[8]	29.8	....	0.77	....	1.30	.....	(4)

See notes on Page 117.

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 radiation is illustrated in

Table 3, which applies to a vertical surface at 80 F, with ambient air at 70 F and effective emissivity equal to 0.83 [3].

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. In a building heated by convection there is only a slight difference between the temperatures of the interior wall surface and the surroundings, but if the building is heated by radiant panels there may be a considerable difference [4]. (See also Chapter 31.)

The convection part of the surface conductance coefficient is affected markedly by air movement. This is illustrated by Fig. 3, which shows the surface conductances for different materials at a mean temperature of 20 F and for wind velocities up to 40 mph. These include the radiation portion of the coefficient, which for ordinary building materials under these conditions would be constant at about 0.7 Btu.

TABLE 3. VARIATION IN SURFACE CONDUCTANCE COEFFICIENT 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
Surface Coefficient—surface and air temperature only—Btu per (hr) (sq ft) (F deg difference of surface and air)	1.10	1.52	1.62	2.36	3.15

Due to these variations for different conditions the selection of surface conductance coefficients for a practical building becomes a matter of judgment. In calculating the over-all heat transmission coefficients for the walls, etc. of Tables 5 to 18, 1.65 has been selected as an *average inside surface conductance* and 6.0 as an *average outside surface conductance* for a 15-mile wind. These values apply only to ordinary building materials and should not be used for bright metal surfaces having a low emissivity.

In special cases, where surface conductance coefficients become important factors in the over-all rates of heat transfer, more selective coefficients may be required. The surface conductance values given in Table 1, **Section A** are based on recent tests and are for still air conditions and surface emissivities of 0.83 and 0.05 respectively, and may be used where it is desirable to differentiate between horizontal and vertical surfaces or where coefficients applicable to low-emissivity surfaces are required.

Air Space Conductance

The transfer of heat across an air space involves the boundary surfaces as well as the intervening air, consequently the factors influencing surface conductance play an important part in determining the conductance of the air space. The coefficients given for air space conductance represent the total conductance from surface to surface.

The radiation portion of the coefficient is affected by the difference in temperature between the boundary surfaces and by their respective

emissivities and is practically independent of depth. The convection and conduction transfer is controlled by depth and shape of the air space, the roughness of the boundary surfaces, the mean temperature and the direction of heat flow. For air spaces usually employed in building construction, the radiation and convection factors vary independently of each other.

Table 1, **Section B** gives experimentally-determined conductances of vertical air spaces bounded by such materials as paper, wood, plaster, etc., having emissivity coefficients of 0.8 or higher, and having extended parallel surfaces perpendicular to the direction of heat flow. The conductances decrease as the depth is increased but change only slightly

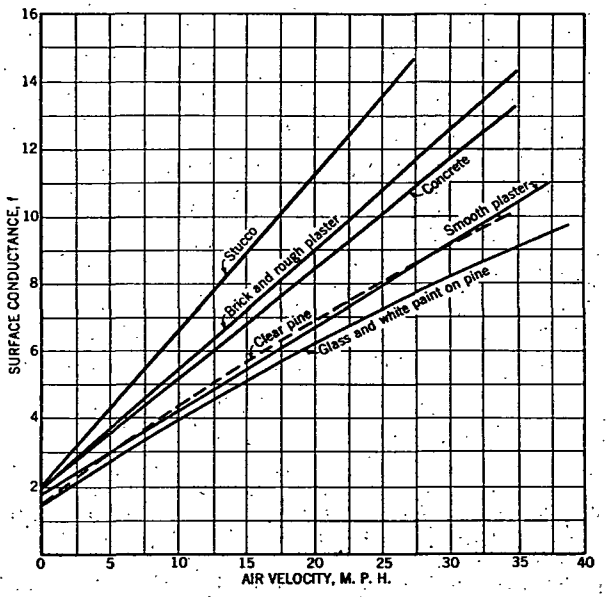


FIG. 3. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT 20 F MEAN TEMPERATURE.

for spaces greater than ¾ in. Air space tests reported by Wilkes and Peterson gave conductance values for air spaces of 3⅝ in. depth having boundary surfaces with emissivity values of 0.83 as follows [5].

Vertical	1.17
Horizontal (heat flow upward)	1.32
Horizontal (heat flow downward)	0.94

Since, in buildings, the same constructions may be used for conditions where the direction of heat flow may be in one direction or its opposite, and since much of the construction involves vertical air spaces, an average value of 1.10 Btu per (hour) (square foot) (Fahrenheit degree temperature difference) was chosen for use in calculating the over-all coefficients in Tables 5 to 18 wherever air spaces ¾ in. or more in depth were involved.

If one or both boundary surfaces of an air space are faced with metals which have low emissivity surfaces, the radiant heat transfer will be greatly reduced in comparison with that occurring from surfaces of ordinary building materials. Table 1, **Section C** gives conductances and resistances of air spaces *bounded by one reflective surface* with an emissivity of 0.05. These values include heat transferred both by radiation and convection, but the radiation component is relatively small for the test conditions.

When reflective materials are installed with single or multiple air spaces, the position (vertical, horizontal or inclined) of the material and the direction of heat flow must be taken into consideration. For example, the resistance to *upward* heat flow is about one-third the resistance to *downward* heat flow in a horizontal position (Table 1, **Section C**). The difference between the conductance through vertical air spaces and that through horizontal and sloping air spaces with upward heat flow is considerably less. For upward heat flow it is recommended that a value of 0.46 be used for the conductance of horizontal or sloping air spaces bounded on one side by reflective materials having an emissivity of approximately 0.05. The same conductance value is also recommended for similar vertical air spaces.

When considering heat transfer to and from reflective surfaces in building construction, the emissivity should be known. This can be determined directly for the long wave length radiation corresponding to average room and wall temperatures. The possibility of change in emissivity with time of exposure due to surface coatings, chemical action, deposition of dust, etc. must be considered in selecting a material for use [6].

PRACTICAL COEFFICIENTS AND THEIR USE

For practical purposes it is necessary to have average coefficients that may be applied to various materials and types of construction without the necessity of making actual tests. In Table 2 coefficients are given for a group of materials which have been selected from tests by various authorities. Since there is some variation in the resulting values due to variations in materials and in test conditions, average values for the usual conditions encountered in building practice have been selected and listed in Table 4. These coefficients were used in the calculation of over-all coefficients given in Tables 5 to 18. These tables constitute typical examples of combinations frequently used, but any special constructions not given can be computed by the use of the conductivity values in Table 4 and the fundamental heat transfer formulae.

Caution

The user should realize that the average conductivity and conductance values given in Tables 2 or 4 do not necessarily apply to all products of the same general description. In using these values judgment should be exercised with regard to the extent to which the product (either as received or as applied) will comply with the tabulated values. Exact conductivities or conductances for specific materials should be obtained from the manufacturer.

Insulating Materials

In order to determine the benefit derived from the addition of insulating materials to a given construction, the over-all coefficient of heat

TABLE 4. CONDUCTIVITIES (k) AND CONDUCTANCES (C) USED IN CALCULATING HEAT TRANSMISSION COEFFICIENTS (U) IN TABLES 5 TO 18

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	CONDUCTIVITY OR CONDUCTANCE		RESISTANCE	
		(k)	(C)	Per Inch Thickness ($\frac{1}{k}$)	For Thickness Listed ($\frac{1}{C}$)
AIR SPACES					
Bounded by ordinary materials	Vertical[a], ¾ in. or more in width	.....	1.10	.....	0.91
Bounded by aluminum foil	Vertical[a], ¾ in. or more in width	.....	0.46	.....	2.17
EXTERIOR FINISHES (Frame Walls)					
Brick Veneer	4 in. thick (nominal)	.....	2.27	.....	0.44
Stucco (1 in.)		12.50	.....	0.08	.....
Wood Shingles		.....	1.28	.....	0.78
Yellow Pine Lap Siding		.....	1.28	.....	0.78
INSULATING MATERIALS					
Aluminum Foil	See Air Spaces	.....	.....	.....	.....
Bats and Blankets	Made from mineral or vegetable fiber or animal hair, enclosed or open	0.27	.....	3.70	.....
Corkboard	Pure, no added binder	0.30	.....	3.33	.....
Insulating Board	Vegetable fiber	0.33	.....	3.03	.....
Mineral Wool	Fiber made from rock, slag or glass	0.27	.....	3.70	.....
Vermiculite	Expanded	0.48	.....	2.08	.....
INTERIOR FINISHES					
Composition Wallboard	⅜ in. to ½ in. thick	0.50	.....	2.00	.....
Gypsum Plaster		3.30	.....	0.30	.....
Gypsum Board (⅜ in.)	Plain or decorated	.....	3.70	.....	0.27
Gypsum Lath (⅜ in.) and Plaster	Plaster thickness assumed ½ in.	.....	2.4	.....	0.42
Insulating Board (½ in.)	Plain or decorated	.....	0.66	.....	1.52
Insulating Board Lath (½ in.) and Plaster	Plaster thickness assumed ½ in.	.....	0.60	.....	1.67
Insulating Board Lath (1 in.) and Plaster	Plaster thickness assumed ½ in.	.....	0.31	.....	3.18
Metal Lath and Plaster	Plaster thickness assumed ¾ in.	.....	4.40	.....	0.23
Plywood (⅜ in.)	Plain or decorated	.....	2.12	.....	0.47
Wood Lath and Plaster		.....	2.50	.....	0.40
MASONRY MATERIALS					
Brick	Adobe, assumed 4 in. thick	.....	0.89	.....	1.12
Brick	Common, assumed 4 in. thick	.....	1.25	.....	0.80
Brick	Face, assumed 4 in. thick	.....	2.30	.....	0.43
Cement Mortar		12.00	.....	0.08	.....
3 in. Clay tile (hollow)		.....	1.28	.....	0.78
4 in. Clay tile (hollow)		.....	1.00	.....	1.00
6 in. Clay tile (hollow)		.....	0.64	.....	1.57
8 in. Clay tile (hollow)		.....	0.60	.....	1.67
10 in. Clay tile (hollow)		.....	0.58	.....	1.72
12 in. Clay tile (hollow)		.....	0.40	.....	2.50
16 in. Clay tile (hollow)		.....	0.31	.....	3.23
Concrete	Light weight aggregate[b]	2.50	.....	0.40	.....
Concrete	Sand and gravel aggregate	12.00	.....	0.08	.....
3 in. Concrete blocks	Hollow, cinder aggregate	.....	1.28	.....	0.78
4 in. Concrete blocks	Hollow, cinder aggregate	.....	1.00	.....	1.00
8 in. Concrete blocks	Hollow, gravel aggregate	.....	1.00	.....	1.00
12 in. Concrete blocks	Hollow, gravel aggregate	.....	0.80	.....	1.25
8 in. Concrete blocks	Hollow, cinder aggregate	.....	0.60	.....	1.66
12 in. Concrete blocks	Hollow, cinder aggregate	.....	0.53	.....	1.88
8 in. Concrete blocks	Hollow, light weight aggregate[b]	.....	0.50	.....	2.00
12 in. Concrete blocks	Hollow, light weight aggregate[b]	.....	0.47	.....	2.13
Gypsum fiber concrete	87½ per cent gypsum and 12½ per cent wood chips	1.66	.....	0.60	.....
3 in. Gypsum tile	Hollow	.....	0.61	.....	1.64
4 in. Gypsum tile	Hollow	.....	0.46	.....	2.18
Stucco		12.50	.....	0.08	.....
Tile and Terrazzo	For flooring	12.00	.....	0.08	.....
Stone		12.50	.....	0.08	.....

[a] Conductance values for horizontal air spaces depend on whether the heat flow is upward or downward, but in most cases it is sufficiently accurate to use the same values for horizontal as for vertical air spaces.

[b] Expanded slag, burned clay or pumice.

TABLE 4. CONDUCTIVITIES (k) AND CONDUCTANCES (C) USED IN CALCULATING HEAT TRANSMISSION COEFFICIENTS (U) IN TABLES 5 TO 18—Concluded

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	CONDUCTIVITY OR CONDUCTANCE		RESISTANCE	
		(k)	(C)	Per Inch Thickness ($\frac{1}{k}$)	For Thickness Listed ($\frac{1}{C}$)
ROOFING MATERIALS					
Asbestos Shingles		.....	6.00	.....	0.17
Asphalt Shingles		.....	6.50	.....	0.15
Built-up Roofing	Assumed thickness ⅜ in.	.....	3.53	.....	0.28
Heavy Roll Roofing		.....	6.50	.....	0.15
Slate	Assumed ½ in.	10.00	20.00	0.10	0.05
Wood Shingles		.....	1.28	.....	0.78
SHEATHING					
Gypsum (½ in.)		.....	2.82	.....	0.35
Insulating Board ($^{25}/_{32}$ in.)		.....	0.42	.....	2.37
Plywood (⅜ in.)		.....	2.56	.....	0.39
Fir or Yellow Pine (1 in.)	Actual thickness $^{25}/_{32}$ in.	.....	1.02	.....	0.98
Fir, plus building paper	Actual thickness $^{25}/_{32}$ in.	.....	0.86	.....	1.16
SURFACES					
Still air	Ordinary non-reflective materials, vertical	.....	1.65	.....	0.61
15 mph wind velocity	Ordinary non-reflective materials, vertical	.....	6.00	.....	0.17
WOODS					
Fir sheathing (1 in.) building paper and Yellow Pine lap siding		.....	0.50	.....	2.00
Maple or Oak		1.15	.....	0.87	.....
Yellow Pine or Fir		0.80	.....	1.25	.....

transmission U_i of the insulated construction may be compared with the corresponding coefficient U without insulation. Attention is called to the necessity of applying the insulating material in accordance with the manufacturer's specification. The engineer must carefully evaluate 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.

Computed Heat Transmission Coefficients

Computed over-all heat transmission coefficients of many common types of building construction are given in Tables 5 to 18, inclusive, each coefficient being identified by a serial number except in Table 18. For example, the coefficient U of a brick-veneer, frame wall with wood sheathing and ½ in. of plaster on gypsum lath is 0.27, (Wall No. 28-C in Table 5) and with 2 in. of blanket or bat insulation the coefficient would be 0.097 (No. 49-B in Table 6).

Example 1. Calculate the coefficient of heat transmission U of a brick-veneer, frame wall with wood sheathing, building paper, and ½ in. plaster on ⅜ in. gypsum lath, based on a wind exposure of 15 mph. Also calculate the coefficient when 2 in. of bat insulation is added, leaving an air space in the wall, and correcting for framing amounting to 15 per cent of the wall area.

Solution: Starting from the exterior, the resistances making up the total heat resistance are (1) exterior surface, (2) brick-veneer, (3) wood sheathing $^{25}/_{32}$ in. thick and building paper, (4) air space 3⅝ in. wide, (5) ⅜ in. gypsum lath and plaster, (6) interior surface. Then using the values from Table 4 in Equation 4:

$$U = \frac{1}{\frac{1}{6.0} + \frac{1}{2.27} + \frac{1}{0.86} + \frac{1}{1.10} + \frac{1}{2.4} + \frac{1}{1.65}}$$

$$= \frac{1}{0.17 + 0.44 + 1.16 + 0.91 + 0.42 + 0.61} = \frac{1}{3.71} = 0.27$$

When 2 in. of bat insulation is added in the air space in the wall, an air space will still be left since it was originally 3⅝ in. wide. Then,

$$U = \frac{1}{3.71 + \frac{2}{0.27}} = \frac{1}{11.11} = 0.090$$

This coefficient (0.090) applies only to the wall where no studding or framing is present and is assumed to constitute 85 per cent of the whole wall. Then for 1 sq ft of wall the heat transfer through the insulated portion is 0.85 × 0.090 or 0.0765 Btu.

In the portion of wall where framing is present, the air space will be replaced with an equivalent thickness of wood framing. Then,

$$U \text{ (through framing)} = \frac{1}{\frac{1}{6.0} + \frac{1}{2.27} + \frac{1}{0.86} + \frac{3.625}{0.80} + \frac{1}{2.4} + \frac{1}{1.65}}$$

$$U = 0.136$$

Then for 1 sq ft of wall the heat transfer through the framing is 15 per cent of 0.136 or 0.0204 Btu. The total heat transfer through 1 sq ft of wall (with 3⅝ in. framing covering 15 per cent of area) equals

$$U \text{ (corrected)} = 0.0765 + 0.0204 = 0.097 \text{ Btu}$$

This is the value shown for No. 49-B in Table 6, which is included to eliminate the need for the calculation of framing corrections when insulation is used in frame construction.

In making the calculations for values of U shown in Tables 5 to 18, the following conditions have been assumed:

Equilibrium or steady-state heat transfer, eliminating effects of heat capacity.
Surrounding surfaces at ambient air temperatures.
Exterior wind velocity of 15 mph.
Surface emissivity of ordinary building materials = 0.83.
No correction for position or direction of heat flow. (Average coefficients used).
Air spaces are ¾ in. or more in width.
Variation of conductivity with mean temperature neglected.
Corrections for framing made on basis of parallel heat flow through 2 × 4 in. (nominal) studs, 16 in. on centers, the framing covering 15 per cent of wall area.
Actual thicknesses of lumber assumed to be as follows:

Nominal	*Actual*
1 in. (S-2-S)	25/32 in.
1½ in. (S-2-S)	1 5/16 in.
2 in. (S-2-S)	1 5/8 in.
2½ in. (S-2-S)	2⅛ in.
3 in. (S-2-S)	2⅝ in.
4 in. (S-2-S)	3⅝ in.
Finish flooring, (maple or oak)	13/16 in.

Coefficients for frame construction are corrected for the effect of framing where such correction would increase the coefficients, but not where the correction would decrease the coefficients [7].

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

TABLE 5. COEFFICIENTS OF TRANSMISSION (U) OF FRAME WALLS

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.

NO INSULATION BETWEEN STUDS[a] (SEE TABLE 6)

EXTERIOR FINISH	INTERIOR FINISH	TYPE OF SHEATHING: GYPSUM (½ IN. THICK) A	PLYWOOD (⅜ IN. THICK) B	WOOD[f] (25/32 IN. THICK) BLDG. PAPER C	INSULATING BOARD (25/32 IN. THICK) D	WALL NUMBER
WOOD SIDING (Clapboard)	Metal Lath and Plaster[b]	0.33	0.32	0.26	0.20	1
	Gypsum Board (⅜ in.) Decorated	0.32	0.32	0.26	0.20	2
	Wood Lath and Plaster	0.31	0.31	0.25	0.19	3
	Gypsum Lath (⅜ in.) Plastered[c]	0.31	0.30	0.25	0.19	4
	Plywood (⅜ in.) Plain or Decorated	0.30	0.30	0.24	0.19	5
	Insulating Board (½ in.) Plain or Decorated	0.23	0.23	0.19	0.16	6
	Insulating Board Lath (½ in.) Plastered[c]	0.22	0.22	0.19	0.15	7
	Insulating Board Lath (1 in.) Plastered[c]	0.17	0.17	0.15	0.12	8
WOOD[d] SHINGLES	Metal Lath and Plaster[b]	0.25	0.25	0.26	0.17	9
	Gypsum Board (⅜ in.) Decorated	0.25	0.25	0.26	0.17	10
	Wood Lath and Plaster	0.24	0.24	0.25	0.16	11
	Gypsum Lath (⅜ in.) Plastered[c]	0.24	0.24	0.25	0.16	12
	Plywood (⅜ in.) Plain or Decorated	0.24	0.24	0.24	0.16	13
	Insulating Board (½ in.) Plain or Decorated	0.19	0.19	0.19	0.14	14
	Insulating Board Lath (½ in.) Plastered[c]	0.19	0.18	0.19	0.13	15
	Insulating Board Lath (1 in.) Plastered[c]	0.14	0.14	0.15	0.11	16
STUCCO	Metal Lath and Plaster[b]	0.43	0.42	0.32	0.23	17
	Gypsum Board (⅜ in.) Decorated	0.42	0.41	0.31	0.23	18
	Wood Lath and Plaster	0.40	0.39	0.30	0.22	19
	Gypsum Lath (⅜ in.) Plastered[c]	0.39	0.39	0.30	0.22	20
	Plywood (⅜ in.) Plain or Decorated	0.39	0.38	0.29	0.22	21
	Insulating Board (½ in.) Plain or Decorated	0.27	0.27	0.22	0.18	22
	Insulating Board Lath (½ in.) Plastered[c]	0.26	0.26	0.22	0.17	23
	Insulating Board Lath (1 in.) Plastered[c]	0.19	0.19	0.16	0.14	24
BRICK VENEER[e]	Metal Lath and Plaster[b]	0.37	0.36	0.28	0.21	25
	Gypsum Board (⅜ in.) Decorated	0.36	0.36	0.28	0.21	26
	Wood Lath and Plaster	0.35	0.34	0.27	0.20	27
	Gypsum Lath (⅜ in.) Plastered[c]	0.34	0.34	0.27	0.20	28
	Plywood (⅜ in.) Plain or Decorated	0.34	0.33	0.27	0.20	29
	Insulating Board (½ in.) Plain or Decorated	0.25	0.25	0.21	0.17	30
	Insulating Board Lath (½ in.) Plastered[c]	0.24	0.24	0.20	0.16	31
	Insulating Board Lath (1 in.) Plastered[c]	0.18	0.18	0.15	0.13	32

[a] Coefficients not weighted; effect of studding neglected.
[b] Plaster assumed ¾ in. thick.
[c] Plaster assumed ½ in. thick.
[d] Furring strips (1 in. nominal thickness) between wood shingles and all sheathings except wood.
[e] Small air space and mortar between building paper and brick veneer neglected.
[f] Nominal thickness, 1 in.

TABLE 6. COEFFICIENTS OF TRANSMISSION (U) OF FRAME WALLS WITH INSULATION BETWEEN FRAMING[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.

COEFFICIENT WITH *NO* INSULATION BETWEEN FRAMING	COEFFICIENT WITH INSULATION BETWEEN FRAMING: MINERAL WOOL OR VEGETABLE FIBERS IN BLANKET OR BAT FORM[c] (Thickness below)				NUMBER
	1 IN.	2 IN.	3 IN.	3½ IN. MINERAL WOOL BETWEEN FRAMING[d]	
	A	B	C	D	
0.11	0.078	0.063	0.054	0.051	33
0.12	0.083	0.067	0.056	0.053	34
0.13	0.088	0.070	0.058	0.055	35
0.14	0.092	0.072	0.061	0.057	36
0.15	0.097	0.075	0.062	0.059	37
0.16	0.10	0.078	0.064	0.060	38
0.17	0.10	0.080	0.066	0.062	39
0.18	0.11	0.082	0.067	0.063	40
0.19	0.11	0.084	0.069	0.065	41
0.20	0.12	0.086	0.070	0.066	42
0.21	0.12	0.088	0.072	0.067	43
0.22	0.12	0.089	0.073	0.068	44
0.23	0.12	0.091	0.074	0.069	45
0.24	0.12	0.093	0.075	0.070	46
0.25	0.13	0.094	0.076	0.071	47
0.26	0.13	0.096	0.077	0.072	48
0.27	0.14	0.097	0.078	0.073	49
0.28	0.14	0.098	0.079	0.073	50
0.29	0.14	0.10	0.080	0.075	51
0.30	0.14	0.10	0.080	0.075	52
0.31	0.14	0.10	0.081	0.076	53
0.32	0.15	0.10	0.082	0.077	54
0.33	0.15	0.10	0.083	0.077	55
0.34	0.15	0.10	0.083	0.078	56
0.35	0.15	0.11	0.084	0.078	57
0.36	0.15	0.11	0.085	0.079	58
0.37	0.16	0.11	0.085	0.080	59
0.38	0.16	0.11	0.086	0.080	60
0.39	0.16	0.11	0.086	0.081	61
0.40	0.16	0.11	0.087	0.082	62
0.41	0.16	0.11	0.087	0.082	63
0.42	0.16	0.11	0.088	0.082	64
0.43	0.17	0.11	0.088	0.082	65
0.44	0.17	0.11	0.089	0.083	66

[a] This table may be used for determining the coefficients of transmission of frame constructions with the types and thicknesses of insulation indicated in Columns A to D inclusive between framing. Columns A, B and C may be used for walls, ceilings or roofs with only one air space between framing but are not applicable to ceilings with *no* flooring above. (See Table 11.) Column D is applicable to walls only. *Example:* Find the coefficient of transmission of a frame wall consisting of wood siding, 25/32 in. insulating board sheathing, studs, gypsum lath and plaster, with 2 in. blanket insulation between studs. According to Table 5, a wall of this construction with *no insulation* between studs has a coefficient of 0.19 (Wall No. 4D). Referring to Column B above, it will be found that a wall of this value with 2 in. blanket insulation between the studs has a coefficient of 0.084.

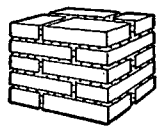
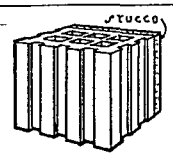
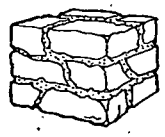
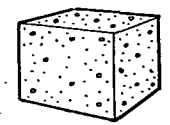
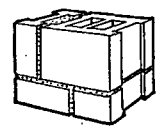
[b] Coefficients corrected for 2 x 4 framing, 16 in. on centers—15 per cent of surface area.

[c] Based on one air space between framing.

[d] No air space.

TABLE 7. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS

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.

TYPE OF MASONRY		THICKNESS OF MASONRY INCHES	INTERIOR FINISH (PLUS INSULATION WHERE INDICATED): A Plain Walls—No Interior Finish	B Plaster (½ in.) on Walls	C Metal Lath and Plaster[f] Furred[h]	D Gypsum Board (⅜ in.) Decorated—Furred[h]	E Gypsum Lath (⅜ in.) Plastered—Furred[h]	F Insulating Board (½ in.) Plain or Decorated—Furred[h]	G Insulating Board Lath (½ in.) Plastered—Furred[h]	H Insulating Board Lath (1 in.) Plastered—Furred[h]	I Gypsum Lath Plastered Plus 1 in. Blanket Insulation—Furred[h]	WALL NUMBER
			A	B	C	D	E	F	G	H	I	
Solid[a] Brick		8	0.50	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	67
		12	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	68
		16	0.28	0.27	0.21	0.21	0.20	0.17	0.16	0.13	0.12	69
Hollow[b] Tile (Stucco Exterior Finish)		8	0.40	0.37	0.27	0.27	0.26	0.20	0.20	0.15	0.13	70
		10	0.39	0.37	0.27	0.27	0.26	0.20	0.19	0.15	0.13	71
		12	0.30	0.28	0.22	0.22	0.21	0.17	0.17	0.13	0.12	72
		16	0.24	0.24	0.19	0.19	0.18	0.15	0.15	0.12	0.11	73
Stone[c]		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	74
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	75
		16	0.49	0.45	0.31	0.31	0.29	0.22	0.22	0.16	0.14	76
		24	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	77
Poured Concrete[d]		6	0.79	0.71	0.42	0.41	0.39	0.27	0.26	0.19	0.16	78
		8	0.70	0.64	0.39	0.38	0.36	0.26	0.25	0.18	0.16	79
		10	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	80
		12	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	81
Hollow Concrete Blocks		**Gravel Aggregate**										
		8	0.56	0.52	0.34	0.34	0.32	0.24	0.23	0.17	0.15	82
		12	0.49	0.46	0.32	0.31	0.30	0.22	0.22	0.16	0.14	83
		Cinder Aggregate										
		8	0.41	0.39	0.28	0.28	0.27	0.21	0.20	0.15	0.13	84
		12	0.38	0.36	0.26	0.26	0.25	0.20	0.19	0.15	0.13	85
		Light Weight Aggregate[e]										
		8	0.36	0.34	0.26	0.25	0.24	0.19	0.19	0.15	0.13	86
		12	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	87

[a] Based on 4 in. hard brick and remainder common brick.

[b] The 8 in. and 10 in. tile figures are based on two cells in the direction of heat flow. The 12 in. tile is based on three cells in the direction of heat flow. The 16 in. tile consists of one 10 in. and one 6 in. tile each having two cells in the direction of heat flow.

[c] Limestone or sandstone.

[d] These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.

[e] Expanded slag, burned clay or pumice.

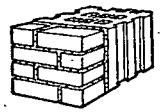
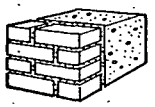
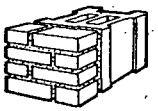
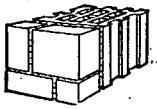
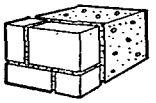
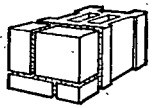
[f] Thickness of plaster assumed ¾ in.

[g] Thickness of plaster assumed ½ in.

[h] Based on 2 in. furring strips; one air space.

TABLE 8. COEFFICIENTS OF TRANSMISSION (*U*) OF BRICK AND STONE VENEER MASONRY WALLS

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.

TYPICAL CONSTRUCTION	FACING	BACKING	INTERIOR FINISH (PLUS INSULATION WHERE INDICATED)									WALL NUMBER
			Plain Walls—no Interior Finish	Plaster (½ in.) on Walls	Metal Lath and Plaster—Furred[e]	Gypsum Board (⅜ in.), Decorated—Furred[e]	Gypsum Lath (⅜ in.) Plastered[f]—Furred[e]	Insulating Board (½ in.) Plain or Decorated—Furred[e]	Insulating Board Lath (½ in.) Plastered[f]—Furred[e]	Insulating Board Lath (1 in.) Plastered[f]—Furred[e]	Gypsum Lath Plastered[f] Plus 1 in. Blanket Insulation—Furred[e]	
			A	B	C	D	E	F	G	H	I	
	4 in. Brick Veneer[a]	6 in. Hollow Tile[b]	0.35	0.34	0.25	0.25	0.24	0.19	0.18	0.14	0.13	88
		8 in. Hollow Tile[b]	0.34	0.32	0.25	0.24	0.23	0.19	0.18	0.14	0.13	89
		6 in. Concrete	0.59	0.54	0.35	0.35	0.33	0.24	0.23	0.17	0.15	90
		8 in. Concrete	0.54	0.50	0.33	0.33	0.31	0.23	0.23	0.17	0.15	91
		8 in. Concrete Blocks[c] (Gravel Aggregate)	0.44	0.41	0.29	0.29	0.28	0.21	0.21	0.16	0.14	92
		8 in. Concrete Blocks[c] (Cinder Aggregate)	0.34	0.33	0.25	0.24	0.24	0.19	0.18	0.14	0.13	93
		8 in. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.31	0.29	0.23	0.23	0.22	0.18	0.17	0.14	0.12	94
	4 in. Cut Stone Veneer[a]	6 in. Hollow Tile[b]	0.37	0.35	0.26	0.26	0.25	0.19	0.19	0.15	0.13	95
		8 in. Hollow Tile[b]	0.35	0.34	0.25	0.25	0.24	0.19	0.19	0.14	0.13	96
		6 in. Concrete	0.63	0.58	0.37	0.36	0.34	0.25	0.24	0.18	0.15	97
		8 in. Concrete	0.57	0.53	0.35	0.34	0.33	0.24	0.23	0.17	0.15	98
		8 in. Concrete Blocks[c] (Gravel Aggregate)	0.47	0.44	0.30	0.30	0.29	0.22	0.21	0.16	0.14	99
		8 in. Concrete Blocks[c] (Cinder Aggregate)	0.36	0.34	0.25	0.25	0.24	0.19	0.19	0.15	0.13	100
		8 in. Concrete Blocks[c] (Light Weight Aggregate)[d]	0.32	0.30	0.23	0.23	0.22	0.18	0.17	0.14	0.12	101

[a] Calculations based on ½ in. cement mortar between backing and facing except in the case of the concrete backing which is assumed to be poured in place.

[b] The hollow tile figures are based on two air cells in the direction of heat flow.

[c] Hollow concrete blocks.

[d] Expanded slag, burned clay or pumice.

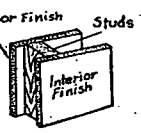
[e] Thickness of plaster assumed ¾ in.

[f] Thickness of plaster assumed ½ in.

[g] Based on 2 in. furring strips; one air space.

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.

INTERIOR FINISH 	SINGLE PARTITION (Finish on one side only of studs)	DOUBLE PARTITION (Finish on both sides of studs)		PARTITION NUMBER
		No insulation between studs	1 in. Blanket[d] between studs. One air space.	
	A	B	C	
Metal Lath and Plaster[b]	0.69	0.39	0.16	1
Gypsum Board (⅜ in.) Decorated	0.67	0.37	0.16	2
Wood Lath and Plaster	0.62	0.34	0.15	3
Gypsum Lath (⅜ in.) Plastered[c]	0.61	0.34	0.15	4
Plywood (⅜ in.) Plain or Decorated	0.59	0.33	0.15	5
Insulating Board (½ in.) Plain or Decorated	0.36	0.19	0.11	6
Insulating Board Lath (½ in.) Plastered[c]	0.35	0.18	0.11	7
Insulating Board Lath (1 in.) Plastered[c]	0.23	0.12	0.082	8

[a] Coefficients not weighted; effect of studding neglected.

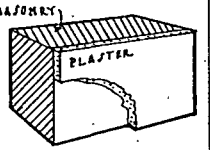
[b] Plaster assumed ¾ in. thick.

[c] Plaster assumed ½ in. thick.

[d] For partitions with other insulations between studs refer to Table 6, using values in Column B of above table in left hand column of Table 6. *Example:* What is the coefficient of transmission (*U*) of a partition consisting of gypsum lath and plaster on both sides of studs with 2 in. blanket between studs? *Solution:* According to above table, this partition with no insulation between studs (No. 4B) has a coefficient of 0.34. Referring to Table 6, it will be found that a wall having a coefficient of 0.34 with no insulation between studs, will have a coefficient of 0.10 with 2 in. of blanket insulation between studs (No. 56B).

TABLE 10. COEFFICIENTS OF TRANSMISSION (*U*) OF MASONRY PARTITIONS

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 PARTITION 		THICKNESS OF MASONRY (INCHES)	TYPE OF FINISH			PARTITION NUMBER
			No Finish (Plain walls)	Plaster One Side	Plaster Both Sides[a]	
			A	B	C	
Hollow Clay Tile		3	0.50	0.47	0.43	9
		4	0.45	0.42	0.40	10
Hollow Gypsum Tile		3	0.35	0.33	0.32	11
		4	0.29	0.28	0.27	12
Hollow Concrete Tile or Blocks	Cinder Aggregate	3	0.50	0.47	0.43	13
		4	0.45	0.42	0.40	14
	Light Weight Aggregate[b]	3	0.41	0.39	0.37	15
		4	0.35	0.34	0.32	16
Common Brick		4	0.50	0.46	0.43	17

[a] 2 in. solid plaster partition, $U = 0.53$.

[b] Expanded slag, burned clay or pumice.

TABLE 11. COEFFICIENTS OF TRANSMISSION (*U*) OF FRAME CONSTRUCTION CEILINGS AND FLOORS

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 CEILING	INSULATION BETWEEN, OR ON TOP OF, JOISTS (No Flooring Above)												WITH FLOORING[a] (On Top of Ceiling Joists)		NUMBER
	None	Insulating Board on Top of Joists		Blanket or Bat Insulation[b] Between Joists[c]			Vermiculite Insulation Between Joists[c]			Mineral Wool Insulation Between Joists[c]			Single Wood Floor[b]	Double Wood Floor[c]	
		½ In.	1 In.	1 In.	2 In.	3 In.	2 In.	3 In.	4 In.	2 In.	3 In.	4 In.			
	A	B	C	D	E	F	G	H	I	J	K	L	M	N	
No Ceiling		0.37	0.24										0.45	0.34	1
Metal Lath and Plaster[d]	0.69	0.26	0.19	0.19	0.12	0.093	0.18	0.14	0.11	0.12	0.093	0.077	0.30	0.25	2
Gypsum Board (⅜ in.) Plain or Decorated	0.67	0.26	0.18	0.19	0.12	0.092	0.18	0.13	0.10	0.12	0.092	0.077	0.30	0.24	3
Wood Lath and Plaster	0.62	0.25	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.24	4
Gypsum Lath (⅜ in.) Plastered[e]	0.61	0.25	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.24	5
Plywood (⅜ in.) Plain or Decorated	0.59	0.24	0.18	0.19	0.12	0.091	0.17	0.13	0.10	0.12	0.091	0.076	0.28	0.23	6
Insulating Board (½ in.) Plain or Decorated	0.36	0.19	0.15	0.16	0.10	0.082	0.14	0.12	0.097	0.10	0.082	0.069	0.22[h]	0.19[h]	7
Insulating Board Lath (½ in.) Plastered[e]	0.35	0.19	0.15	0.15	0.10	0.081	0.14	0.11	0.096	0.10	0.081	0.068	0.21	0.18	8
Insulating Board Lath (1 in.) Plastered[e]	0.23	0.15	0.12	0.12	0.089	0.072	0.12	0.097	0.084	0.089	0.072	0.061	0.16	0.14	9

[a] Coefficients corrected for framing on basis of 15 per cent area, 2 in. x 4 in. (nominal) framing, 16 in. on centers.

[b] ¾₂ in. yellow pine or fir.

[c] ¾₂ in. pine or fir sub-flooring plus ¹³⁄₁₆ in. hardwood finish flooring.

[d] Plaster assumed ¾ in. thick.

[e] Plaster assumed ½ in. thick.

[f] Based on insulation in contact with ceiling and consequently no air space between.

[g] For coefficients for constructions in Columns M and N (except No. 1) with insulation between joists, refer to Table 6. *Example*: The coefficient for No. 3-N of Table 11 is 0.24. With 2 in. blanket insulation between joists, the coefficient will be 0.093. (See Table 6.) (Column D of Table 6 applicable only for 3¾ in. joists.)

[h] For ²⁵⁄₃₂ in. insulating board sheathing applied to the under side of the joists, the coefficient for *single* wood floor (Column M) is 0.18 and for *double* wood floor (Column N) is 0.16. For coefficients with insulation between joists, see Table 6.

TABLE 12. COEFFICIENTS OF TRANSMISSION (*U*) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS

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 CEILING	Thickness of Concrete[a] (Inches)	TYPE OF FLOORING						NUMBER
		No Flooring (Concrete Bare)	Tile or Terrazo Flooring on Concrete	¼ In. Battleship[b] Linoleum Directly on Concrete	Parquet[c] Flooring In Mastic on Concrete	Double Wood Floor on Sleepers[d]		
		A	B	C	D	E		
No Ceiling	3	0.63	0.65	0.45	0.45	0.25		1
	6	0.59	0.56	0.41	0.41	0.23		2
	10	0.50	0.48	0.35	0.35	0.22		3
½ in. Plaster Applied to Underside of Concrete	3	0.62	0.59	0.43	0.43	0.24		4
	6	0.54	0.52	0.39	0.39	0.22		5
	10	0.46	0.44	0.34	0.34	0.21		6
Metal Lath and Plaster—Suspended or Furred	3	0.33	0.37	0.30	0.30	0.19		7
	6	0.35	0.34	0.28	0.28	0.18		8
	10	0.33	0.31	0.26	0.26	0.17		9
Gypsum Board (⅜ in.) and Plaster[e]—Suspended or Furred	3	0.36	0.35	0.28	0.28	0.19		10
	6	0.33	0.32	0.27	0.27	0.18		11
	10	0.30	0.29	0.24	0.24	0.17		12
Insulating Board Lath (½ in.) and Plaster[e] Suspended or Furred	3	0.25	0.24	0.21	0.21	0.15		13
	6	0.23	0.23	0.20	0.20	0.15		14
	10	0.22	0.21	0.19	0.19	0.14		15

[a] Thickness of tile assumed to be 1 in.

[b] Conductance of linoleum ¼ in. thick is 1.36. Values in Column C may be used with sufficient accuracy for concrete floors covered with carpet.

[c] Thickness of wood assumed to be ¹³⁄₁₆ in.; thickness of mastic, ⅛ in. ($k = 4.5$).

[d] Based on ²⁵⁄₃₂ in. yellow pine or fir sub-flooring and ¹³⁄₁₆ in. hardwood finish flooring with an air space between sub-floor and concrete.

[e] Thickness of plaster assumed to be ¾ in.

[f] Thickness of plaster assumed to be ½ in.

[g] For other thicknesses of concrete, interpolate.

TABLE 13. COEFFICIENTS OF TRANSMISSION (*U*) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING

$U = 0.10^{a}$ Btu per (hour) (square foot) (Fahrenheit degree temperature difference between the ground and the air over the floor).

[a] Until more complete data are available, it is recommended that a coefficient of 0.10 be used for all types of concrete floors on the ground, with or without insulation. For basement wall below grade, use the same average coefficient (0.10). A lower ground temperature should, however, be used for walls than floors as explained in Chapter 14. For further data see A.S.H.V.E. Research Report No. 1213—Heat Loss Through Basement Walls and Floors, by F. C. Houghten, S. I. Taimuty, Carl Gutberlet and C. J. Brown (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 369).

exact compliance with specifications, and a factor of safety may be employed as a precaution when it is judged desirable.

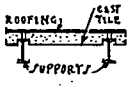
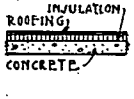
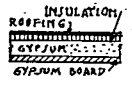
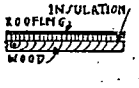
Roof Coefficients

Computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Values for roofs containing Spanish and French clay roofing tile are assumed the same as for slate roofs. Values for pitched roofs in Table 16 apply where the roof is over a heated attic or top floor so that the heat passes directly through the roof structure, including any interior finish material.

TABLE 14. COEFFICIENTS OF TRANSMISSION (*U*) OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. NO CEILING—UNDER SIDE OF ROOF EXPOSED

(See Table 15 for Flat Roofs with Ceilings)

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.

TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	NO INSULATION	INSULATION ON TOP OF DECK (COVERED WITH BUILT-UP ROOFING)							NUMBER
			INSULATING BOARD (THICKNESS BELOW)				CORKBOARD (THICKNESS BELOW)			
			½ In.	1 In.	1½ In.	2 In.	1 In.	1½ In.	2 In.	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck[a]		1.06	0.39	0.24	0.18	0.14	0.23	0.17	0.13	1
Precast Cement Tile 	1½ in.	0.84	0.37	0.24	0.17	0.14	0.22	0.16	0.13	2
Concrete 	2 in.	0.82	0.36	0.24	0.17	0.14	0.22	0.16	0.13	3
	4 in.	0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	4
	6 in.	0.65	0.33	0.22	0.16	0.13	0.21	0.15	0.12	5
Gypsum Fiber Concrete[b] on ½ in. Gypsum Board 	2½ in.	0.38	0.24	0.18	0.14	0.12	0.17	0.13	0.11	6
	3½ in.	0.31	0.21	0.16	0.13	0.11	0.15	0.12	0.10	7
Wood[c] 	1 in.	0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	8
	1½ in.	0.37	0.24	0.17	0.14	0.11	0.17	0.13	0.11	9
	2 in.	0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	10
	3 in.	0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	11

[a]Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per (hour)(square foot of projected area)(Fahrenheit degree difference in temperature) based on an outside wind velocity of 15 mph.

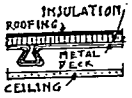
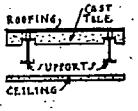
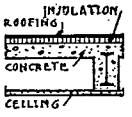
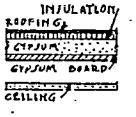
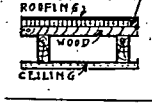
[b]87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ½ in. gypsum board.

[c]Nominal thicknesses specified—actual thicknesses used in calculations.

TABLE 15. COEFFICIENTS OF TRANSMISSION (*U*) OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING. WITH LATH AND PLASTER CEILINGS[a]

(See Table 14 for Flat Roofs with No Ceilings)

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.

TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	NO INSULATION	INSULATION ON TOP OF DECK (COVERED WITH BUILT-UP ROOFING)							NUMBER
			INSULATING BOARD (THICKNESS BELOW)				CORKBOARD (THICKNESS BELOW)			
			½ In.	1 In.	1½ In.	2 In.	1 In.	1½ In.	2 In.	
		A	B	C	D	E	F	G	H	
Flat Metal Roof Deck 		0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11	12
Precast Cement Tile 	1½ in.	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11	13
Concrete 	2 in.	0.42	0.26	0.19	0.14	0.12	0.18	0.14	0.11	14
	4 in.	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	15
	6 in.	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	16
Gypsum Fiber Concrete[b] on ½ in. Gypsum Board 	2½ in.	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097	17
	3½ in.	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091	18
Wood[c] 	1 in.	0.31	0.21	0.16	0.13	0.11	0.15	0.12	0.10	19
	1½ in.	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095	20
	2 in.	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092	21
	3 in.	0.18	0.14	0.12	0.10	0.087	0.11	0.095	0.082	22

[a]Calculations based on metal lath and plaster ceilings, but coefficients may be used with sufficient accuracy for gypsum lath or wood lath and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

[b]87½ per cent gypsum, 12½ per cent wood fiber. Thickness indicated includes ½ in. gypsum board

[c]Nominal thicknesses specified—actual thicknesses used in calculations.

TABLE 16. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS

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.

TYPE OF CEILING (APPLIED DIRECTLY TO ROOF RAFTERS)	WOOD SHINGLES (ON 1 x 4 WOOD STRIPS[a] SPACED 2 IN. APART)				ASPHALT SHINGLES OR ROLL ROOFING (ON SOLID WOOD SHEATHING)[a]				SLATE OR TILE[b] (ON SOLID WOOD SHEATHING)[a]				NUMBER
	INSULATION BETWEEN RAFTERS				INSULATION BETWEEN RAFTERS				INSULATION BETWEEN RAFTERS				
	None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			None	Blanket or Bat (Thickness Below)			
		1 In.	2 In.	3 In.		1 In.	2 In.	3 In.		1 In.	2 In.	3 In.	
	A	B	C[a]	D[a]	E	F	G[a]	H[a]	I	J	K[a]	L[a]	
No Ceiling Applied to Rafters	0.48[f]	0.15	0.10	0.081	0.52[f]	0.15	0.11	0.084	0.55[f]	0.16	0.11	0.085	1
Metal Lath and Plaster[d]	0.31	0.14	0.10	0.081	0.33	0.15	0.10	0.083	0.34	0.15	0.10	0.083	2
Gypsum Board (⅜ in.) Decorated	0.30	0.14	0.10	0.080	0.32	0.15	0.10	0.082	0.33	0.15	0.10	0.083	3
Wood Lath and Plaster	0.29	0.14	0.10	0.080	0.31	0.14	0.10	0.081	0.32	0.15	0.10	0.082	4
Gypsum Lath (⅜ in.) Plastered[e]	0.29	0.14	0.10	0.079	0.31	0.14	0.10	0.081	0.32	0.15	0.10	0.082	5
Plywood (⅜ in.) Plain or Decorated	0.29	0.14	0.099	0.079	0.30	0.14	0.10	0.081	0.31	0.15	0.10	0.081	6
Insulating Board (½ in.) Plain or Decorated	0.22	0.12	0.090	0.072	0.23	0.12	0.091	0.074	0.24	0.13	0.092	0.074	7
Insulating Board Lath (½ in.) Plastered[e]	0.22	0.12	0.088	0.072	0.22	0.12	0.090	0.073	0.23	0.12	0.091	0.074	8
Insulating Board Lath (1 in.) Plastered[e]	0.16	0.10	0.078	0.064	0.17	0.10	0.079	0.065	0.17	0.10	0.080	0.066	9

[a] Coefficients corrected for framing on basis of 15 per cent area, 2 in. x 4 in. (nominal), 16 in. on centers.
[b] Figures in Columns I, J, K and L may be used with sufficient accuracy for rigid asbestos shingles on wood sheathing. Layer of slater's felt neglected.
[c] Sheathing and wood strips assumed $^{25}/_{32}$ in. thick.
[d] Plaster assumed ¾ in. thick.
[e] Plaster assumed ½ in. thick.
[f] No air space included in 1-A, 1-E or 1-I; all other coefficients based on one air space.

TABLE 17. COMBINED COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a] AND HORIZONTAL CEILINGS—BASED ON CEILING AREA[b]

Coefficients are expressed in Btu per (hour) (square foot of ceiling area) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on an outside wind velocity of 15 mph.

CEILING COEFFICIENT (FROM TABLE 11)	TYPE OF ROOFING AND ROOF SHEATHING						NUMBER
	WOOD SHINGLES ON WOOD STRIPS[d]			ASPHALT SHINGLES[c] OR ROLL ROOFING ON WOOD SHEATHING[d]			
	No Roof Insulation (Rafters Exposed) (U_r = 0.48)	½ In. Insulating Board on Under Side of Rafters (U_r = 0.22)	1 In. Insulating Board on Under Side of Rafters (U_r = 0.16)	No Roof Insulation (Rafters Exposed) (U_r = 0.53)	½ In. Insulating Board on Under Side of Rafters (U_r = 0.23)	1 In. Insulating Board on Under Side of Rafters (U_r = 0.17)	
	A	B	C	D	E	F	
0.10	0.085	0.073	0.066	0.087	0.074	0.067	19
0.11	0.092	0.078	0.070	0.094	0.079	0.071	20
0.12	0.099	0.082	0.074	0.10	0.083	0.075	21
0.13	0.11	0.087	0.078	0.11	0.088	0.079	22
0.14	0.11	0.091	0.081	0.11	0.093	0.083	23
0.15	0.12	0.096	0.084	0.12	0.097	0.086	24
0.16	0.13	0.10	0.087	0.13	0.10	0.089	25
0.17	0.13	0.10	0.090	0.13	0.10	0.092	26
0.18	0.14	0.11	0.093	0.14	0.11	0.095	27
0.19	0.14	0.11	0.095	0.15	0.11	0.098	28
0.20	0.15	0.11	0.098	0.15	0.12	0.10	29
0.21	0.15	0.12	0.10	0.16	0.12	0.10	30
0.22	0.16	0.12	0.10	0.17	0.12	0.11	31
0.23	0.16	0.12	0.11	0.17	0.12	0.11	32
0.24	0.17	0.13	0.11	0.18	0.12	0.11	33
0.25	0.17	0.13	0.11	0.18	0.13	0.11	34
0.26	0.18	0.13	0.11	0.19	0.13	0.11	35
0.27	0.18	0.13	0.11	0.19	0.13	0.12	36
0.28	0.19	0.14	0.12	0.19	0.14	0.12	37
0.29	0.19	0.14	0.12	0.20	0.14	0.12	38
0.30	0.20	0.14	0.12	0.20	0.14	0.12	39
0.34	0.21	0.15	0.13	0.22	0.15	0.13	40
0.35	0.22	0.15	0.13	0.22	0.15	0.13	41
0.36	0.22	0.15	0.13	0.23	0.15	0.13	42
0.37	0.23	0.15	0.13	0.23	0.16	0.13	43
0.45	0.25	0.17	0.13	0.26	0.17	0.14	44
0.59	0.29	0.18	0.14	0.30	0.19	0.15	45
0.61	0.30	0.19	0.15	0.31	0.19	0.15	46
0.67	0.31	0.19	0.15	0.33	0.20	0.15	47
0.69	0.31	0.19	0.15	0.33	0.20	0.16	48

[a] Calculations based on ¼ pitch roof (n = 1.2) using the following formula:

$$U = \frac{U_r \times U_{co}}{U_r + \frac{U_{co}}{n}}$$

U = combined coefficient to be used with ceiling area.
U_r = coefficient of transmission of the roof.
U_{co} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

[b] Use ceiling area (not roof area) with these coefficients.
[c] Coefficients in Columns D, E and F may be used with sufficient accuracy for tile, slate and rigid asbestos shingles on wood sheathing.
[d] Based on 1 x 4 in. strips spaced 2 in. apart.
[e] Sheathing assumed $^{25}/_{32}$ in. thick.
[f] Values of U_{co} to be used in this column may be selected from Table 11.

TABLE 18. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS, SKYLIGHTS AND GLASS BLOCK WALLS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in the temperature between the air inside and outside of the door, window, skylight or wall) and are based on an outside wind velocity of 15 mph.

Section A. Windows and Skylights		SINGLE	DOUBLE	TRIPLE
	U	1.13[ac]	0.45[ae]	0.281[ae]

	NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U EXPOSED DOOR	U[d] WITH GLASS STORM DOOR
Section B. Solid Wood Doors[bc]	1	25/32	0.69	0.42
	1¼	1 1/16	0.59	0.38
	1½	1 5/16	0.52	0.35
	1¾	1 5/8	0.51	0.35
	2	1 5/8	0.46	0.32
	2½	2 1/8	0.38	0.28
	3	2 5/8	0.33	0.25

	DESCRIPTION	U STILL AIR BOTH SIDES	U STILL AIR INSIDE 15 MPH OUTSIDE
Section C. Hollow Glass Block Walls	Smooth surface glass blocks 7¾ x 7¾ x 3⅞ in. thick	0.40	0.49
	Ribbed surface glass blocks 7¾ x 7¾ x 3⅞ in. thick	0.38	0.46

[a] See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

[b] Computed using $C = 1.15$ for wood; $f_i = 1.65$ and $f_o = 6.0$.

[c] It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels as that of single panes of glass, namely, 1.13 Btu per (hour) (square foot) (degree difference between inside and outside air temperatures).

[d] These values may also be used with sufficient accuracy for wood storm doors. Neglect storm doors if loose and use values for exposed doors.

[e] Air spaces assumed to be ¾ in. or more in width.

Combined Ceiling and Roof Coefficients

If the attic space between ceiling and roof is unheated, the combined coefficient from room air below the ceiling to exterior air can be calculated from the following formula.

$$U = \frac{U_r \times U_{ce}}{U_r + \frac{U_{ce}}{n}} \qquad (5)$$

where

U = combined coefficient to be used with ceiling area.

U_r = coefficient of transmission of roof.

U_{ce} = coefficient of transmission of ceiling.

n = ratio of roof area to ceiling area.

It should be noted that the over-all coefficient U should be multiplied by the ceiling area to determine heat loss and not by the roof area. Values of U_r and U_{ce} should be calculated using a value of 2.2 (the reciprocal of one-half the air space resistance) rather than 1.65 for the conductances of surfaces facing the attic, since the attic is 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 large vertical wall areas in the attic are involved, it is preferable to estimate the attic temperature as illustrated in Chapter 14 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 [8] at the A.S.H.V.E. Research Laboratory indicate a heat flow of approximately 2.0 Btu per (hour) (square foot) through an uninsulated concrete basement floor with a temperature difference of 20 F between ground temperature and the air temperature 6 in. above the floor. Based on this result, a coefficient of 0.10 Btu per (hour) (square foot) (Fahrenheit degree temperature difference) is recommended for calculation where it is desirable to allow for the small basement floor heat loss, *e.g.* for heated basements.

For basement walls the same coefficient may be used, but due to closer proximity to the surface of the ground, 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, approximately twice that of the same floor area.

For concrete slab floors laid in contact with the ground at grade level, recent tests [9] indicate that for small floor areas (equal to that of a house 25 ft 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 (hour) (lineal foot of exposed edge) (Fahrenheit degree difference between the inside air temperature and the average outside air temperature). It should be noted that this may be appreciably reduced by insulating the edges of the floor from the abutting wall.

CONDENSATION IN BUILDINGS

Water vapor in the air within a building condenses if it comes in contact with surfaces at or below its dew-point temperature. It also will be transmitted into or through a wall, floor or ceiling, if a vapor pressure difference exists between the opposite sides, at a rate determined by the permeability of the materials encountered [10] (see Table 14 Chapter 15). Building practice must take account of these facts in avoiding (1) surface condensation on interior building surfaces (walls, ceilings, roofs or glass) and (2) interstitial condensation or accumulation of condensation in the voids within the structure. The conditions under which surface condensation will take place are directly dependent on surface temperature and upon the relative humidity of the air in contact. Limiting maximum relative humidities for walls, roofs or glass having transmission coefficients up to 1.2 Btu for outside temperatures from −30 F to 40 F and for 70 F inside temperature may be obtained from Fig. 4 [11].

Surface condensation may be controlled by air conditioning, ventilation for the purpose of removing water vapor, particularly from laundries

and from kitchens during lengthy periods of cooking, elimination of sources of vapor such as unvented gas stoves, or by local application of heat or insulation to raise surface temperatures.

Interstitial condensation results from vapor transmission which is dependent on vapor pressure differential and the ratio of the rate at which vapor may enter the materials of the structure to that at which it passes out of the structure. The proper installation of a vapor barrier on the warm side of the structure, where the higher vapor pressure usually exists, will reduce greatly the amount of vapor entering the building construction and will minimize the possibility of objectionable condensation. Such a barrier may consist of a coated paper applied

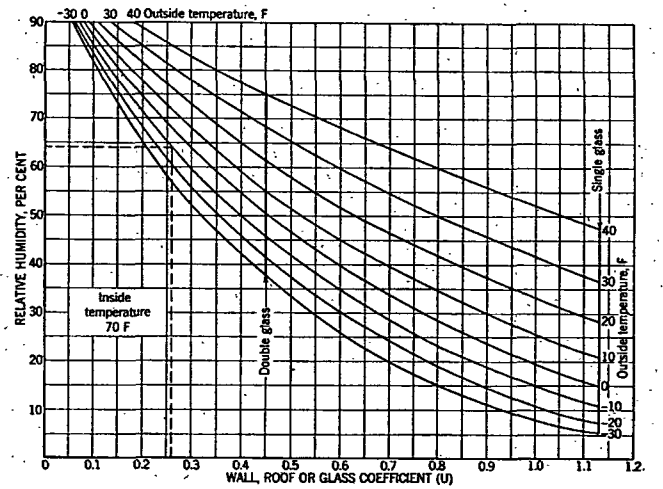


FIG. 4. PERMISSIBLE RELATIVE HUMIDITIES FOR VARIOUS TRANSMISSION COEFFICIENTS

under the plaster, a coated plaster base, or a vapor-resistant finish applied to the interior wall surface [12]. A vapor barrier is tentatively defined as a material having a water vapor permeability not exceeding 1.25 grains per (hour) (square foot) (inch of Hg vapor pressure differential).

To prevent condensation on the under side of roofs above attic spaces over insulated ceilings, ventilation with outside air may be provided through suitable louvers or other roof ventilators.

REFERENCES

1—Standard Method of Test for Thermal Conductivity by Means of the Guarded Hot Plate, sponsored by A.S.H.V.E., *A.S.T.M.*, *A.S.R.E.*, and *N.R.C.* and approved as a Tentative Code by A.S.H.V.E. and *A.S.T.M.* in 1942 (*A.S.T.M.* designation C-177-42T).

2—Heat Transmission Through Building Materials, by F. B. Rowley and A. B. Algren (University of Minnesota, *Engineering Experiment Station Bulletin* No. 8, p. 11).

3—Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 513).

4—Radiation Corrections for Basic Constants Used in the Design of All Types of Heating Systems, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December 1944, p. 705).

5—Radiation and Convection Across Air Spaces in Frame Construction, by G. B. Wilkes and C. M. F. Peterson (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 351).

6—Thermal Test Coefficients of Aluminum Insulation for Buildings, by G. B. Wilkes, F. G. Hechler and E.R. Queer (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 109).

7—Effect of Studs and Joists on Heat Flow Through Frame Walls and Ceilings, by Paul D. Close (*Heating, Piping and Air Conditioning*, October, 1943, p. 529).

8—A.S.H.V.E. RESEARCH REPORT No. 1213—Heat Loss Through Basement Walls and Floors, by F. C. Houghten, S. I. Taimuty, Carl Gutberlet and C. J. Brown (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 369).

9—Measurements of Heat Losses from Slab Floors, by R. S. Dill, Wm. C. Robinson and H. E. Robinson (*National Bureau of Standards*, Building Materials and Structures Report BMS 103).

10—Methods of Moisture Control and Their Application to Building Construction, by F. B. Rowley, A. B. Algren and C. E. Lund (University of Minnesota *Engineering Experiment Station Bulletin* No. 17).

Moisture Migration: A Survey of Theory and Existing Knowledge, by P. F. McDermott (*Refrigerating Engineering*, August 1941, p. 103).

11—Permissible Relative Humidities in Humidified Buildings, by Paul D. Close (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1939, p. 766).

12—Condensation Within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 95).

BIBLIOGRAPHY

A.S.H.V.E. RESEARCH REPORTS:

No. 852—Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 123).

No. 895—Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 301).

No. 914—Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 33).

No. 915—Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 47).

No. 964—The Heat Conductivity of Wood at Climatic Temperature Differences, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 329).

No. 996—Insulating Value of Bright Metallic Surfaces, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 413).

No. 1026—Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Clifford Carlson (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 33).

No. 1048—Thermal Properties of Concrete Construction, by F. B. Rowley, A. B. Algren and Robert Lander (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 33).

Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 285).

Condensation of Moisture and Its Relation to Building Construction and Operation, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 231).

A Theory Covering the Transfer of Vapor Through Materials, by F. B. Rowley (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 545).

Thermal Conductivity of Wood, by J. D. MacLean (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 323).

The Specific Heat of Thermal Insulating Materials, by G. B. Wilkes and C. O. Wood (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 493).

Heat Loss Studies in Four Identical Buildings to Determine the Effect of Insulation, by D. B. Anderson (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 471).

Simultaneous Heat and Vapor Transfer Characteristics of an Insulating Material, by F. G. Hechler, E. R. McLaughlin and E. R. Queer (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 505).

Comparative Resistance to Vapor Transmission of Various Building Materials, by L. V. Teesdale (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 124).

The Diffusion of Water Vapor Through Various Building Materials, by J. D. Babbitt (*Canadian Journal of Research*, Vol. 17, February, 1939, p. 15).

Effect of Ceiling Insulation upon Summer Comfort, by T. D. Phillips (*National Bureau of Standards*, Report BMS52, July 1, 1940).

Moisture Condensation in Building Walls, by H. W. Wooley (*National Bureau of Standards*, Report BMS63, December 14, 1940).

Heat Transmission Through Building Materials, by F. B. Rowley and A. B. Algren, (University of Minnesota *Engineering Experiment Station Bulletin* No. 8).

Building Insulation, by P. D. Close (*American Technical Society*, Chicago, 1945).

CHAPTER 7

Heat Transfer Surface Coils

Construction and Arrangement, Steam Coils, Water Coils, Direct-Expansion Coils, Flow Arrangement, Applications, Heat Transfer and Air Flow Resistance, Performance of Heating, Dry Cooling and Dehumidifying Coils, Coil Selection

THE coils described in this chapter are 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 on each installation.

For heating service, coils are used as 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 to remove sensible heat in connection with chemical moisture-absorption apparatus. By proper coil selection it is possible to handle both sensible cooling and dehumidification together as explained later. The assembly usually includes an 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, there are cases of cooling coils 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 water becomes impractical. 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 a certain degree of 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 the 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. The safety of the occupant must be kept in mind in comfort conditioning applications. Some localities have refrigeration codes which restrict the use of direct-expansion coils in the air stream, and hence local codes should be consulted by the engineer before a system employing direct expansion methods is designed. The choice between spray dehumidifiers and coils depends upon the necessities and the economic aspects of each case and no general rule can be given. There are many installations in which either may be used.

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 resistance is from the air to the surface of the tube, usually called the outside surface resistance or air-film resistance. Second is the resistance to the flow of heat by conduction 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 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 over-all heat transfer is not necessarily increased much by such an arrangement, 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. In some cases these are staggered and in others in line with respect to the air flow. The staggered arrangement gives a somewhat higher heat transfer value but also a higher resistance to air flow and in some cases makes the header and return bend arrangement more complicated. 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 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 is assured in a number of ways. The assembled coil may be coated with tin, zinc, etc., after fabrication. The spiral type 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 the materials most generally used are copper, steel and aluminum. Sometimes aluminum or brass fins are used on copper tubes. Some types of heating coils are made of cast-iron. For equal performance brass and aluminum fins must be of greater thickness than copper fins on account of their lower thermal conductivities. Copper coils are frequently tin-dipped and steel coils galvanized to protect them from corrosion and to assure a bond between fin and tube.

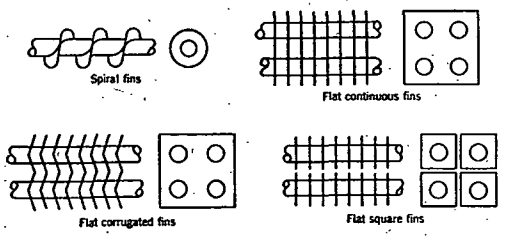


FIG. 1. TYPES OF FIN COIL ARRANGEMENT

Cooling coils for water or for volatile refrigerants most frequently have copper fins and tubes, although aluminum fins on copper tubes are also used. For brines such as sodium or calcium chloride and for ammonia, steel fins and tubes are common.

Although there are many variations for special cases, tube and fin sizes and spacings for air conditioning coils, both heating and cooling, fall within fairly narrow limits. The tubes are usually $\frac{3}{8}$, $\frac{1}{2}$, $\frac{5}{8}$, or $\frac{3}{4}$ in. OD, and the fins spaced from 4 to 8 per inch, 7 per inch being a common design. The tube spacing generally varies from about $1\frac{1}{8}$ to 2 in. on centers. Small tube size and close fin spacing give large capacity with small space demand, but the resistance, both over the surface and through the tubes, is higher than with larger tubes and more widely spaced fins. Moreover, too close a fin spacing may result in trouble from dirt accumulation, especially on dehumidifying coils, and may also cause trouble from water held between the fins, particularly with air flow vertically upward. This condition increases the air resistance and decreases the capacity of the coil. Water hold-up sometimes causes flooding trouble in vertical air flow units by accumulating too much water for the drain to handle all at once when the fan is stopped.

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

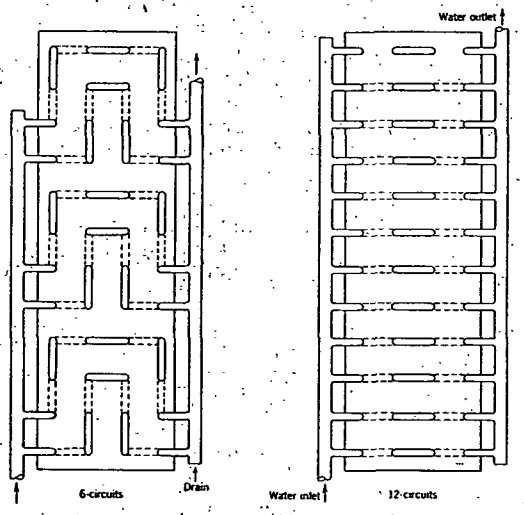


FIG. 2. VARIOUS WATER CIRCUIT ARRANGEMENTS

heating loads is also minimized. This is especially valuable for outside air preheaters. Methods of air and condensate elimination are discussed in Chapters 23 and 26.

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 24. 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 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 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 if located where they will be exposed to freezing. Fig. 2 shows such construction. The drains may be provided in the water piping although they are often arranged in the coil headers.

Direct-Expansion Coils

Coils for volatile refrigerants present more complex problems of fluid distribution than do water, brine or steam. 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 auto-

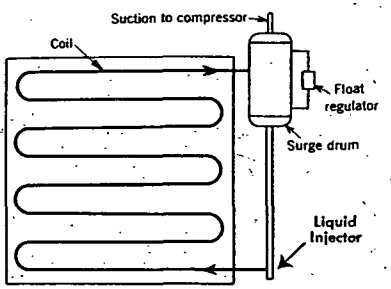


FIG. 3. DIRECT-EXPANSION COIL WITH FLOODED SYSTEM

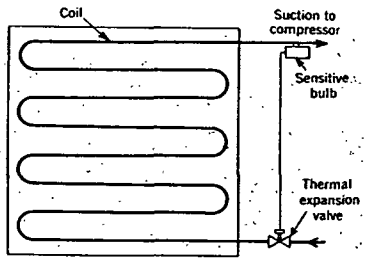


FIG. 4. DIRECT-EXPANSION COIL WITH THERMAL VALVE SYSTEM

matically feeding 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 10 deg. The thermal valve arrangement is in common use for the type of coils covered by this chapter, while the flooded system is comparatively rare.

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. No auxiliary distributing devices are required. With the thermal valve system there are two factors to consider. There must be, generally, more than one refrigerant feed through the coil per thermal valve to keep the pressure drop through the refrigerant circuit within practical limits and to reduce the corresponding penalty in increased evaporating temperature. At the same time the coil must be so arranged that the required suction superheat can be attained with a minimum sacrifice in the performance of the coil as a whole. It is general practice to attain this superheat within the coil itself and not by the use of external heat exchangers or other auxiliary devices.

With thermal expansion valves it is advantageous to keep the pressure drop through the refrigerant feeds as low as possible. The feeds are laid out to expose each to the same mean temperature difference so that it

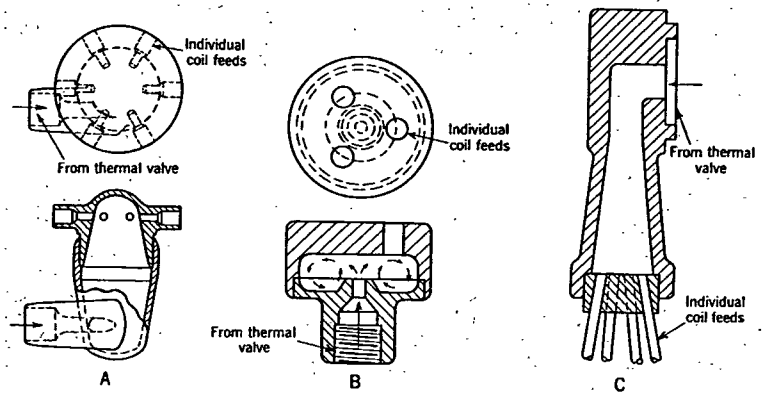


FIG. 5. TYPES OF REFRIGERANT FEED DISTRIBUTING HEADS

handles the same refrigerating load. A distributing means is imposed 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

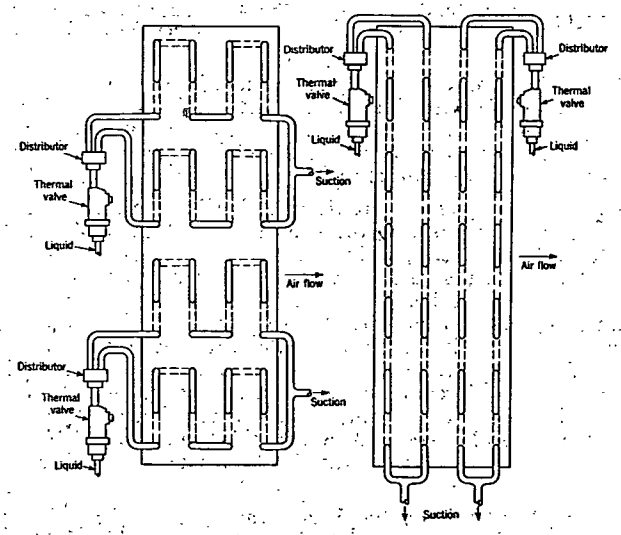


FIG. 6. ARRANGEMENT FOR FACE CONTROL

FIG. 7. ARRANGEMENT FOR DEPTH CONTROL

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. Although there are other forms of distributors those mentioned are typical examples. 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.

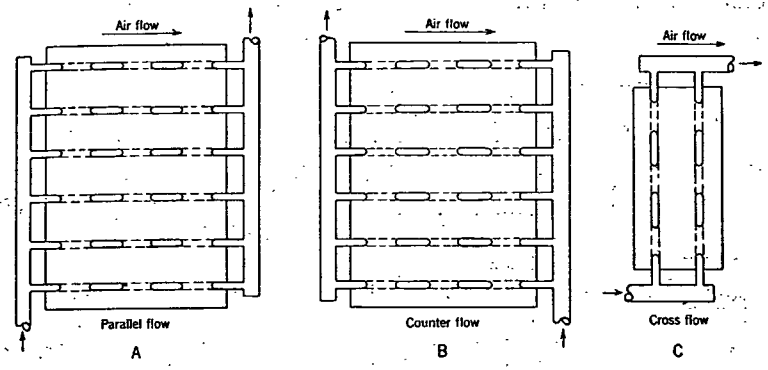


FIG. 8. FLOW OF MEDIA IN TUBES IN RELATION TO AIR FLOW

The arrangement of Fig. 7 should be avoided since it offers the disadvantage of unequal load on the two parallel circuits.

Flow Arrangement

The relative direction of flow of the air outside the tubes and the medium within them influence the performance of the surface. There are three types of relative flow in common use. Fig. 8A shows parallel-flow in which the air and the medium in the tubes proceed through the coil in the same direction. Fig. 8B shows counter-flow in which the medium in the tubes proceeds in a direction opposite to the flow of air. Fig. 8C shows cross-flow in which the air and the medium in the tubes pass at right angles to each other. The counter-flow arrangement is almost universally used in brine or water coils to take advantage of the highest possible mean temperature difference for given entering water and air temperatures. It is also commonly used in coils fed with volatile refrigerant to take advantage of the higher air temperature for superheating the leaving gas. In deep coils, however, it is sometimes advantageous to use parallel flow from second row to last row and then to complete the circuit by passing through the first row to take advantage of the higher air temperature for superheating. Complete evaporation and superheating of the refrigerant are essential to proper operation of the thermal expansion valve. Cross-flow is common in steam 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. Cross-flow is to be avoided in coils with volatile refrigerants on account of unequal loading of parallel circuits and the danger of short circuiting of liquid refrigerant which disturbs proper functioning of the thermal expansion valve.

Applications

Heating coils in field assembled banks are used for a number of purposes as described in Chapter 43. 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 neces-

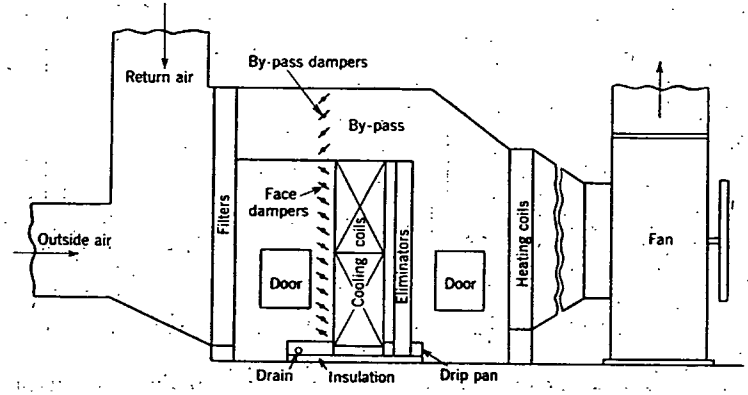


FIG. 9. TYPICAL ARRANGEMENT OF COOLING COILS IN A CENTRAL SYSTEM

sity 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 result in freezing 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. If these are used, there is little danger of freezing the condensate as long as the leaving air temperature is not allowed to fall below about 40 F. As an added precaution with both steam and water coils the outside air inlet dampers are often closed automatically when the fan is stopped to avoid trouble caused by very cold outside air drifting in during *off* periods.

A typical arrangement of cooling coils is shown in Fig. 9. Some means should be provided to filter all the entering air to keep dirt and foreign matter from accumulating on the coils. The assembly is provided with a drip-pan to catch the condensate during summer dehumidifying duty and to collect the non-evaporated water from the humidifying sprays in winter. The drip connection should be made ample in size and liberally provided with plugged tees and crosses for cleaning. It should not be

exposed to freezing temperatures in winter if the apparatus is used on winter humidifying duty. Access doors should be provided for servicing filters, humidifying nozzles, and fan bearings and for cleaning the coils. With certain designs of coils when used for dehumidifying, eliminators must be used beyond the coil to catch any water which may be blown into the air stream. It is customary to include these eliminators when the air velocity exceeds about 450 fpm with the individual fins and about 600 fpm for the continuous flat fin type. Where a number of coil sections are stacked one upon another, and where the velocities are low, so that eliminators need not be used, occasional trouble results when water splashes down from one coil to the next and blows out into the air stream. In such cases drip troughs as shown in Fig. 10 are used to collect this water and conduct it to the condensate pan.

Sometimes finned surface coils on summer cooling and dehumidifying

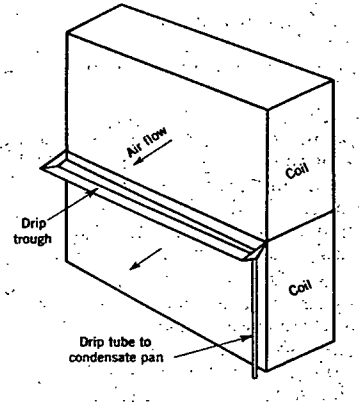


FIG. 10. COIL ARRANGED WITH DRIP TROUGH

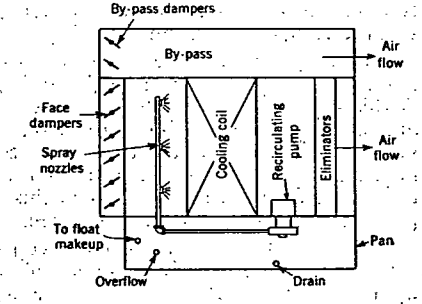


FIG. 11. RECIRCULATING SPRAY SYSTEM FOR CLEANING COILS

duty are provided with water sprays. These sprays are of two types. In the first type a set of spray nozzles is arranged for intermittent cleaning. These sprays are not operative when the system is in use and no recirculating pump is provided. The second arrangement requires a collecting tank and a recirculating pump. The water is in circulation whenever the apparatus is in operation, and assists in keeping the coil clean and in absorbing odors. Fig. 11 illustrates such an arrangement. Wherever air by-passes are used around a coil on summer duty for control purposes, it is advantageous to direct only return air through the by-pass rather than a mixture of return and outside air. The casing should be arranged accordingly. To maintain the air quantity handled by the fan reasonably constant, and to assure the required design quantity of by-passed air when the by-pass damper is open, cooling coil banks are frequently furnished with both face and by-pass dampers as shown in Fig. 9.

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 to the coil connections. (See Chapter 23.)

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. For combined cooling and dehumidification, the logarithmic difference does not apply strictly and such problems should be handled as described later. 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 spite of this, heat transfer and ratings for coils using volatile refrigerants are usually based on a refrigerant temperature corresponding to the average pressure in the coil.

The design and surface arrangement of the coil includes 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 as against flat fins. Staggered tubes increase the total heat transfer as against the in-line arrangement and corrugated fins may be more effective than flat. It is this design and surface arrangement that determines 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 often based on standard air at 70 F and a barometric pressure of 29.92 in. Hg. The use of air volume in coil *rating* information may be misleading. *The significant value is mass velocity in pounds per (minute) (square foot of face area) and not cubic feet per minute*, because for a fixed volume the corresponding weight may vary widely, depending upon the temperature and barometric pressure.

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 practical installations. The 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 the coils. This reduces the capacity, but can be avoided by proper layout or by the use of directing baffles.

Heat transfer depends also upon the velocity of the medium in the tubes and upon its character, whether flowing water, condensing steam or evaporating volatile refrigerant. Heat transfer rates expressed as Btu per (square foot of internal surface) (degree logarithmic mean effective temperature difference between the fluid and tube wall) are, for example:

about 150 to 300 for evaporating dichlorodifluoromethane, about 350 to 1200 for water at 2 and 6 fps and about 1200 for condensing steam. The influence of the medium in the tubes on the over-all heat transfer rate is, therefore, apparent.

Because of these variables, reliable rating and performance information for any design of coil must be based on actual tests on that coil under the expected conditions of operation. A comparison between the performance of two designs, unless based on such tests on each, may lead to entirely erroneous conclusions.

PERFORMANCE OF HEATING AND DRY COOLING COILS

Heating and dry cooling coils are heat exchangers and as such their performance depends in general upon:

1. The over-all 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:

$$q_t = U \times (MTD) \times A \times N \qquad (1)$$

where

q_t = total heat transferred by the coil, Btu per (hour) (square foot of coil face area).

U = over-all 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).

MTD = 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, per row of coil depth.

N = number of rows of coil depth.

Over-all Coefficient of Heat Transfer

Of all factors affecting the performance of heating or dry cooling coils, the over-all coefficient of heat transfer is the most difficult to determine as it is influenced by several factors which depend upon coil design and conditions of operation.

Considering any coil, whether of bare pipe or of finned type, the over-all 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 coefficient of heat transfer through the coil material—tube wall, fins, ribs, etc.
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 over-all coefficient of heat transfer in accordance with the basic laws. For a *bare pipe* coil the over-all coefficient of heat transfer, whether for heating or for

cooling (dry), can be expressed by a simplified basic formula as follows:

$$U = \frac{1}{\frac{R}{h_r} + \frac{L}{k} + \frac{1}{h_a}} \qquad (2)$$

where

U = over-all coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and fluid within the coil).

h_r = 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).

h_a = 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 per cent of the over-all coefficient). Thus, in its simplest form, for bare pipe:

$$U = \frac{1}{\frac{R}{h_r} + \frac{1}{h_a}} \qquad (3)$$

For finned coils the formula[1] for the over-all coefficient of heat transfer can be conveniently written:

$$U = \frac{1}{\frac{R}{h_r} + \frac{1}{\eta_f h_a}} \qquad (4)$$

in which the term η_f, 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. In the discussions which follow, coefficients h_r and $\eta_f h_a$ will be considered separately, and also various ways of combining them will be outlined.

The performances of all heating and dry cooling coils are influenced by the same factors. But, when cooling coils operate wet or act as dehumidifying coils, the performance cannot be predicted on the basis of over-all coefficients and an analysis must be made on the basis of individual film coefficients as will be explained.

PERFORMANCE OF DEHUMIDIFYING COILS

When a cooling coil operates with a surface temperature which is below the dew-point of the air entering the coil, moisture is condensed and the air leaves the coil with a humidity ratio lower than it had when

it entered the coil. To understand the performance of surface coils under such conditions, assume that air enters a cooling coil at conditions corresponding to point 1 in Fig. 12. As long as the surface temperature of the coil is above the dew-point, the air is cooled without dehumidification, and its condition leaving the coil will be somewhere on line 1-A. Its exact position on this line depends on the air velocity and the external film coefficient as well as upon the surface temperature. When the surface temperature just equals the dew-point, the air leaves with conditions represented by point A. If the surface temperature is below the dew-point, condensation takes place, and the air has a final condition somewhere along the line A-2-3 which is a line at a constant horizontal distance from the saturation curve. It should be understood that the line 1-A-2-3 is not intended to represent the path of the condition of the air as it passes through the coil from row to row. It is simply the path traced

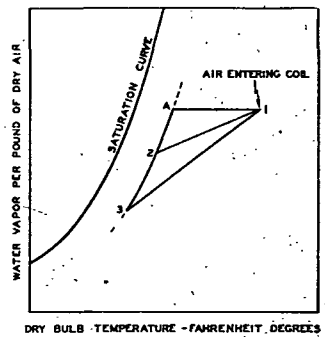


FIG. 12. PERFORMANCE OF DEHUMIDIFYING COIL

by the exit air conditions as the surface temperature is gradually reduced with other conditions remaining constant[2].

In the process of dehumidification, since heat is being transferred to the coil surface by two different mechanisms, (convection and condensation), it is evident that an over-all coefficient of heat transfer cannot be determined by the same method used for heating and for dry cooling coils. However, if it is assumed that the sensible heat transfer of a dehumidifying coil is unaffected by the presence of moisture on its surface, Equation 5 may be obtained to express this part of the heat transfer in terms of the external film coefficient and the surface temperature.

$$q_s = h_a \times A \times N \times (MTD_a) \qquad (5)$$

where

q_s = sensible heat transferred, Btu per (hour) (square foot of coil face area).

t_1 = dry-bulb temperature of air entering coil, Fahrenheit degrees.

t_2 = dry-bulb temperature of air leaving coil, Fahrenheit degrees.

t_s = average temperature of coil external surface, Fahrenheit degrees.

MTD_a = logarithmic mean temperature difference between air and coil surface $= \frac{t_1 - t_2}{\log_e \frac{t_1 - t_s}{t_2 - t_s}}$

If this equation is combined with another equation expressing sensible heat transfer in terms of mass velocity and temperature difference, the variables may be arranged in the following form (which is useful for the solution of dehumidification problems and for the determination of h_a from test data):

$$\frac{h_a A N (t_1 - t_2)}{\log_e \frac{t_1 - t_s}{t_2 - t_s}} = 0.243 G (t_1 - t_2)$$

or,

$$\frac{h_a A N}{0.243 G} = \log_e \frac{t_1 - t_s}{t_2 - t_s} \quad (6)$$

where

0.243 = specific heat of humid air, Btu per (pound) (Fahrenheit degree).
G = air mass velocity, pounds per (hour) (square foot of coil face area).

An examination of Fig. 12 will reveal that when t_s is at the dew-point of the entering air:

$$\frac{t_1 - t_s}{t_a - t_s} = \frac{t_1 - t_{dp1}}{t_a - t_{dp1}}$$

and when t_s is below the dew-point:

$$\frac{t_1 - t_s}{t_2 - t_s} = \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$$

Therefore, Equation 6 may be written in its most useful form as:

$$\frac{h_a A N}{0.243 G} = \log_e \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \log_e \frac{t_1 - t_s}{t_2 - t_s} \quad (7)$$

where

t_a = minimum dry-bulb possible without dehumidification, Fahrenheit degrees.
t_{dp1} = dew-point of air entering coil, Fahrenheit degrees.
t_{dp2} = dew-point of air leaving coil, Fahrenheit degrees.

This equation may be used to establish a line as A-2-3 for a given coil if h_a is known for the coil, or it may be used to determine h_a from test data for the purpose of rating coils. The use of this equation for coil selection is illustrated in Example 1 at the end of the chapter. Equation 7 is also important as a means of determining the external film coefficient.

External Film Coefficient

While formulae have been developed expressing the film coefficient h_a for air passing parallel to a plane surface, they cannot be used directly for fins on tubes because of air turbulence and because of the temperature gradient prevalent from the edge of a fin to its center. It is therefore necessary to make tests to evaluate the combined term $\eta_f h_a$. The term, $\eta_f h_a$, will be written merely h_a in this discussion as there is no necessity for separately evaluating η and because values of h_a are usually applied only to the particular coils for which tests are made.

The air side coefficient, h_a, of a coil of particular dimensions is an exponential function of the mass velocity of the air:

$$h_a = Z G^n \quad (8)$$

where

h_a = film coefficient of heat transfer, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and average surface temperature).

G = air mass velocity, pounds per (hour) (square foot of coil face area).
Z and n = constants which depend upon both air turbulence and surface arrangement.

Evaluation of constants Z and n may be accomplished through the use of test data in Equation 7 which gives values of h_a directly from the results of any *wet coil test.* If h_a, calculated in this manner, is plotted against values of G which prevailed during the tests a straight line should result on logarithmic coordinates. The slope of this line is the value of n. The value of Z may then be determined by direct substitution in Equation 8.

Internal Film Coefficient

The internal film coefficient, h_r which appears in Equation 3, is evaluated in various ways, depending upon the nature of the fluid, and whether the fluid is changing state.

When evaporating refrigerants are used in tubes, the temperature of the fluid is fairly constant, being affected principally by pressure drop through the tubes, by superheat of the evaporated refrigerant, and by the presence of oil in solution. To obtain maximum coil capacity it is necessary to keep the pressure drop through the tubes at a minimum (¼ lb per square inch), to keep the superheat as low as possible without carrying liquid back to the compressor, and to arrange for good separation and return of oil to the compressor. Another important factor is the removal of gas to keep the tube surface flooded with liquid as much as possible. The internal film coefficient is markedly increased by heavy heat loads, because the increased turbulence and gas velocity cause good contact of the liquid with the tubes. Values of h_r usually lie between 150 and 450. For rating of dehumidifying coils, satisfactory results are obtainable by first determining the average external surface temperature from Equation 7, and then using the difference between the external film temperature and the refrigerant for evaluating h_r in Equation 9.

$$h_r = \frac{q}{\frac{A N}{R} (t_s - t_r)} \quad (9)$$

where

h_r = internal film coefficient of heat transfer, Btu per (hour) (square foot of internal tube surface) (Fahrenheit degree).
t_r = average refrigerant temperature, Fahrenheit degrees.

The term $(t_s - t_r)$ is commonly written Δt. To evaluate h_r by this method the same tests that were required to determine h_a may be used.

When water is the cooling medium in tubes, the rate of heat transfer is a function of its velocity, which influences the number of contacts of the water molecules with the tube surface, per unit of time. Increased water velocity and reduced tube diameter cause increased heat transfer. Heat transfer is also greater at higher temperatures of the water. The basic formula for the film coefficient of heat transfer for flow of water is as follows:

$$h_r = 1.5 (t + 100) \frac{V^{0.8}}{D^{0.2}} \quad (10)$$

where

V = water velocity, feet per second.
D = internal diameter of tube, inches.
t = average water temperature, Fahrenheit degrees.

In the case of finned tubes, values of h_t may be lower than those obtained by use of Equation 10 and actual test results are preferred if available.

When saturated steam is condensed in the tubes of coils, the film coefficient h_r varies from 1000 to 2000, depending on freedom from air in the steam, and upon good drainage of the tubes. The coefficient is fairly constant for a particular coil, giving values of Δt that are directly proportional to q_t.

Performance of Coils and Refrigeration Compressor

Practically all data published by various makers of direct expansion cooling coils are based upon maintaining a predetermined refrigerant temperature within the coils. While it is often possible to maintain a

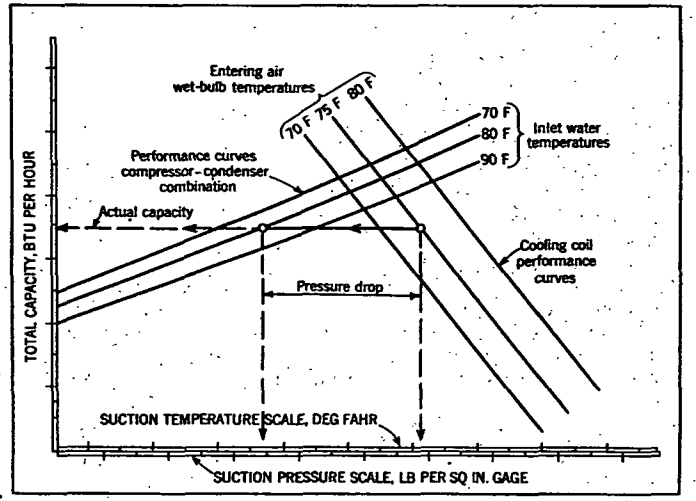


FIG. 13. GRAPHICAL ANALYSIS OF COIL-COMPRESSOR PERFORMANCE

definite refrigerant temperature within a given cooling coil, for the greater part it is either impossible or impractical. This is due to the fact that the capacity of standard refrigeration compressors is usually fixed and in matching a given cooling coil and a standard compressor the capacity of the latter is often somewhat smaller or greater than that of the former. Consequently, the refrigerant temperature resulting within a cooling coil and correspondingly the capacity of the coil-compressor combination are very often not what they were originally calculated to be.

In order to determine the actual performance of a given coil-compressor combination under varying conditions of operation, a graphical solution of the balance point is highly desirable. A typical procedure for graphical analysis of a coil-compressor combination performance is shown in Fig. 13, which is constructed as follows:

1. The equipment capacity scale, total Btu per hour, is laid out along the vertical axis while the refrigerant suction temperature scale is laid out along the horizontal axis using rectangular coordinates.

2. The performance curve of a given compressor with a definite condenser (combination usually called a *condensing unit*) is plotted as a function of suction temperature corresponding to the saturation suction pressure at the compressor suction service valve for a given inlet water temperature and quantity supplied to the condenser.

3. The performance curve of the given cooling coil is next plotted as a function of mean suction temperature within the coil, and the wet-bulb temperature of air entering the coil, for a given mass air velocity over the coil.

4. The refrigerant pressure drop between the center of the cooling coil and the compressor suction service valve is computed and converted into the terms of temperature difference. This temperature difference is then fitted in horizontally between the performance curves of the cooling coil and the compressor, as shown, and the total capacity of the coil-compressor combination is read along the horizontal line upon which the above mentioned *temperature-difference* segment falls.

COIL SELECTION

In the selection of a coil it is necessary to consider several factors:

1. The duty required—heating, cooling, dehumidifying.
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.
6. Allowable resistances in air circuit and through tubes.
7. Peculiarities of individual designs of coils.
8. Individual installation requirements, such, for example, as type of automatic control to be used.

The duties required may be determined from information in Chapters 6, 8, 14 and 15. There may or may not be a choice of cooling and heating media, as well as temperatures available, 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 fixed by the use of old ventilating ducts as an air distribution system for new air conditioning apparatus, or may be dictated by requirements of 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 by the maximum fan peripheral velocity which requirements of quietness may permit. The friction through the water or brine circuit may be dictated by the head available from a given size of pump and pump motor. As the fan and pump motor inputs represent a refrigerating load on cooling installations, it is economical to keep them low.

Proper performance of a surface heating or cooling coil depends upon correct choice of the original equipment and upon certain other factors. The usual coil ratings are based on a uniform face velocity of air. If the air is brought in at odd angles or if the fan is located so as to block part of the air flow, the performance as given in the manufacturer's ratings cannot usually be obtained. To obtain this performance it is necessary also that the air quantity be adjusted on the job to that used in determining the coil selection, and must also be kept at this value. The most

common causes for a reduction of air quantity are the fouling of the filters and collection of dirt in the coils. These difficulties can be avoided by proper design and proper servicing. There are a number of ways in which coils may be cleaned. A common method is to wash them off 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 most satisfactory solution, however, is to keep the filters serviced, and thus make the cleaning of the coils unnecessary.

The proper selection of coils requires an understanding of the necessities of each case and should be based on an economic analysis of the plant design as a whole. No general rule can, therefore, be laid down for the selection of heating or cooling coils. It is possible, however, 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 lb, sometimes up to 350 lb per square inch.
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 lb, 5 lb being common.
Hot Water Temperature—150 to 225 F.
Water Velocity—2 to 6 fps.
Water Quantity—Based on about 20 F temperature drop through a hot-water coil.
Air Resistance—The total resistance through heating coils is usually limited to from $\frac{3}{8}$ to $\frac{5}{8}$ in. of water gage 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 per cent of total, depending on the application. (See Chapter 43.) 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 usual comfort installations air face velocities between 400 and 600 fpm are frequent, 500 being a common value. Refrigerant temperatures ordinarily vary between 40 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 selection of coils for combined cooling and dehumidifying duty is more involved than for heating or sensible cooling and requires consideration of both dry- and wet-bulb air temperatures. It is further complicated by the fact that the proportional amount of dehumidification required is also highly variable. The methods outlined previously under Heat Transfer and Resistance may be used to determine whether it is possible for a coil to perform the duty required. If entering and leaving air conditions are arbitrarily specified, the corresponding duty sometimes cannot be obtained at all without the use of reheat. As with heating and sensible cooling coils, there are combinations of face areas, depth, air velocity and refrigerant temperatures which will give the required performance. This is illustrated in Table 1.

TABLE 1. VARIOUS COOLING COIL ARRANGEMENTS

SELECTION	1	2	3	4
Total cooling capacity, tons	100	100	100	100
Sensible cooling capacity, tons	69	69	69	69
Latent cooling capacity, tons	31	31	31	31
Ratio total to sensible heat	1.45	1.45	1.45	1.45
Air quantity, cfm	47,800	41,700	37,100	46,800
Cfm per total ton	478	417	371	468
Face velocity, fpm	325	423	500	600
Resistance, in. water	0.11	0.27	0.51	0.37
Coil face area, sq ft	147	99.0	74.2	78.1
Coil rows deep	4	6	8	4
Coil evaporator temp. F deg	45	45	45	38

It is possible as shown in Table 1 to perform approximately the same duty at a given refrigerant temperature with small face area and large thickness or vice versa. The large face area coil gives low air velocity and resistance but high air quantities per ton. The coil of small face area and great depth requires small air quantities per ton of refrigeration, high resistance and high air velocities. As shown also in Table 1 the same sensible, latent and total cooling capacity may be obtained with various refrigerant temperatures by proper choice of coil. This makes it possible to keep the evaporating temperature high enough to carry the load with a chosen size of condensing unit. High evaporating temperatures with

correspondingly small compressor operating expense can be attained but at the expense of coil surface, air quantity or both. The choice will be determined by the necessities of individual installations.

For a given quantity and condition of entering air the evaporating temperature of a volatile refrigerant coil is determined by a balance between the condensing unit and the coil. The total, sensible and latent cooling capacity can then be determined from the coil rating information. If the condensing unit and cooling coil have been properly balanced for the required load and, due to miscalculated duct resistance or improper choice of fan speed, the air quantity is reduced, the total cooling capacity will also be reduced. The decrease is generally in the sensible capacity. This is the effect also when the air by-pass or volume control is used.

It is necessary that not only the total capacity but also the sensible and latent cooling requirements both be met. The installation of an excess of coil results in an increase in total capacity, but not a proportional gain in latent heat capacity. On installations controlled from dry-bulb temperature the operating time is shortened because of the added sensible cooling capacity. This results in less moisture pick-up and higher relative humidity than calculated. If an oversize condensing unit is installed, the opposite situation occurs. Generally, this is not a disadvantage except that it results in a load from outside air greater than calculated, as well as in increased power consumption. If oversize equipment is furnished, a balance should be made to assure that the ratio of total to sensible capacity is the same as in the estimated load.

TABLE 2. CAPACITY BALANCES FOR MAXIMUM AND MINIMUM LOAD CONDITIONS

CONDITIONS	CAPACITY IN TONS			RATIO $\frac{\text{TOTAL}}{\text{SENSIBLE}}$
	Total	Sensible	Latent	
Required at peak load conditions	10.90	7.90	3.00	1.38
Required at minimum load conditions	6.62	3.36	3.26	1.98
Peak load equipment balance	10.90	7.90	3.00	1.38
Same equipment balanced at minimum load conditions	9.85	6.58	3.26	1.50
Same equipment balanced at maximum load conditions with 40 per cent by-pass	8.38	5.05	3.33	1.66
Same equipment balanced at minimum load conditions with 38,800 Btu per hour reheat	6.62	3.36	3.26	1.98

Sometimes arbitrary air quantities are specified for ventilation or other reasons independent of the selection of the cooling coil. As shown in Table 1, the coil selection can be altered to take care of various air quantities for the same duty.

Where coil and condensing unit are selected for the peak load condition, and the sensible load partially disappears due to fall of outside temperature or other cause, the condensing unit and coil rebalance. This may result in more sensible and less latent capacity than required at the light load condition, with an increased relative humidity in the conditioned space. Such a condition is shown in Table 2. If approximately 40 per cent of the total air is by-passed, the condition is improved as indicated. The situation may be entirely avoided by using reheat, where it is possible to handle any ratio of sensible and latent loads and maintain the design temperature and humidity[3].

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 by-pass is used so that less air is passed through the coil at light loads. It may be even worse if the control is arranged for decrease of inside temperature with fall of that outside. Freezing can be avoided by making the full load balance a high evaporating temperature and checking the balance at the minimum load.

Care should be exercised in the design of humidity control to minimize the cycling of the refrigerating compressor because of re-evaporation of moisture from the fins. It is sometimes necessary to by-pass air around a coil when the compressor is not operating.

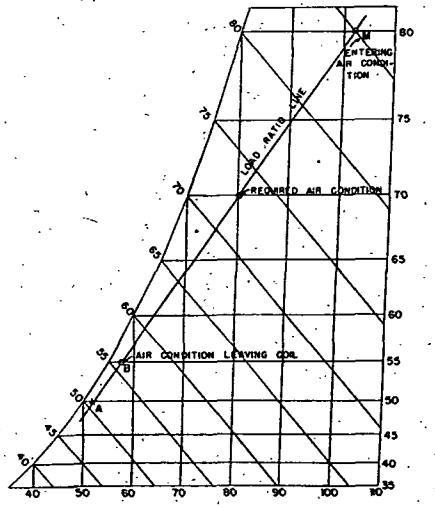


FIG. 14. PSYCHROMETRIC LAYOUT FOR COIL SELECTION

Determining Size of Cooling Coil

The procedure for selecting the proper size of coil for cooling and for determining exit air condition, coil surface temperature, total coil load and refrigerant temperature is illustrated in Example 1.

Example 1. An industrial application requires the cooling of a certain quantity of air from a condition of 102 F dry-bulb and 85 F wet-bulb to a final condition of 80.5 F dry-bulb and 73 F wet-bulb. The air velocity across the coil is to be 400 fpm and coil data are as follows: $h_a = 10.7$ at 400 fpm, $h_r = 325$, external surface area = 15 sq ft per (square foot of face) (row of coil depth), ratio of external surface area to internal surface area = 15.

Solution. (1) Lay out the problem psychrometry as indicated in Fig. 14 and note that the minimum horizontal distance between the load ratio line and the saturation curve is 1.8 F dry-bulb at point A Fig. 14. This means that $t_2 - t_{dp2}$ in Equation 7 must not be less than 1.8. Therefore, Equation 7 should be solved for N to determine the proper number of rows to be used for the coil.

$$\frac{h_a A_1 N}{0.243 G_1} = \log_e \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = \log_e \frac{102 - 80}{1.8} = \log_e 12.22 = 2.5$$

Then substituting values for h_a, A_1, and G_1, N may be found as follows:

$$\frac{10.7 \times 15N}{0.243 \times 1740} = 2.5 \text{ from which, } N = 6.58$$

(2) This establishes the maximum whole number of coil rows that can be used as 6 and it is now possible to determine the actual location of the exit air conditions from Equation 7 by solving for the actual value of $t_2 - t_{dp2}$ for a 6 row coil.

$$\frac{10.7 \times 15 \times 6}{0.243 \times 1740} = \log_e \frac{102 - 80}{t_1 - t_{dp2}} = 2.275$$

This establishes values of 9.78 for $\frac{t_1 - t_{dp1}}{t_2 - t_{dp2}} = R$ and 2.25 for $t_2 - t_{dp2}$.

(3) Next, the exit air condition at 57.3 F dry-bulb and 56 F wet-bulb as shown at B, is found by locating a point on the load ratio line at a horizontal distance of 2.25 dry-bulb degrees from the saturation curve.

(4) The surface temperature may now be found from Equation 7 which may also be written as:

$$t_s = \frac{Bt_2 - t_1}{B - 1}$$

where

$$B = \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$$

$$t_s = \frac{9.78 \times 57.3 - 102}{8.78} = 52.3$$

(5) The total coil load may be calculated from the enthalpy difference across the coil and the air quantity using the weight of dry air instead of the weight of the mixture.

$$q_t = G_a (h_1 - h_2)$$
$$= 1700 (49.24 - 23.77)$$
$$= 43{,}200 \text{ Btu per (hour) (square foot of face area)}$$

where

G_a = Weight of dry air per (hour) (square foot of coil face area).
h_1 = enthalpy of air vapor mixture entering coil, Btu per pound of dry air.
h_2 = enthalpy of air vapor mixture leaving coil, Btu per pound of dry air.

(6) The refrigerant temperature may be found from Equation 9

$$\frac{43{,}200}{15 \times 6 \times \frac{325}{15}} = (t_s - t_r) = 22.1$$

Therefore, $t_r = (52.3 - 22.1) = 30.2$.

Thus a coil 6 rows deep operating at a refrigerant temperature of 30.2 F and a face velocity of 400 fpm is required and it will carry a total load of 43,200 Btu per (hour) (square foot of face area). The air conditions leaving the coil are too low for the conditions of the problem and therefore it is necessary to by-pass air at the entering condition to obtain the desired result of 80.5 F dry-bulb and 73 F wet-bulb.

Although the preceding solution is satisfactory, it may be more desirable in some cases to use a higher refrigerant temperature and employ reheat to obtain the desired load ratio. Such a solution is shown in Fig. 15. In this case the coil load ratio line intersects the saturation curve and therefore a coil of any depth may be selected.

If a coil depth of 6 rows is maintained, the exit air conditions for the coil are indicated at point B Fig. 15 as 72.3 F dry-bulb and 70.8 F wet-bulb and the surface temperature will be:

$$t_s = \frac{9.78 \times 72.3 - 102}{878} = 69.0$$

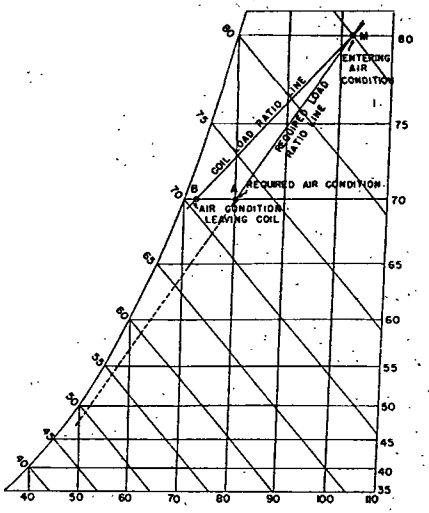


FIG. 15.. PSYCHROMETRIC LAYOUT FOR COIL SELECTION

The coil load will be:

$q_t = 1700 (49.24 - 34.66) = 24{,}800$ Btu per (hour) (square foot of face area) and the refrigerant temperature will be found from Equation 9:

$$\frac{24{,}800}{15 \times 6 \times \frac{325}{15}} = (t_s - t_r) = 12.7$$

Therefore, $t_r = 69.0 - 12.7 = 56.3$.

Thus, for the case where reheat is used, a coil 6 rows deep operating at a refrigerant temperature of 56.3 F is required. The total coil load will be 24,800 Btu per (hour) (square foot of face area) but the actual effective load will be less by the amount of reheat required. Therefore, for a given load, a larger coil and more refrigerating capacity are required when reheat is used.

LETTER SYMBOLS USED IN CHAPTER

η = fin efficiency.
A = external area of coil, square feet per (square foot of coil face area) (row of coil depth).
$B = \frac{t_1 - t_{dp1}}{t_2 - t_{dp2}}$
D = internal diameter of tube, inches.
G = air mass velocity, pounds per (hour) (square foot of coil face area).
G_a = dry air mass velocity, pounds dry air per (hour) (square foot of coil face area).
h_1 = enthalpy of air-vapor mixture entering coil, Btu per pound of dry air.
h_2 = enthalpy of air-vapor mixture leaving coil, Btu per pound of dry air.
h_a = film coefficient of heat transfer between air and external coil surface, Btu per (hour) (square foot external surface) (Fahrenheit degree mean temperature difference between air and coil).

h_r = film coefficient of heat transfer between fluid and internal coil surface, Btu per (hour) (square foot internal surface) (Fahrenheit degree mean temperature between fluid and surface).

k = conductivity of pipe or tube material, Btu (square foot) (hour) (Fahrenheit degree per inch thickness).

L = thickness of tube wall, inches.

MTD = abbreviation—mean temperature difference between fluid in coil and air passing over coil, Fahrenheit degrees.
NOTE: MTD—usually logarithmic mean.

MTD_a = logarithmic mean temperature difference between air and coil surface.

N = number of rows of coil depth.

n = a constant, exponent of G in Equation 8, obtained by plotting, on logarithmic coordinates, G against values of h_a. The value of n is the slope of the line.

q_s = sensible heat transferred Btu per (hour) (square foot of coil face area).

q_t = total heat transferred by coil, Btu per (hour) (square foot of face area).

R = ratio between external and internal surface of tube.

t = average water temperature, Fahrenheit degrees.

t_1 = dry-bulb temperature of air entering coil, Fahrenheit degrees.

t_2 = dry-bulb temperature of air leaving coil, Fahrenheit degrees.

t_a = minimum dry-bulb temperature possible without dehumidification, Fahrenheit degrees.

t_{dp1} = dew-point of air entering coil, Fahrenheit degrees.

t_{dp2} = dew-point of air leaving coil, Fahrenheit degrees.

t_r = average refrigerant temperature, Fahrenheit degrees.

t_s = average temperature of external surface of coil, Fahrenheit degrees.

$\Delta t = t_s - t_r$.

U = over-all coefficient of heat transfer, Btu per (hour) (square foot of external coil surface) (Fahrenheit degrees temperature difference between fluid in coil and air flowing over coil).

V = water velocity, feet per second.

Z = a constant for use in Equation 8 obtained by plotting on logarithmic coordinates G against values of h_a.

NOTE: Numerical subscripts refer to condition entering and leaving respectively.

REFERENCES

1—Rational Development and Rating of Extended Air Cooling Surface, by H. B. Pownall (*Refrigerating Engineering*, October, 1935, p. 211).

2—Performance of Surface-Coil Dehumidifiers for Comfor tAir Conditioning, by G. L. Tuve and L. G. Seigel (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 523).

3—Reheating by Means of Refrigerant Compressor Discharge Gas, by S. F. Nicoll (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 239).

CHAPTER 8

Air Leakage

Causes of Infiltration, Infiltration Due to Wind Pressure, Infiltration Through Walls, Window and Door Leakage, Crack Method, Air Change Method, Infiltration Due to Temperature Difference, Sealing of Vertical Openings

THE air leakage which takes place through various apertures in buildings must be considered in heating and cooling calculations, and properly evaluated. This *infiltration* as it is sometimes designated takes place through cracks around doors and windows, through solid walls and through fireplaces and chimneys. Although the latter sources of leakage may be considerable, they are often neglected on the assumption that dampers would be closed during periods of extreme cold weather or else that the fireplace will be in use at such times and will therefore contribute to the heat supplied and lessen the heating load.

CAUSES OF INFILTRATION

The displacement of heated air in buildings by unheated outside air is due to two causes, namely, (1) the pressure exerted by the wind and (2) the difference in density of outside and inside air because of differences in temperature. The former is generally referred to as *infiltration* and the latter as *stack* or *chimney effect*.

In either case 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) the variations in wind velocity and direction, (3) the exposure of the building with respect to air leakage openings and with respect to adjoining buildings, (4) the variations in outside temperatures which influence the chimney effect, (5) the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors, and (6) the influence of a planned air supply and the related outlet vents. Tight construction is essential for preventing large heat loss due to infiltration.

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 per cent 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 rate at which air is specified to be introduced 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 outside air.

Infiltration Through Walls

Data on infiltration through brick and frame walls are given in Table 1[1]. The brick walls listed in this table are walls which 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 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story 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 per cent or a practically negligible quantity, which indicates the importance of good workmanship in 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

TABLE 1. INFILTRATION THROUGH WALLS[a]

Expressed in cubic feet per square foot per hour

TYPE OF WALL		WIND VELOCITY, MILES PER HOUR					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	2	4	8	12	19	23
	Plastered	0.02	0.04	0.07	0.11	0.16	0.24
13 in. Brick Wall	Plain	1	4	7	12	16	21
	Plastered	0.01	0.01	0.03	0.04	0.07	0.10
Frame Wall, with lath and plaster[b]		0.03	0.07	0.13	0.18	0.23	0.26

[a]The values given in this table are 20 per cent 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]Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster.

shingles is indicated by 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 purely arbitrary air change method, provided the variables entering into the crack method, 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[2]. 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. Not all the window crack in any given room is necessarily used in estimating the infiltration heat

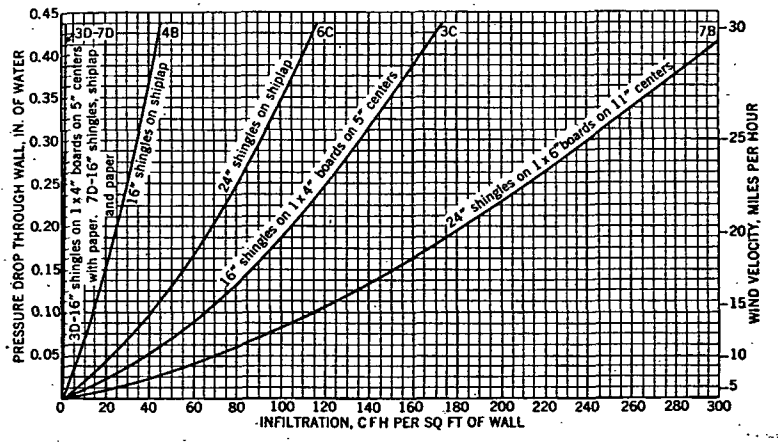


FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

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 14.

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

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, 1/64-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, 1/32-in. crack and 1/32-in. clearance.[e] 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, 1/64-in. cracks[g]	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 per cent 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 1/64-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 per cent 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.

[g] Industrial pivoted window generally used in industrial buildings. Ventilators horizontally 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.

[k] With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-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.

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 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 per cent of the values for industrial pivoted windows as given in Table 2.

When storm sash are applied to well fitted windows, very little reduction in infiltration is secured, but the application of the sash does give an air space which reduces the heat transmission and helps prevent the frosting of the windows[3]. By applying storm sash to poorly fitted windows, a reduction in leakage of 50 per cent may be obtained, the effect so far as air leakage is concerned being roughly equivalent to that obtained by the installation of weatherstrips.

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 which is frequently opened, such as might be found in a store, should have a value applied which is 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 and in the absence of adequate research data the values given in Table 3 represent current engineering practice[4]. These values are based on the average number of persons in a room at a specified time, which may also

TABLE 3. INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS[a]

Expressed in Cubic Feet per Minute per Person Entering Room

APPLICATION	PAIR 36 IN. SWINGING DOORS, SINGLE ENTRANCE[b]	APPLICATION	PAIR 36 IN. SWINGING DOORS, SINGLE ENTRANCE[b]
Bank	7.5	Hospital Room	3.5
Barber Shop	4.5	Lunch Room	5.0
Broker's Office	7.0	Men's Shop	3.5
Candy and Soda	6.0	Office	3.0
Department Store	8.0	Office Building	2.0
Dress Shop	2.5	Public Building	2.5
Drug Store	7.0	Restaurant	2.5
Furrier	2.5	Shoe Store	3.5

[a] For doors located in only one wall or where doors in other walls are of revolving type.

[b] Vestibules with double pair swinging doors, infiltration may be assumed 75 per cent of swinging door values.

Infiltration for 72 in. revolving doors may be assumed 60 per cent of swinging door values.

be the same occupancy assumed for determining the outside ventilation requirements outlined in Chapters 12 and 15.

Air Change Method

The amount of air leakage is sometimes roughly 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. This method may be used to advantage as a check on the calculations made in the more exact manner. On the other hand, where it is not possible to determine or pre-determine with accuracy the width of crack or clearance of windows, or where other sources of air leakage cannot readily be evaluated, as is often the case, the use of the air change method may be justified.

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 for vestibules due to opening and closing of doors 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 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, inside to outside, is a chimney effect, causing air to enter through openings at lower levels and to leave at higher levels[5]. Although it is not appreciable in low buildings, this loss should be considered in tall, single story buildings with openings near the ground level and near the ceiling. Also in tall, multi-story buildings it may be a considerable item unless the sealing between various floors and rooms is quite perfect.

In tall buildings, temperature difference or chimney effect will produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower

TABLE 4. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION[a]

KIND OF ROOM OR BUILDING	NUMBER OF AIR CHANGES TAKING PLACE PER HOUR	KIND OF ROOM OR BUILDING	NUMBER OF AIR CHANGES TAKING PLACE PER HOUR
Rooms, 1 side exposed	1	Rooms with no windows or outside doors	½ to ¾
Rooms, 2 sides exposed	1½	Entrance Halls	2 to 3
Rooms, 3 sides exposed	2	Reception Halls	2
Rooms, 4 sides exposed	2	Bath Rooms	2
		Stores	1 to 3

[a]For rooms with weatherstripped windows or storm sash, use ½ these values, where applicable.

levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors, thereby preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone*[6] is located at mid-height of a building, and that the temperature difference is 70 F, Equations 1 and 2 may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$V_e = \sqrt{V^2 - 1.75\,a} \quad (1)$$

$$V_e = \sqrt{V^2 + 1.75\,b} \quad (2)$$

where

V_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
V = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height, feet.
b = distance if *below* mid-height, feet.

The coefficient 1.75 allows for about one-half the temperature difference head.

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist.

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 per cent to 20 per cent. 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 per cent 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 residences normally supplies the air required for combustion by fuel burning appliances, but in some residences weather-

stripping, sealing and caulking may reduce infiltration to the point that special openings must be provided to supply adequate air to the heating appliances.

REFERENCES

1—A.S.H.V.E. RESEARCH REPORTS No. 786—Infiltration Through Plastered and Unplastered Brick Walls, by F.C. Houghten and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 377). No. 826—Air Infiltration Through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 183). No. 851—Air Infiltration Through Various Types of Wood Frame Construction, by G. L. Larson, D. W. Nelson and C. Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 99).

2—A.S.H.V.E. RESEARCH REPORTS No. 686—Air Leakage, by F. C. Houghten and C. C. Schrader (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 105). No. 704—Air Leakage Around Window Openings, by C. C. Schrader (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 313). No. 803—Air Leakage on Metal Windows in a Modern Office Building, by F. C. Houghten and M. E. O'Connell (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 321). No. 815—Air Leakage Through a Pivoted Metal Window, by F. C. Houghten and M. E. O'Connell (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 519). No. 817—Effect of Frame Calking and Storm Sash on Infiltration Around and Through Windows, by W. M. Richtmann and C. Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1938, p. 547). No. 909—Air Infiltration Through Double-Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 571). The Weathertightness of Rolled Section Steel Windows, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 527). Pressure Differences Across Windows in Relation to Wind Velocity, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 83). Air Infiltration Through Steel Framed Windows, by D. O. Rusk, V. H. Cherry and L. Boelter (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 169).

3—Fuel Saving Resulting from the Use of Storm Windows and Doors, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 87).

4—The Infiltration Problem of Multiple Entrances, by A. M. Simpson and K. B. Atkinson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1936, p. 345). Infiltration Characteristics of Entrance Doors, by A. M. Simpson (*Refrigerating Engineering*, June, 1936).

5—A.S.H.V.E. RESEARCH REPORTS No. 994—Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 387). No. 1069—Heating Requirements of an Office Building as Influenced by the Stack Effect, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 437). Flue Action in High Buildings, by H. L. Alt (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1932, p. 376). Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 377).

6—Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 59).

CHAPTER 9

Natural Ventilation

Wind Forces, Temperature Difference Forces, Heat Removal, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Dairy Barn Ventilation, Garage 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 15 for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 1, Chapter 14, 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 1 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:

$$Q = EAV \qquad (1)$$

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[1].)

The accuracy of the results obtained by the use of Equation 1 depends upon the placing of the openings, as the formula assumes that ventilating

openings have a flow coefficient 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. 1).

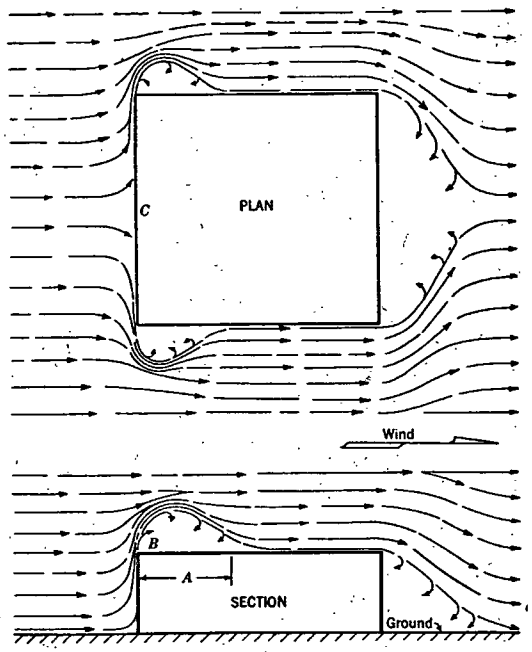


FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (*A*—LENGTH OF SUCTION AREA; *B*—POINT OF MAXIMUM INTENSITY OF SUCTION; *C*—POINT OF MAXIMUM PRESSURE)

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[2]

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 outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\, A \sqrt{h\,(t - t_o)} \qquad (2)$$

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 = *average* temperature of indoor air in height h, Fahrenheit degrees.
t_o = temperature of outdoor air, Fahrenheit degrees.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (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

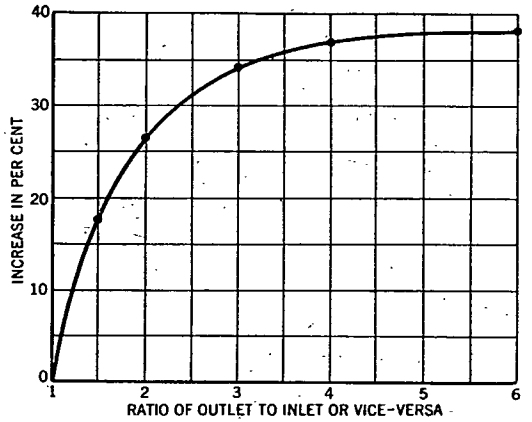


FIG. 2. INCREASE IN FLOW CAUSED BY EXCESS OF ONE OPENING OVER ANOTHER

of air to be passed through the building per minute to maintain this temperature difference can be determined by means of Equation 3.

$$H = 0.0175\, Q\, (t - t_o) \qquad (3)$$

where

H = heat removed, Btu per minute.
Q = air flow, cubic feet per minute.
$t - t_o$ = inside-outside temperature difference, Fahrenheit degrees.

EFFECT OF UNEQUAL OPENINGS

The largest flow per unit area of openings is obtained when inlets and outlets are equal, and the equations given previously 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. 2.

COMBINED FORCES OF WIND AND TEMPERATURE

Equations for determining the air flow due to temperature difference and wind have already been given. 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 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 per cent 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 and the larger 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

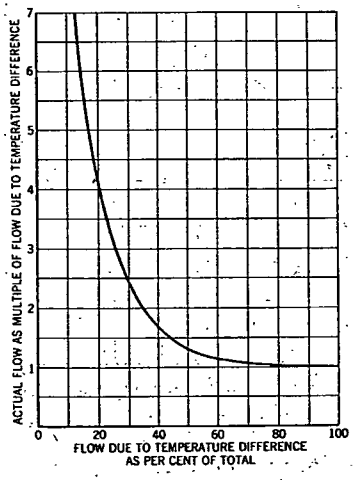


FIG. 3. DETERMINATION OF FLOW CAUSED BY COMBINED FORCES OF WIND AND TEMPERATURE DIFFERENCE

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, as a percentage, 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. 3.

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 pound is used in this shop at the rate of 15 gal per hour (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 $H = \frac{15 \times 7.75 \times 18{,}000}{60} =$ 34,875 Btu per minute.

By Equation 3, the air flow required to remove this heat with an average temperature difference of 10 deg is:

$$Q = \frac{H}{0.0175\ (t - t_o)} = \frac{34{,}875}{0.0175 \times 10} = 199{,}286 \text{ cfm.}$$

This is equal to about 20 air changes per hour. From Equation 2 the inlet (or outlet) opening area should be:

$$A = \frac{Q}{9.4\sqrt{h\ (t - t_o)}} = \frac{199{,}286}{9.4\sqrt{30 \times 10}} = 1224 \text{ sq ft.}$$

The flow per square foot of inlet or outlet would be 199,286 ÷ 1224 = 163 cfm with all windows open.

Solution for Wind Only. With 1,224 sq ft of inlet openings distributed around the sidewalls, there would be about 410 sq ft in each long side and 202 sq ft in each end. The outlet area will be equally distributed on the two sides of the monitor, or 612 sq ft on each side. With the wind perpendicular to the long side, there will be 410 sq ft of opening in its path for inflow and 612 in the lee side of the monitor for outflow with the windward side closed. The air flow, as calculated by Equation 1, will be:

$$Q = 0.60 \times 410 \times 704 = 173{,}200 \text{ cfm.}$$

This gives 17.3 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. 2. This chart shows that when inlets are twice the size of the outlets, in this case 1,224 sq ft in the sidewalls and 612 sq ft in the monitor, the flow will be increased 26.5 per cent 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:

$$612 \times 163 \times 1.265 = 126{,}200 \text{ cfm.}$$

Adding the two computed flows:

Temperature Difference	= 126,200	= 42 per cent.
Wind	= 173,200	= 58 per cent.
Total	299,400	= 100 per cent.

From Fig. 3, it is determined that when the flow, due to temperature difference, is 42 per cent of the total, the actual flow, due to the combined forces, will be about 1.6 times that calculated for temperature difference alone, or 201,920 cfm.

The original flow, due to temperature difference alone, was 199,286 cfm with all openings in use. The effect of the wind is to increase this to 201,920 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.

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 these devices should be considered a part of the ventilation system.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet. These are actuated by the same forces of wind and temperature head, which 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 duct work offers 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 discharged by a roof ventilator depends on wind velocity and temperature difference, and, in general, their 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, or so-called heat valve, 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 RULES

A few of the important requirements 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. Outside air will then be supplied to the zone to be ventilated.

2. Inlet openings should not be obstructed by buildings, trees, sign boards, etc., outside nor by partitions inside.

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 sidewalls 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 that 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 less 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, 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 changes in occupancy are possible, or to provide for extremely hot days. 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.

DAIRY BARN VENTILATION[3]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume and construction permit adequate heating by the stabled animals, the air supply need not be heated. The air should be supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cu ft per hour of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per (hour) (cubic foot of barn space), and 0.197 to 0.305 Btu per (hour) (square foot of barn exposure).

GARAGE VENTILATION

On account of the 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 overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code of Minimum Requirements for Heating and Ventilating Garages, adopted in 1935, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings 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 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out-of-doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage[4].

Research

Research on garage ventilation, undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., in cooperation with the A.S.H.V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory, has resulted in authoritative papers on the subject.

Some of the conclusions from work at the Laboratory are listed in the following statements:

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 is had with complete 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 cu ft per hour, with an average rate of 35 cu ft per hour.

5. An air change of 350,000 cu ft per hour per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

REFERENCES

[1]—Predetermining Airation of Industrial Buildings, by W. C. Randall and E. W. Conover (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 605).

[2]—Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 59).

[3]—Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 181). Cow Barn Ventilation, by Alfred J. Offner (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 149). For additional information on this subject refer to *Technical Bulletin, U. S. Department of Agriculture* (1930), by M. A. R. Kelley. Also see Air Conditioning of Farm Buildings, by F. L. Fairbanks (*Agricultural Engineering*, November, 1937, p. 485).

[4]—Code of Minimum Requirements for Heating and Ventilating Garages (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 30).

Airation Study of Garages by W. C. Randall and L. W. Leonhard (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 233).

A.S.H.V.E. RESEARCH REPORT No. 874—Carbon Monoxide Concentration in Garages, by A. S. Langsdorf and R. R. Tucker (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 511).

A.S.H.V.E. RESEARCH REPORT No. 935—Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 439).

A.S.H.V.E. RESEARCH REPORT No. 934—Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 424).

A.S.H.V.E. RESEARCH REPORT No. 967—Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 395).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 263).

CHAPTER 10

Air Contaminants

Classification of Air Contaminants; Sizes of Airborne Particles; Air Pollution by Smoke, Ash and Cinders; Smoke Abatement; Odor Nuisance; Maximum Allowable Concentrations of Industrial Air Contaminants; Flammable Gases and Vapors; Combustible Dusts; Atmospheric Pollen, Airborne Bacteria

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 substances such as: solid or gaseous diluents for the prevention of explosions; germicidal mists or aerosols for bacteria control; masking substances for odor control; or a substitute for one of the normal gases, as is the case when helium is used to replace nitrogen in atmospheres for compressed air workers or divers.

The control of air quality is one of the functions of complete air conditioning, and some knowledge of the composition, concentration and properties of air contaminants under various circumstances is therefore essential.

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, using distinctions that are necessarily arbitrary in some cases.

CLASSIFICATION OF AIR CONTAMINANTS

Dusts, Fumes, and Smokes are known as solid particulate air contaminants.

Dusts are solid particles projected into the air by natural forces, such as wind, volcanic eruption or earthquake, and by mechanical or man-made processes, such as crushing, grinding, milling, drilling, demolition, shovelling, conveying, screening, bagging and sweeping. Some of these forces *produce* dust from larger masses, while the others simply *disperse* materials that are already in dust or pulverized form. Generally particles are not called *dust* unless they are smaller than about 100 microns in size. 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 of solid materials and may usually be found above molten metals in industrial environments. 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 predominately below the 1 micron size. Fumes permitted to *age* tend to flocculate into clumps or aggregates of much larger size and this tendency may facilitate their removal from air under controlled conditions.

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 characteristically much less than 1 micron in size, often in the range of 0.1 to 0.3 micron.

Mists and Fogs are known as liquid particulate air contaminants.

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. The very small droplets expelled or atomized into the air by sneezing constitute mists containing microorganisms 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 are known as gaseous non-particulate air contaminants.

Vapors are the gaseous phase of substances that are either liquid or solid in their commonly known state, examples being gasoline, kerosene, benzene, carbon tetrachloride, mercury, iodine, camphor. Vapors may be changed to the solid or 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, therefore, 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 themselves 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, the size and specific gravity of the particle, the wind velocity, the orientation of the collecting surface, and the 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.

The great bulk of industrial dust particles are of the order of 1 micron

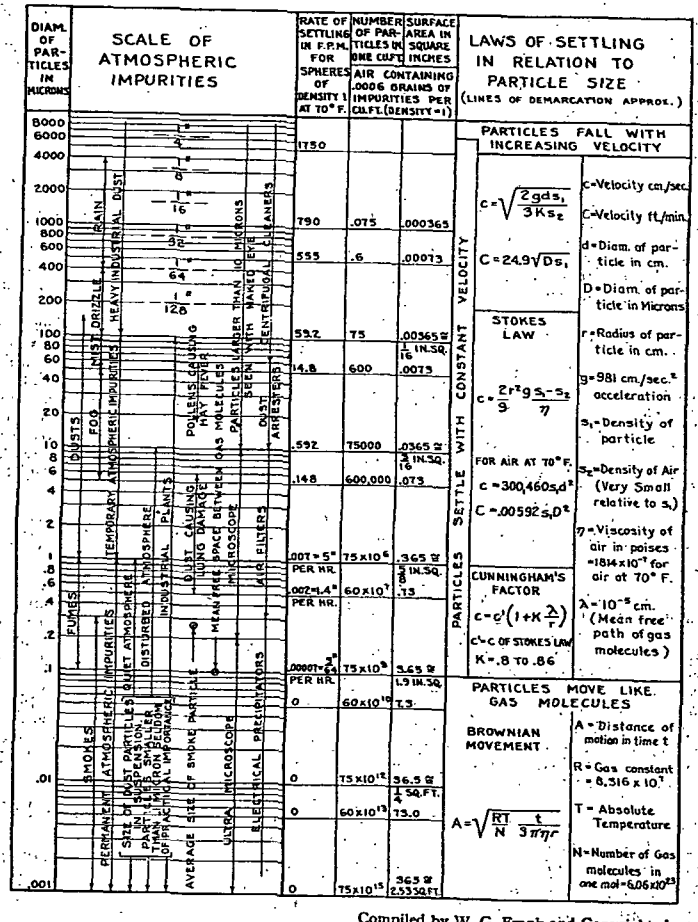


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FIG. 1. SIZES AND CHARACTERISTICS OF AIRBORNE PARTICULATE MATTER

in size. Tremendous numbers are also present in the sub-microscopic range below 0.5 micron, but those below 0.1 micron are not believed at present to be of any 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 their supporting 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 particle to be 0.5 micron, as collected by the Owens jet dust counter and measured under the microscope. The inability of the *light field* microscope to

reveal particles in the 0.1 micron vicinity may account in part for this observed statistical average of 0.5 micron.

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, the intensity and quality of the light, and the nature of the background or the opportunity for contrast. Under ideal conditions a particle of 10-micron size may be recognized, while under less favorable conditions it may be impossible to distinguish a particle smaller than 50 microns. The lower limit of visibility should, therefore, be considered as a physiological range, probably 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 *sub-screen* or microscopic sizes. The 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 with 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.

Mineral particles, such as grains of sand, bits of rock, volcanic ash, or fly-ash, can be transported long distances under unusual circumstances. Thus, the dust storms of 1935 in the Kansas district resulted in vast amounts of fine top soil being thrown high into the air. Solar illumination as far east as Boston was affected noticeably and particles as large as 40 to 50 microns were actually carried half way across the continent before they settled out. In similar manner volcanic ash has been carried even farther. It is not surprising, therefore, that fly-ash from furnace gases, cement dust and the like, can be carried for considerable distances and that, occasionally, the engineer is confronted with the problem of removing such material before the air in question is suitable for use in building ventilation.

AIR POLLUTION BY SMOKE, ASH AND CINDERS

The 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 dusts causing illness or lung disease.

Dust concentrations by weight cannot be converted directly 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 (lightfield 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.

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.

Smoke Abatement

Successful abatement of atmospheric pollution caused by smoke requires the combined efforts of the combustion engineer, industrial executive, public health officer, city planning commission and the community at large. Electrification of industry and railroads, increase in the use of domestic oil and gas furnaces, and segregation of industrial districts is 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. Boilers in these plants are of large size, designed and operated to burn the fuel without wasteful smoke, and equipped in some cases with dust collecting devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of scattered buildings equipped with separate boiler plants.

Time, temperature and turbulence are fundamental requirements for smokeless combustion. Increase of any one of these factors will reduce the quantity of smoke discharged, although excessive turbulence in furnaces may increase the output of ash and cinders. Special care must be taken in hand firing the bituminous coals. (See Chapters 16, 17 and 18 for further discussion on fuel burning technic).

Legislative measures at the present time are largely concerned with reduction of the visible smoke discharged from chimneys of boiler plants. Practically all ordinances limit the number of minutes in any one hour

that smoke of a specified density may be discharged, as measured by comparison with a Ringelmann Chart (Chapter 11, Instruments and Measurements). Ordinances generally do not make specific provision for control of the corrosive and irritant gases, such as sulfur dioxide and trioxide, which are discharged with the gases of combustion. Where high sulfur coals are burned, these sulfur gases present a serious hazard to property, health and vegetation.

In foggy weather the accumulation of these gases in the lower strata of the atmosphere may be such as to cause irritation of the eyes, nose and respiratory passages, and possibly even more dangerous consequences. The Meuse Valley (Belgium) fog disaster has become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulfur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons.

Dusts and cinders in flue gas may be caught by various available devices, such as settling chambers, centrifugal separators, electric precipitators, and gas scrubbers. The difficulty of retaining the dust and cinder particles is principally a function of their size, specific gravity, and resistance to wetting if a washer is used. Descriptions of dust collecting devices are given in Chapter 33.

Absorption of Solar Radiation

The loss of light, particularly the absorption of solar ultraviolet 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 ultraviolet light in the country was 50 per cent greater than in the city. In New York City [7] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

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 the installation of air cleaning devices, the alteration of manufacturing processes, or by legal termination of the offensive operation in residential or commercial districts.

The development and manufacture of more effective and economical air cleaning devices for application to industrial plants will make it possible for many industries to continue operation in their original locations if desired, even though the community standards of air purity become more rigid. To a limited extent such developments already are benefiting the manufacturer, his employees, and his neighbors, when cleaning of the air from ventilated dusty processes makes it possible to return the reclaimed or recovered air to the workroom atmosphere instead of discharging it to the outside with its full load of contamination (Chapter 33).

The 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 much 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, and best of all, substitution of less offensive materials whenever possible [8, 9, 10, 11].

The control of air quality within buildings ventilated for human occupancy is discussed in Chapter 12. 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 have an even greater bearing than indoor contamination on 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 undesirable effects are known to the public and 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 the maximum allowable concentrations for industrial air contaminants as currently accepted in most sections of the country. They apply to *exposures of 8 hours per day*, and refer to the quantities of contaminant permissible in the *workers' breathing zone*. Some of these figures may be altered as the result of continuous research, and some may differ from those in force in a few cities or states. The prudent engineer will design equipment using these values as the *upper* limits of air contamination, and will incorporate a reasonable margin of safety in his estimates of ventilation capacity.

Information on the properties and effects, with respect to health, of specific industrial air contaminants has developed rapidly within the past decade into an extensive literature. Some of the more readily available publications are listed at the end of this chapter.

FLAMMABLE GASES AND VAPORS

Adequate ventilation is a primary requirement for eliminating or 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.

The ability of a flammable liquid to form explosive mixtures is determined largely by its vapor pressure, volatility, or rate of evaporation.

TABLE 3. PHYSIOLOGICAL RESPONSE TO GASES AND VAPORS[a]
Concentrations in Parts of Substance per Million Parts of Air (ppm)

SUBSTANCE	RAPIDLY FATAL	DANGEROUS TO LIFE IN ½ TO 1 HR	MAXIMUM ALLOWABLE CONCENTRATION FOR DAILY EXPOSURES
Acrolein	2,000	100	1
Ammonia	5,000	2,500	100
Amyl acetate	.....	.....	400
Aniline	.....	200	5
Arsine	250	10	1
Benzene (benzol)	20,000	5,000	100[b]
Bromine	500	40	1
Butyl acetate	.....	10,000	400
Carbon dioxide	100,000	50,000	5,000
Carbon disulfide	2,000	1,000	20[b]
Carbon monoxide	4,000	1,000	100[b]
Carbon tetrachloride	50,000	.....	100[c]
Chlorine	1,000	10	1
Dichlorobenzene	.....	.....	75
Dichloroethyl ether	.....	500	15
Ether (diethyl)	.....	35,000	400
Ethyl acetate	40,000	10,000	400
Ethylene dichloride	.....	4,000	100
Formaldehyde	.....	.....	10[b]
Gasoline	.....	.....	1,000
Hydrogen chloride	1,000	.....	10
Hydrogen cyanide	200	100	20
Hydrogen fluoride	.....	50	3
Hydrogen sulfide	600	200	20[b]
Methyl alcohol	.....	.....	200[b]
Methyl bromide	20,000	2,000	50
Methyl chloride	150,000	20,000	100
Methylene chloride	.....	.....	500
Monochlorobenzene	.....	.....	75
Nitrobenzene	.....	.....	5
Nitrogen oxides	300	100	25[b]
Phosgene	50	5	1
Phosphine	1,000	400	1
Styrene	.....	.....	400[b]
Sulfur dioxide	400	150	10
Tetrachloroethane	7,000	.....	10
Tetrachloroethylene	.....	.....	200
Toluene (toluol)	20,000	5,000	200
Trichloroethylene	.....	.....	200
Turpentine	.....	.....	200
Xylene (xylol)	20,000	5,000	200[b]

[a]Adapted from: Manual of Industrial Hygiene, by W. M. Gafafer et al, U. S. Public Health Service (W. B. Saunders Co., 1943); Analytical Chemistry of Industrial Poisons, Hazards and Solvents, by M. B. Jacobs (Interscience Publishers, 1941); Noxious Gases, by Henderson and Haggard (Reinhold Publishing Co., N. Y., 1943); and other authoritative sources.

[b]Adopted by the *American Standards Association* (American Standard Z-37).

[c]50 ppm recommended by the Detroit Bureau of Industrial Hygiene.

TABLE 4. MAXIMUM ALLOWABLE CONCENTRATIONS OF DUSTS, FUMES AND MISTS[b]

SUBSTANCE	MILLIGRAMS PER CUBIC METER, DAILY EXPOSURES[a]
Arsenic; arsenic trioxide	0.15[c]
Cadmium, and compounds	0.1[c]
Chlorodiphenyls	1.0
Chromic acid mist	0.1
Lead; lead carbonate; lead chloride; lead nitrate; lead oxides; lead sulfate	0.15[c]
Manganese, and compounds	6.0[c]
Mercury, and compounds	0.1
Pentachloronaphthalene	0.5
Trichloronaphthalene	5.0
Zinc oxide fume	15.0

[a]1 milligram per cubic meter = 0.44 grain per 1000 cu ft.

[b]Adapted from: Manual of Industrial Hygiene, by W. M. Gafafer, et al, U. S. Public Health Service (W. B. Saunders Company, 1943); and other authoritative sources.

[c]Adopted by the *American Standards Association* (American Standard Z-37).

TABLE 5. MAXIMUM ALLOWABLE CONCENTRATIONS OF DUSTS[a]

SUBSTANCE	MILLION PARTICLES PER CUBIC FOOT OF AIR, DAILY EXPOSURES[b]
Aluminum oxide abrasive	15– 50
Asbestos	5
Carborundum (silicon carbide)	15– 50
Cement (Portland)	50–100
Coal (less than 5 per cent quartz)	50–100
Dusts containing less than 10 per cent *free* silica	15–100
Granite (Barre)	10– 20
Gypsum (hydrated calcium sulfate)	50–100
Limestone (calcium carbonate)	50–100
Marble (calcium carbonate)	50–100
Mica	10– 50
Nuisance dusts (non-toxic, non-silica)	50–100
Quartz (silicon dioxide)	5
Sand (silica, silicon dioxide)	5
Sandstone	5
Silica (*free* or uncombined silicon dioxide)	5
Silicates (combined silicon dioxide)	15–100
Slate	15– 50
Talc	10– 50
Total (maximum concentration for mixed dusts)	50–100

[a]Adapted from: Study of Asbestosis in Asbestos Textile Industry, U. S. Public Health Service Bulletin No. 241, 1938; Industrial Dust, by Drinker and Hatch (McGraw Hill Book Co., 1936); Industrial Code Bulletin No. 35, New York State Department of Labor; recommendations of state and local industrial hygiene agencies compiled by the National Conference of Governmental Industrial Hygienists; and other authoritative sources.

[b]Includes only particles from 1 to 10 microns approximately, as determined by the light field microscope counting technic, using the 10× objective. Dark field counts (and the corresponding allowable concentrations) are anywhere from 2 to 100 times the light field counts for the same sample, according to the proportion of dust smaller than 1 micron (See Industrial Dust, Chapter VII, by Drinker and Hatch, McGraw Hill Book Co.).

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 of flame 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.

The 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 [12, 13].

Design of equipment for the control of combustible anesthetics is outlined in Chapter 13. Construction of equipment for handling air containing flammable substances, or operating in atmospheres so contaminated, is discussed in Chapter 46.

It is customary to report the concentrations of flammable gases or vapors in per cent by volume, or *volume per cent.* Comparison with concentrations on the *part per million* scale used in chemical, medical or industrial hygiene literature is readily made by the conversion: 1 per cent = 10,000 ppm (parts of contaminant per million parts of air, 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 per cent, while the *maximum allowable concentrations* for gases and vapors in Table 3 are below 1000 ppm or 0.1 per cent in most cases. Therefore, control of toxic or injurious vapors in workrooms 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 operations. (See Chapter 46).

The intensity of a dust explosion depends upon: the chemical and thermal properties of the dust; the particle size and shape; the concentration in air; the proportion of inert dust in the air; the moisture content and composition of the air; the size and temperature of the ignition source; and the degree of dispersion of the dust cloud. Investigations on the explosibility of dusts require a determination of the maximum pressure developed during an 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 fact must be weighed when comparing results from several sources [14].

The 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

TABLE 6. APPROXIMATE LIMITS OF FLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a]

GAS OR VAPOR	LOWER LIMIT PER CENT BY VOLUME	UPPER LIMIT PER CENT BY VOLUME	*Closed Cup* FLASH POINT F DEG.[c]
Acetaldehyde	4.0	57	−17
Acetone	2.1	13.0	0
Acetylene[b]	2.5	80	.....
Allyl alcohol	2.4	.....	70
Ammonia	16.0	27.0	.....
Amyl alcohol	1.2	.....	91
Amyl chloride	1.4	.....	.....
Amylene	1.6	.....	.....
Benzene (benzol)	1.4	8.0	12
Benzyl chloride	1.1	.....	140
Butane	1.6	8.5	−76
Butyl acetate	1.7	15.0	72
Butyl alcohol	1.7	.....	84
Butylene	1.7	9.0	.....
Carbon disulfide	1.0	50	−22
Carbon monoxide	12.5	74	.....
Crotonaldehyde	2.1	15.5	55
Cyclohexane	1.3	8.4	1
Cyclopropane	2.4	10.3	.....
Decane	0.67	2.6	115
Dichloroethylene (1, 2)	9.7	12.8	43
Diethyl selenide	2.5	.....	.....
Dioxan	2.0	22.2	65
Ethane	3.1	15.0	.....
Ether (diethyl)	1.7	48.0	−20
Ethyl acetate	2.2	11.5	24
Ethyl alcohol	3.3	19.0	55
Ethyl bromide	6.7	11.3	.....
Ethyl cellosolve	2.6	15.7	104
Ethyl chloride	3.6	14.8	−58
Ethylene	3.0	34.0	.....
Ethylene dichloride	6.2	15.9	56
Ethyl formate	2.7	16.5	−4
Ethyl nitrite	3.0	.....	−31
Ethylene oxide	3.0	80	.....
Furfural (125 C)	2.1	.....	140
Gasoline	1.3	6.5	−50
Heptane	1.0	6.0	25
Hexane	1.2	6.9	−7
Hydrogen cyanide	5.6	40.0	0
Hydrogen	4.1	74	.....
Hydrogen sulfide	4.3	45.5	.....
Illuminating gas	5.3	31.0	.....
Iso-butyl alcohol	1.7	.....	82
Iso-pentane	1.3	.....	.....
Iso-propyl acetate	1.8	7.8	43
Iso-propyl alcohol	2.5	.....	53
Methane[b]	5.0	15.0	.....

[a]Adapted from: Limits of Inflammability of Gases and Vapors, by H. F. Coward and G. W. Jones (U. S. Bureau of Mines, Bulletin No. 279, 1939); 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]Turbulent mixture.

[c]*Closed cup* refers to the equipment used in flash point determinations.

TABLE 6. APPROXIMATE LIMITS OF FLAMMABILITY OF SINGLE GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[a] (Continued)

GAS OR VAPOR	LOWER LIMIT PER CENT BY VOLUME	UPPER LIMIT PER CENT BY VOLUME	*Closed Cup* FLASH POINT F DEG.[c]
Methyl acetate	3.1	15.5	15
Methyl alcohol	6.0	36.5	54
Methyl bromide	13.5	14.5	.....
Methyl butyl ketone	1.2	8.0	.....
Methyl chloride	8.0	19.7	.....
Methyl cyclohexane	1.1	.....	25
Methyl ethyl ether	2.0	10.1	−35
Methyl ethyl ketone	1.8	11.5	30
Methyl formate	5.0	22.7	−2
Methyl propyl ketone	1.5	8.2	.....
Natural gas	4.8	13.5	.....
Naptha (benzine)	1.1	6.0	20–45
Naphthalene	0.9	.....	174
Nonane	0.74	2.9	88
Octane	0.84	3.2	56
Paraldehyde	1.3	.....	.....
Pentane	1.4	8.0	.....
Propane	2.4	9.5	.....
Propyl acetate	1.8	8.0	58
Propyl alcohol	2.5	.....	59
Propylene	2.0	11.1	.....
Propylene dichloride	3.4	14.5	59
Propylene oxide	2.1	21.5	.....
Pyridine (70 C)	1.8	12.4	68
Toluene (toluol)	1.3	7.0	40
Turpentine	0.8	.....	95
Vinyl ether	1.7	27.0	.....
Vinyl chloride	4.0	22.0	.....
Water gas (variable)	6.0	70	.....
Xylene (xylol)	1.0	6.0	63

meter of air. Maximum pressures generated have been reported as high as 500 psi, although they are more likely to be of the order of 50 psi. Investigations on the flammable characteristics of dusts are currently made at 0.1 and 0.5 oz per cubic foot [15-21].

ATMOSPHERIC POLLEN

The properties of pollen grains discharged by weeds, grasses and trees and responsible for hay fever are of special interest to engineers who design equipment for their removal from indoor air (see Allergic Disorders in Chapter 13, and Air Cleaning Devices, Chapter 33). 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 and found near dusty industrial 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. The specific gravity of air-dried pollen is reported to vary from 0.4 to 1.2, and accordingly their weight will either increase or decrease with the addition of moisture[22]. Their shapes and surface designs are marvelously complex, though they tend to be spherical, especially when fully distended with moisture. Illustrations and data on individual pollen grains are available in the botanical literature[22, 23, 24]. The 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 still uncertain 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[29, 30, 31].

AIRBORNE BACTERIA

Study of the occurence and significance of micro-organisms in the atmospheres of the indoor world is currently 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 occupied and unoccupied 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 number of airborne organisms may vary from 1 to 1000 per cubic foot of air, depending on the method of testing[32]. Many are attached to the dust particles which also inhabit 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 13), 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 ultraviolet radiation is commercially feasible for the protection or preservation of pharmaceuticals, cosmetics, and food products.

REFERENCES

1—Atmospheric Pollution of American Cities for the Years 1931 to 1933, J. E. Ives et al (*U. S. Public Health Service Bulletin No.* 224, March, 1936).

2—Micromeritics, The Technology of Fine Particles, by J. M. DallaValle (Pitman Publishing Corporation, 1943).

3—Atmospheric Pollution Due to Smoke, by A. C. Stern (*Heating and Ventilating*, May, 1945).
Atmospheric Pollution Due to Dust and Cinders, by A. C. Stern (*Heating and Ventilating*, July, 1945).

4—Sootfall Studies for New York City, by J. Siegel and B. Feiner (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, p. 495, September, 1945).

5—The Use of Fuel Consumption and Equipment Data in the Abatement of Atmospheric Pollution, by A. C. Stern (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, p. 447-454, August, 1945).

6—Effects of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H. Coblentz and F. A. Korff (*American Journal of Public Health*, Vol. 19, 1929, p. 7).

7—Studies in Illumination—III: A Study of the Loss of Light Due to Smoke on Manhattan Island, by J. E. Ives (*U. S. Public Health Service Bulletin No.* 197, June, 1930).

8—Study and Control of Industrial Atmospheric Pollution Nuisances, by F. M. Stead (*American Journal of Public Health*, Vol. 35, p. 491-498, May, 1945).

9—Evaluation of Odor Nuisance in the Manufacture of Kraft Paper, by J. M. DallaValle and H. C. Dudley (*U. S. Public Health Service* Reprint No. 2022, Public Health Reports, Vol. 54, p. 35-43, January 13, 1939).

10—Disposal of Refinery Wastes, Section II: Waste Gases, Vapors, Sludges and Dusts (*American Petroleum Institute*, New York City, 1938).

11—Offensive Trades, by David Ronald (William Hodge and Co., London).

12—Limits of Inflammability of Gases and Vapors, by H. F. Coward and G. W. Jones (*U. S. Bureau of Mines*, Bulletin No. 279, 1939).

13—Inflammation Limits and Their Practical Application in Hazardous Industrial Operations, by G. W. Jones (*Chemical Reviews*, Vol. 22, February, 1938).

14—Private Communication, by Hylton R. Brown (*Bureau of Mines*, College Park, Maryland).

15—Explosibility of Agricultural and other Dusts as Indicated by Maximum Pressure and Rates of Pressure Rise, by P. W. Edwards and L. R. Leinbach (*U. S. Department of Agriculture Technical Bulletin No.* 490, October, 1935).

16—Dust Explosion Hazards in Plants Producing or Handling Aluminum, Magnesium, or Zinc Powder, by H. R. Brown (*U. S. Bureau of Mines Information Circular No.* 7148, March, 1941).

17—Inflammability and Explosibility of Metal Powders, by I. Hartman, J. Nagy and H. R. Brown (*U. S. Bureau of Mines Report of Investigation No.* 3722, October, 1943).

18—Inflammability and Explosibility of Powders Used in the Plastics Industry, by I. Hartman and J. Nagy (*U. S. Bureau of Mines Report of Investigation No.* 3751, May, 1944).

19—Industrial Dust Explosions, by H. R. Brown (*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*, Boston, May, 1945).

21—National Fire Codes for the Prevention of Dust Explosions (*National Fire Protection Association*, Boston, 1944). Contains codes for aluminum, magnesium, coal, pulverized fuel, flour, spice, starch, sugar, cocoa, sulfur and wood.

22—An introduction to Pollen Analysis, by G. Erdtman (Chronica Botanica Co., Waltham, Mass, 1943).

23—Pollen Grains, by R. P. Wodehouse (McGraw Hill Book Co., New York, 1935).

24—Atmospheric Pollen, by R. P. Wodehouse (Aerobiology, p. 8-31, Publication No. 17, *American Association for the Advancement of Science*, Washington, D.C., 1942).

25—Hayfever Plants, by R. P. Wodehouse (Chronica Botanica Co., Waltham, Mass., 1945).

26—Hay Fever: A Geographical and Botanical Survey, by E. R. Squibb and Sons, New York, 1937.

27—Techniques for Appraising Air-Borne Populations of Microorganisms, Pollen and Insects (*Phytopathology*, Vol. 31, p. 201-225, March, 1941).

28—Apparatus for Determining the Pollen Concentration of the Atmosphere, by B. J. Cody, W. F. Kinney and N. A. Kerstein (Research Department, the Detroit Edison Company, Detroit).

29—The Volumetric Incidence of Atmospheric Allergens, by O. C. Durham (*Journal of Allergy*, Vol. 14, p. 455-461, September, 1943).

30—The Volumetric Incidence of Atmospheric Allergens, II: Simultaneous Measurements by Volumetric and Gravity Slide Methods, by O. C. Durham (*Journal of Allergy*, Vol. 15, p. 226-235, May, 1944).

31—Air-Borne Fungus Spores as Allergens, O. C. Durham (Aerobiology, p. 32-47, Publication No. 17, *American Association for the Advancement of Science*, Washington, D. C., 1942).

32—Sampling Devices, by H. G. DuBuy and A. Hollaender (*American Journal of Medical Science*, Vol. 209, p. 172-177, February, 1945).

BIBLIOGRAPHY

Abstracts and Bulletins (monthly, annual and special) Industrial Hygiene Foundation, Inc., Pittsburgh, Pa.

Aerobiology, Publication No. 17 (*American Association for the Advancement of Science*, Washington, D. C., 1942).

Air Sanitation and Industrial Ventilation, by W. N. Witheridge (Detroit Department of Health, 1945).

Analytical Chemistry of Industrial Poisons, Hazards and Solvents, by M. B. Jacobs (Interscience Publishers, New York, 1941).

Bibliography of Industrial Hygiene, 1900-1943 (*U. S. Public Health Service Bulletin* No. 289, 1945).

Clouds and Smokes, by W. E. Gibbs (P. Blakiston's Son & Co., Philadelphia, Pa., 1924).

Determination and Control of Industrial Dust, by Bloomfield and DallaValle (*U. S. Public Health Service Bulletin* No. 217, 1935).

Dust, by S. C. Blacktin (The Sherwood Press, Cleveland, 1934).

(The) Environment and Its Effect upon Man (Harvard School of Public Health, Boston, 1937).

Health Practices Pamphlets; Industrial Data Sheets (*National Safety Council*, Chicago).

Industrial Dust, by Drinker and Hatch (McGraw Hill Book Co., New York, 1936).

Industrial Hygiene Bulletin (monthly) (New York State Department of Labor).

Industrial Medicine (monthly) (Industrial Medicine Publishing Co., Chicago, Ill.).

Journal of Industrial Hygiene and Toxicology (monthly) (Harvard School of Public Health, Boston, Mass.): See cumulative abstract and subject indexes.

Manual of Industrial Hygiene, by W. M. Gafafer et al (*U. S. Public Health Service*, W. B. Saunders Co., Philadelphia, 1943).

Noxious Gases and the Principles of Respiration Influencing their Action, by Henderson and Haggard (*American Chemical Society Monograph Series No.* 35, Reinhold, New York, 1943).

Occupation and Health, two volumes (*International Labor Office*, Washington, D. C.).

Occupation Hazards and Diagnostic Signs, by Dublin and Vane (*U. S. Dept. of Labor*, Division of Labor Standards Bulletin No. 41, 1942).

Toxicology and Hygiene of Industrial Solvents, by Lehmann and Flury, translated by Eleanor King and H. F. Smyth, Jr. (Williams and Wilkins, Baltimore, 1943).

War Gases, by M. B. Jacobs (Interscience Publishers, New York, 1942).

Yearbooks (*American Public Health Association*).

CHAPTER 11

Instruments and Measurements

Temperature Measurement, Pressure Measurement, Measurement of Air Movement, Air Change Measurements, Measurement of Relative Humidity, Dust Determination, Heat Transfer Through Building Materials, Measurement of Heat Exchange for Comfort Conditions, Combustion Analysis, Smoke Density Measurements

THIS chapter presents a description of many test instruments used for heating, ventilating and air conditioning tests and presents a discussion of their use.

TEMPERATURE MEASUREMENT

Changes in the intensity of heat may be determined by several methods [1] such as measuring the change in volume of a liquid, the change in internal pressure of a confined gas, the current set up between dissimilar metals joined in a circuit, or the change in resistance of an electrical circuit.

Thermometers

The most common method used is the change in volume of a liquid such as mercury or alcohol enclosed in glass. Mercurial thermometers may be used for measuring temperatures from −40 F to approximately 1000 F. The lower limit is set by the freezing point of mercury. Since the boiling point of mercury is only about 675 F, the space above the mercury in thermometers designed for higher temperatures must be filled with an inert gas under pressure. Alcohol thermometers may be used for temperatures from −94 F to +248 F.

The more accurate thermometers have divisions etched on the stem and are preferred for test purposes due to their low heat capacity. During their manufacture, the freezing and boiling points of water are first marked while the thermometer is immersed in a test bath. The required divisions between these points, whether Centigrade or Fahrenheit, are then marked and etched on the stem. In spite of these precautions the probable error in etched stem thermometers is plus or minus one scale division, which makes calibration after manufacture necessary for most test work. Such thermometers are usually calibrated for complete stem immersion.

When incompletely immersed, a stem correction should be made. At ordinary atmospheric temperatures the correction is negligibly small, but it usually is important when measuring high temperatures such as those of steam and flue gas. The emergent stem correction may be calculated by the equation:

$$K = 0.00009\, D\, (t_1 - t_2) \qquad (1)$$

where

K = correction to be added, Fahrenheit degrees.
D = number of degrees on the thermometer scale which are not immersed.
t_1 = temperature indicated on the thermometer, Fahrenheit degrees.
t_2 = temperature of the non-immersed mercury column, Fahrenheit degrees.
0.00009 = difference in the coefficient of expansion of the mercury and glass.

Since the bulb has considerable area, radiant energy may affect temperature readings [2]. In measuring room temperatures, care must be taken to locate thermometers away from hot surfaces, such as radiators, or cold surfaces, such as walls or windows. Where this is impracticable, bright, polished shields should be used to screen the bulb from the radiant energy.

Errors may also be avoided: (*a*) by allowing the thermometer time to reach equilibrium, (*b*) by providing sufficient circulation to give a true average temperature of the medium observed, and (*c*) by reading with the eye at the same level as the top of the liquid, *i.e.*, avoiding parallax.

Two types of industrial thermometers for permanent installation in ducts or pipes are shown in Fig. 1, and consist of a glass thermometer protected by a metal case. The better grades have metal scales made specially for each instrument. These are ruled in the same manner as

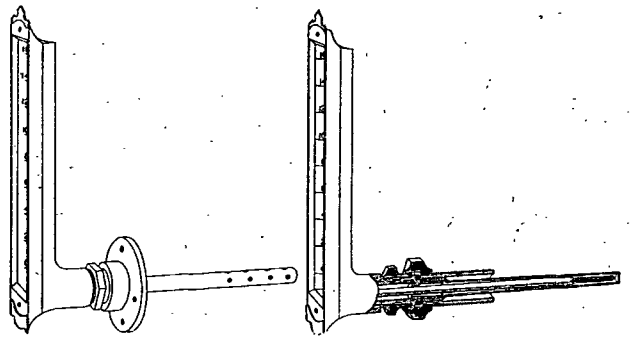


FIG. 1. INDUSTRIAL THERMOMETERS FOR PERMANENT INSTALLATION

described for etched stem thermometers. Due to the heat capacity and heat conductance of the jacket, it is more difficult to obtain the true temperature at a point with these than with the exposed etched stem type. The latter is usually preferred for test purposes.

Thermometer Wells

Where temperatures of fluids or gases in vessels or conduits are to be measured, it is often necessary to resort to thermometer wells. These are especially designed to contain thermometers and thermocouples. Since they separate the temperature sensitive element from the medium to be observed, large errors may result from their improper use [3]. These errors may be due both to poor heat transfer from the wall of the well to the sensitive element, and also to the tendency of heat to travel along the length of the well itself. To improve heat transfer in the case of thermometers, the void should be filled with a liquid of minimum practical viscosity. At the well mouth, the stem should be packed to check evaporation and heat transfer to the atmosphere. To reduce the temperature gradient over the length of the well, the wall of the conduit should be carefully insulated in the area surrounding the observation point.

Thermocouples

When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force is developed. By maintaining the cold junction at a constant temperature (or providing equivalent electrical compensation), the magnitude of the electromotive force developed is a direct measure of the temperature of the warm junction. By proper selection of metals, any temperature up to 2900 F may be measured. Readings are obtained by means of a potentiometer or sensitive galvanometer which may be calibrated in degrees. If a potentiometer is used, there is no current flowing through the thermocouple wires at the time of measurement and therefore the resistance of the thermocouple circuit does not affect the accuracy of the reading. If a galvanometer is used instead of a potentiometer, the instrument is usually built with a high internal resistance to minimize the effect of the resistance of the thermocouple circuit and also, special

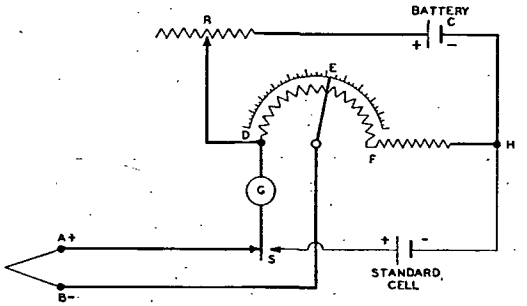


FIG. 2. BASIC CIRCUIT AND CONNECTIONS FOR THERMOCOUPLE AND POTENTIOMETER

lead wires of known resistance are used to connect the galvanometer to the thermocouples. The basic potentiometer circuits are shown in Fig. 2. 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 correct for variations in voltage from battery C, switch S is momentarily thrown over to the standard cell circuit and rheostat R is then adjusted so that the galvanometer shows zero current. Battery C then exerts the known voltage of the standard cell at DH.

Calibration of thermocouples for high temperatures may be made against known melting points of metals. Radiation effects may be minimized by using *the smallest size of wires consistent with mechanical strength*. The use of small wires also makes the thermocouple sensitive to minute fluctuations in temperature. Wires of No. 33 B & S gage are small enough to be sensitive and to minimize radiation effects while still rugged enough for many purposes. By the use of thermocouples, temperatures at remote points may be indicated or recorded on conveniently located instruments; average temperature may be readily obtained by connecting several couples in parallel or in series; and tem-

peratures 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 give an emf which, divided by the number of couples to give the average emf [4] per couple, may be used to find the average 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 emf only if the electrical resistances of the parallel junctions are the same [4, 5].

The temperature of the surface is at best difficult to obtain accurately [6]. The thermocouple is most readily adaptable for this purpose. In a metal surface, a common method is to peen the couple into a small drilled hole, bearing in mind that the temperature indicated is that existing at the last point of junction in the couple. Other methods involve

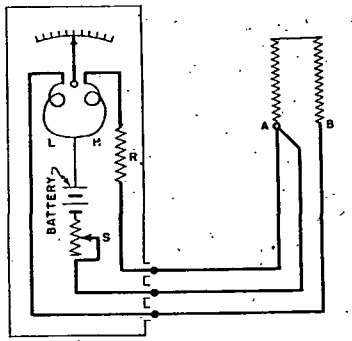


FIG. 3. TYPICAL RESISTANCE THERMOMETER CIRCUIT AND CONNECTIONS

fastening the couple to the surface with adhesive cellophane, or cementing the couple with litharge in a surface scratch, and grinding it flush with the surface. Wires may also be fastened to the surface by brazing, care being taken to add as little extra metal as possible. This is a useful method for obtaining the temperature of cast-iron heating surfaces.

In any of these methods the leads should be of as fine wire as practicable, since conductance along the leads to the couple may be a source of considerable error.

Resistance thermometers depend for their operation upon the change of resistance of metal with change in 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. 3. In this design a differential galvanometer is used, in 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.

Instruments of this type are frequently connected through a selector switch to a number of thermal resistor elements and used to indicate temperatures in remote locations in large buildings. The direct reading feature is advantageous in this case.

Pyrometers

For measuring high temperatures, such as in furnaces, *pyrometers* are often used. Radiation pyrometers concentrate the radiant energy on a thermopile, and the reading is obtained on a galvanometer or potentiometer. Optical pyrometers require visual matching of a narrow spectral band, usually red, emitted by the object with that from a standard electric lamp.

PRESSURE MEASUREMENT

Barometer

The most accurate barometer for determining the atmospheric pressure is the mercurial type, consisting of a tube over 30 in. long closed at the top and standing in a mercury well. The barometric pressure is expressed as the height of the mercury column above the level of the mercury in the well. Such barometers are equipped with an adjustment to compensate for change in level of mercury in the well. The reading should be taken at the top of the meniscus and is obtained on a Vernier scale.

Correction for variation of the density of the mercury column and for expansion of the brass scale, which are usually calibrated for 32 F mercury and 62 F scale temperature, should be made by subtracting from the observed height in inches the value of C determined by Equation 2.

$$C = \frac{h\,(t - 28.630)}{(1.1123\,t - 10978)} \qquad (2)$$

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. 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 as observed or corrected to sea level.

Atmospheric pressure may also be measured by an aneroid barometer which is easily portable. In this type, variations in atmospheric pressure bend the thin surface of a sealed box or tube. The aneroid type is not as accurate as the mercurial and needs frequent calibration. Most of the pressure gages used in engineering work indicate 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.

Pressure Gages

The Bourdon type gage is a widely used device for measuring pressures. The Bourdon tube is elliptical in cross-section and circular in form, and is connected by suitable linkage to a pointer which moves over a dial. An increase in pressure tends to straighten the tube and a decrease has the opposite effect. When used with high temperature steam, the tube must be protected by a water seal. When used with ammonia it must be made of steel or other material not attacked by this substance. When used for sub-atmospheric pressure, the gage is known as a vacuum gage, and is usually graduated in inches of mercury. For pressures above atmospheric, it is termed a pressure gage and is graduated in pounds per square inch. Some are made to read in both directions and are termed compound gages. Calibration is usually made by means of a dead weight tester, consisting of a platform and weights resting on a piston floating on oil. The pressure at all points in the fluid is determined from the area of the piston and the total weight resting on the oil. Adjustments

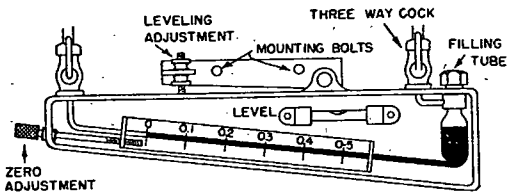


FIG. 4. INCLINED DRAFT GAGE

are provided in the gage linkage to make necessary corrections. A correction chart may be used for accurate work.

Manometers

For comparatively low gage pressures or differences in pressure between two points in a duct system, the vertical *U* tube is a simple and accurate gage and is often used for test work with various fluids such as mercury, water, kerosene, or alcohol. Readings may be in inches of any of these fluids.

For measuring pressure differences of a few inches of water, or less, *U* gages are often set at an angle for scale amplification. In commercial gages of this type, commonly termed *draft gages*, only one tube of small bore is used and the other leg is replaced by a reservoir. Although the scale is calibrated to read in inches of water, a fluid having the density and characteristics of kerosene is often used. It is necessary, in such a gage, 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 to detect errors in gravity of fluid. For more accurate calibration the gage may be checked against a micromanometer or a calibrating device known as a hook gage [7]. The accuracy of a draft gage is dependent on the slope of the tubes 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.

The better grades of inclined draft gages are equipped as shown in Fig. 4. This includes a built-in level and leveling adjustment, a means of adjusting the scale to zero, and three-way vent cock connectors for checking purposes.

For measuring low pressure differences to within 0.001 in. of water very sensitive *micromanometers* are available, such as the Illinois or Wahlen, the Askania, and the Emswiler [8, 9]. Calibration of these is impossible, and readings are converted to pressure units by fundamental calculations involving the specific gravity of the fluids used and the design principles involved.

Static Holes and Tubes

In the case of low pressure air flow, the type of pressure opening used and its location are quite as important as the accuracy of the gage to

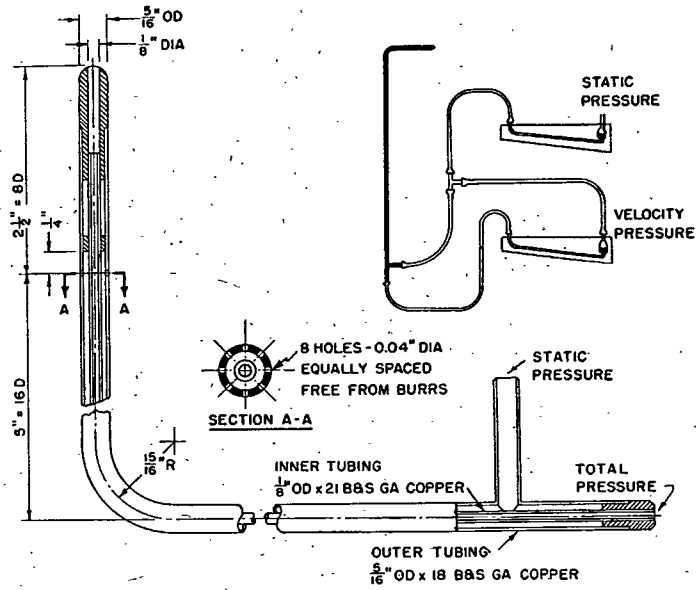


FIG. 5. STANDARD PITOT TUBE

which it is connected. Where velocities are low, as in certain plenum chambers, or where flow is free of large eddies and parallel to the walls of the conduit, a carefully drilled opening cleared of burrs and at right angles to the stream will give accurate readings [10]. The drilled depth of this hole through the wall should be at least two diameters. A second method less likely to involve error is the use of the static pressure element of a Pitot tube shown in Fig. 5. A static tube of the same general design as the Pitot may also be used, omitting the center tube. In using this instrument it should be pointed upstream, avoiding impact or eddies.

MEASUREMENT OF AIR MOVEMENT

The problem of measuring air movement may be divided into three main parts: air confined in ducts, air circulating in free spaces, and air entering or leaving such space through openings such as grilles. Other

gases may be measured by the same methods, but emphasis here is on air measurements [11].

For determining the velocity, and therefore the volume of air flowing in a duct, the Standard Pitot Tube [7] shown in Fig. 5 is probably most often used. When connected as illustrated, this instrument will give readings of both static and velocity pressure directly. From the latter the air velocity may be found from tables, or calculated from the following relation:

$$V = 1096.5 \sqrt{\frac{h_v}{d}} \qquad (3)$$

where

V = velocity, feet per minute.
h_v = velocity pressure, inches of water.
d = density of air, pounds per cubic foot.

Air flow in a duct is seldom uniform. In general the velocity is lowest

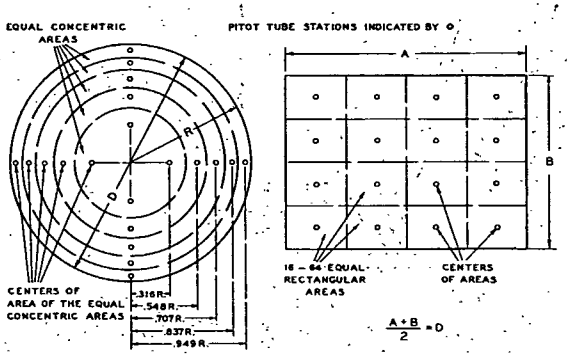


FIG. 6. PITOT TUBE TRAVERSE FOR ROUND AND RECTANGULAR DUCTS

near the edges or corners, and greatest at or near the center. For this reason a large number of readings should be taken in the manner shown in Fig. 6. In the case of round ducts not less than 20 should be taken along two diameters at centers of equal annular areas. 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.

For small pipes it is sometimes necessary to construct a Pitot tube smaller than the standard size. Such a small Pitot tube should be geometrically similar to the standard tube. Pulsating or disturbed flow will give erroneous results and every effort should be made to remove disturbances in the Pitot tube section.

The velocities used in many ducts are too low for measurement by means of pressure gages and it therefore becomes necessary to resort to other types of instruments.

Many forms of Pitot tubes other than the one described have been used and calibrated [12]. A double-ended tube [13], one end pointing down-stream, and one up-stream, 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 on exhaust inlets, such as on hoods placed around grinding wheels.

The rounded approach orifice or nozzle of the general type described in the A.S.H.V.E. Unit Heater [14] and Unit Ventilator [15] Codes is an accurate air measuring device. When it is well made, the coefficient closely approaches unity. The discharge from such a nozzle is uniform [16] and provides a good location for calibration of air velocity instruments [17].

The Venturi meter is like the nozzle except for the addition of a down-stream transition section that reduces the pressure drop through the measuring apparatus.

The thin-plate square-edged orifice has a coefficient of approximately 0.60. The exact value depends on the location of the connections, the pressure drop, the diameter ratio of orifice to pipe, and the sharpness of the edge [18].

Another method of air measurement uses the thermal electric principle where, by means of a measured amount of current, heat is put into the air stream. The weight of air flowing is calculated from the heat equivalent of the electrical input and the temperature rise of the air. Heat should be applied uniformly to the mass of air passing, and the small temperature rise must be determined accurately.

In certain applications the air velocity through a duct, heater coil, or heating unit may be most conveniently obtained 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 plane of temperature measurement.

Air Currents in Free Spaces

One of the instruments useful in determining the velocity of air currents in free spaces is the Kata-thermometer. It is essentially an alcohol thermometer with a large bulb. The stem has two marks, one corresponding to 95 F, and the other 100 F. The instrument is heated above 100 F and then the time in seconds required for it to cool from 100 to 95 when located in the air current gives a measure of the non-directional velocity. It is important to have the Kata-thermometer dry before taking the reading. Each Kata has its own factor etched on the stem, and this factor must be used with its cooling formula or chart for obtaining the velocity. The Kata-thermometer is useful in exploring ventilated spaces to determine whether the proper air movement and distribution are being maintained. It 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 [19].

Another instrument for measuring low velocity air currents is the

heated thermometer anemometer [20]. This consists of a mercurial glass thermometer with a resistance winding on the bulb. Current is supplied from an external source in a measured amount. The difference between the temperature of this heated thermometer and that of an ordinary thermometer at the same location, together with the current supplied, makes it possible to calculate the non-directional velocity of the air stream.

The heated thermocouple anemometer employs a thermocouple instead of a thermometer [21].

Another instrument is the hot wire anemometer which has been made in several patterns. In general, a measured current is supplied to raise the temperature of a fine bare wire above the temperature of the surrounding air. With the use of a very fine wire, minute fluctuations in velocity may be measured, and the area exposed to radiant exchange with heated or cooled surfaces is at a minimum. This instrument is easily adapted to remote reading or recording. A group of them may be connected together to give the average velocity in a space, or the velocity at individual points within a test space, by suitable switching arrangements [22, 23].

Deflecting Vane Anemometer

The deflecting vane anemometer consists of a pivoted vane enclosed in a case, against which air exerts a pressure as it passes through the instrument from an up-stream to a down-stream 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. When used in fluctuating velocities, it is necessary to average visually the swings of the needle to obtain average velocities. This instrument is very useful for studying mixing of air in a room [24] and in locating and measuring peak velocities that may be objectionable. Various attachments are available, such as the double tube arrangement for obtaining velocities in ducts, and a device for measuring static pressures. Each instrument and the attachments for it must receive individual calibration.

Propeller or Revolving Vane Anemometer

The propeller or revolving vane anemometer consists of a light revolving 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.

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. It is rare to have access to the interior of duct sections where the flow is sufficiently uniform for measurement. 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 [25, 26]. 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 get 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 [27].

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 [28].

Any of the other anemometers described can be 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 [29]. 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 [30].

While hardly a quantitative instrument, smoke is very useful in studying air streams and currents. The application of a more accurate instrument is often made more exact by a preliminary exploration with smoke. A mixture of potassium chlorate and powdered sugar in equal portions gives a very satisfactory non-irritating smoke. It is fired by a match, and since considerable heat is evolved, it should be placed in a pan away from inflammable objects.

AIR CHANGE MEASUREMENTS

Atmospheric air contains a certain amount of carbon dioxide. Its concentration is increased within enclosures by the carbon dioxide given off by occupants. The total air change through open windows, infiltration, and mechanical ventilation, may be measured by the carbon dioxide concentration but absorption of CO_2 by walls and other materials may introduce errors [31] and consequently some investigators prefer hydrogen to CO_2 for such tests. Since occupants also give off moisture, the increase in humidity may also be used as an index of ventilation within a space. Usually more direct methods of measuring air supply and air distribution are in favor.

MEASUREMENT OF RELATIVE HUMIDITY

Wet- and dry-bulb mercurial thermometers are usually used to determine relative humidity. The sling psychrometer is a common mounting of the thermometers to permit swinging. The wet-bulb wick and water

for wetting it must be clean, and the temperature of the water should preferably be slightly above the wet-bulb temperature. An air stream velocity of 900 fpm is recommended, although higher velocities will result in greater accuracy [32]. This velocity is obtained with the sling psychrometer by whirling it rapidly, followed by reading the wet-bulb thermometer quickly before its indication changes. Owing to the human element involved in this method, more accurate results are obtained with types of psychrometers in which air is drawn over the thermometers by a fan, or by a bulb operated aspirator. In ducts, the air flow itself gives the proper evaporating conditions. Several observations should be made until the minimum temperature is reached. Relative humidity may be obtained from tables or psychrometric charts [33]. Although it is common practice to use the charts which are based on a barometric pressure of 29.92 in. Hg, a correction for barometric pressure is necessary for extreme accuracy. This correction is made by multiplying the relative humidity as determined from the chart by the ratio of the observed barometric pressure and the standard barometric pressure.

For temperatures below 32 F, the water on the wick is allowed to freeze, during which time the temperature will drop below the true wet-bulb. A thin film of ice is more desirable than a thick one, and it is satisfactory to remove the wick and freeze a thin film directly on the bulb. Care must be taken to read the temperatures accurately due to the slight wet-bulb depressions. Tables for ice conditions must be used [34].

Dew-point apparatus for humidity measurements consists of a polished plated container cooled by the evaporation of a volatile liquid within. The temperature at which the first slight water vapor forms on the polished surface is the dew-point. If the temperature is below 32 F, the deposit will appear as frost. Another method of determining humidity is by chemical means in which the water vapor is removed by a drying agent and weighed on a chemical balance. A thermal conductivity method is available for temperatures above 212 F or for extremely low humidities [35].

DUST DETERMINATION

The measurement of dust is complicated by the many kinds involved. Some of the collecting methods are impingement on viscous surfaces, impingement at high velocity under water, collection on porous crucibles or filter paper through which air passes, and electric precipitation. Determination may be by direct weighing of samples or by microscopic counting. A commonly used method employs the Smith-Greenburg impinger which collects samples in water or alcohol and in which particles are counted under a microscope in various cells such as the Hatch or Dunn cells. Another method employs the Lewis [36] sampling tube with the analytical determination of the increase in weight of a porous crucible. All reports should state the method of sampling and counting. The A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work specifies the porous crucible method [37].

HEAT TRANSFER THROUGH BUILDING MATERIALS

The A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls [38] describes the construction and use of the guarded hot box for determining over-all heat transmission coefficients of built-up sections including surface resistances. This apparatus consists of inner and outer insulated boxes, each having one open side which is sealed tightly against

the wall specimen to be tested. The outer box serves as a guard and a zero temperature difference is maintained between the inner box and the air space separating it from the outer box. Electric heaters are provided to maintain any desired temperature within the boxes while at the same time the outer surface of the wall specimen is subjected to a low temperature. A common method of applying this low temperature is to fit the specimen into an opening in the wall of a refrigerated room. The air temperature difference between the two sides of the specimen, and the rate of heat input to the guarded hot box are measured, from which data an over-all heat flow coefficient can be calculated.

The Nicholls heat meter is very useful for determining the heat flow through walls of buildings [39]. This apparatus consists of a guarded plate on both sides of which are mounted a series of thermocouples. The plate is calibrated so that the rate of heat flow through it can be determined by observing the temperature difference between the two surfaces. In use, the meter is clamped tightly against the surface of the wall in question, and readings of heat flow of accuracy sufficient for many test purposes may be obtained.

In June 1942, the A.S.H.V.E. adopted a standard test procedure for determining the conductivity of materials by use of the guarded hot plate [40]. This method is adapted to homogeneous materials, and conductivities obtained do not include surface coefficients. The method involved is one in which two identical samples are arranged one on each side of a heated plate having a guard and a central test section. The resulting sandwich is placed between cold plates. In operation the same temperature is maintained in the guard section and test section of the heated plate and a lower temperature is maintained in each of the cold plates from a common source of cold water. Thermal conductivity is determined by measuring the amount of heat supplied to the center or guarded portion of the hot plate, the temperatures at the hot and cold surfaces and the thickness of the specimen. The apparatus requires careful, precise construction and its operation requires care and skill. The actual design of basic types of hot plates is described in an *A.S.T.M.* publication [41].

MEASUREMENT OF HEAT EXCHANGE FOR COMFORT CONDITIONS

Several instruments have been devised to measure the effect of various factors as they relate to the comfort of the body [42]. The principal ones are the Kata-thermometer, Dufton's eupatheoscope, Vernon's globe thermometer, Winslow and Greenburg's thermo-integrator, and Yaglou's heated globe [42, 44]. These instruments were designed to obtain a quantitative measurement of the thermal exchange between the human body and its environment.

COMBUSTION ANALYSIS

The analysis of flue gases to determine completeness and efficiency of combustion is usually made chemically with an Orsat apparatus. This consists of a measuring burette, a leveling bottle, and three pipettes. Carbon dioxide is absorbed in the first pipette by potassium hydroxide, oxygen in the second by potassium pyrogallate, and carbon monoxide in the third by cuprous chloride. A known volume of gas is drawn in, and

after each of the three absorptions the reduced volume is again measured in the burette. Pressure and temperature of the gas sample are kept constant while measuring. Several passes are made through each pipette which contains tubes or glass beads to increase the wetted surface. It is essential that each reaction be completed before the next reaction is started. Since the life of the reagents is limited, it is well to keep a record of the number of samples tested. Care is needed in operation to prevent the pulling of reagents out of the pipettes into the capillary tubing and burette. Many types of recording gas analyzers are available and are usually found in the larger boiler plants.

Carbon Monoxide Measurement

A method of analyzing for low carbon monoxide concentrations completes the oxidation of the carbon monoxide in a known volume of sample, in the presence of a catalyst. The heat resulting is measured by a thermocouple calibrated in parts per 10,000 of carbon monoxide [45].

SMOKE DENSITY MEASUREMENTS

Smoke density may be judged by assigning to it the number of the Ringelmann Smoke Chart which appears to have the same color when observed at a distance of 50 ft. The charts are numbered 1 to 4 and are made of black lines cross-ruled on white as given in Table 1.

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

Apparatus using the photo-electric cell has been devised for recording smoke densities in large plants.

REFERENCES

1—For a comprehensive treatment of temperature measurement the reader is referred to Temperature, Its Measurement and Control in Science and Industry, a symposium sponsored by the *American Institute of Physics* and published by Reinhold Publishing Corp.

2—Errors in the Measurement of the Temperature of Flue Gases, by P. Nicholls and W. E. Rice (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 473).

3—Temperature Measurement (*A.S.M.E.* Power Test Code, Part 3).

4—Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 94).

5—Parallel-Connected Thermocouples for the Testing of Gas Appliances, by Walter B. Kerk and George J. Pacanovsky (*Gas*, September, 1939, p. 51).

6—Measurement of Surface Temperatures, by F. C. Houghten and H. T. Olson, Temperature, Its Measurement and Control in Science and Industry (Reinhold Publishing Corp.).

7—Standard Test Code for Centrifugal and Axial Fans, Edition of 1938. See also Standard Code for the Testing of Centrifugal and Disc Fans (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407; Vol. 37, 1931, p. 363).

8—Illinois Micromanometer (*University of Illinois, Engineering Experiment Station Bulletin* No. 120, p. 91).

9—The Weathertightness of Rolled Steel Windows, by J. E. Emswiler and W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 527).

10—Pressure Measurement (*A.S.M.E.* Power Test Code 1936, Part 2, Chapter 2).

11—For technical data refer to Fluid Meter Reports, Parts 1—1937, 2—1931, and 3—1933 (*American Society of Mechanical Engineers*).

12—Technical Notes No. 546 (*National Advisory Committee for Aeronautics*, November, 1935).

13—The Characteristics of Double Pitot Tubes, by F. R. Ingram, E. Diez-Canseco and L. Silverman (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1942, p. 708).

14—Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165), adapted January, 1930 by A.S.H.V.E.

15—A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25), adopted June, 1932.

16—Discharge Coefficients of Square Edged Orifices for Measuring the Flow of Air, by H. S. Bean, E. Buckingham and P. S. Murphy (*Bureau of Standards Journal of Research*, Vol. 2, 1929, p. 561).

17—A.S.H.V.E. RESEARCH REPORT No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 645).

18—Flow Measurement by Nozzles and Orifice Plates (*A.S.M.E.* Power Test Codes, Chapter 4 of Part 5, 1940).

19—Temperature, Humidity and Air Motion Effects in Ventilation, by O. W. Armspach and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 103).

20—The Heated Thermometer Anemometer, by C. P. Yaglou (*Journal Industrial Hygiene and Toxicology*, Vol. 20, October, 1938, No. 8).

21—A.S.H.V.E. RESEARCH REPORT No. 1165—Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 351).

22—Development of Testing Apparatus for Thermostats, by D. D. Wile (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 349).

23—Linear Hot Wire Anemometer, Its Application to Technical Physics, by L. V. King (*Journal Franklin Institute*, 1916).

24—A.S.H.V.E. RESEARCH REPORT No. 1204—Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B. Priester and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 241).

25—A.S.H.V.E. RESEARCH REPORTS Nos. 857, 911 and 966—Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 201, Vol. 37, 1931, p. 619, and Vol. 39, 1933, p. 373).

26—A.S.H.V.E. RESEARCH REPORT No. 1162—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 313).

27—A.S.H.V.E. RESEARCH REPORT No. 1092—The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr. and M. H. Dean (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 387).

28—A.S.H.V.E. RESEARCH REPORT No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 403).

29—A.S.H.V.E. RESEARCH REPORT No. 1076—Air Distribution From Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 77).

30—Measuring Air Flow, by G. L. Tuve (*Heating, Piping and Air Conditioning*, December, 1941).

31—A.S.H.V.E. RESEARCH REPORT No. 959—Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

32—The Temperature of Evaporation, by Willis H. Carrier (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 25).

33—Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew-Point (*U. S. Department of Agriculture*, Weather Bureau, Washington, D. C.).

34—A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems, by W. H. Carrier and C. O. Mackey (*A.S.M.E. Transactions*, January, 1937, p. 33; Discussion *A.S.M.E. Transactions*, August, 1937, p. 528).

35—Gas Analysis by Measurement of Thermal Conductivity, by H. A. Daynes (*Cambridge Press*, 1933).

[36]—Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by S. R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

[37]—A.S.H.V.E. Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, adopted January, 1934 (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

[38]—A.S.H.V.E. Standard Test Code for Heat Transmission Through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253), adopted 1928.

[39]—A.S.H.V.E. RESEARCH REPORT No. 685—Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 65).

[40]—Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate (Tentative), adopted July, 1942, by A.S.H.V.E., (*A.S.T.M. Designation* C177-42T).

[41]—Guarded Hot Plate Apparatus Complying With the Requirements of Section 4 of *A.S.T.M.* Tentative Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate, *A.S.T.M.*

[42]—Measurement of the Physical Properties of the Thermal Environment, by D. W. Nelson, F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, A. P. Gagge, John A. Goff, A. E. Hershey, F. C. McIntosh, F. W. Reichelderfer, G. L. Tuve and C. P. Yaglou (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1942, p. 382).

[43]—Instruments and Methods for Recording Thermal Factors Affecting Human Comfort, by C. P. Yaglou, A. P. Kratz and C.-E. A. Winslow (Year Book, *American Journal Public Health*, 36-37).

[44]—The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 149).

[45]—A Carbon Monoxide Recorder, by S. H. Katz, D. A. Reynolds, H. W. Frevert and J. J. Bloomfield (*U. S. Bureau of Mines, Technical Paper* No. 355, 1926).

CHAPTER 12

Physiological Principles

Chemical Vitiation of Air, Physical Impurities in Air, Thermal Interchanges Between the Body and Its Environment, Adaptation to Environmental Conditions, High Temperature Hazards, Acclimatization, Effective Temperature Index, Relation of Air Conditioning Needs to Metabolism

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 not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying only a given quantity.

The term *air conditioning* in its broadest sense implies control of any or all of the physical or chemical qualities of the air. When applied to comfort air conditioning, however, the A.S.H.V.E. 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."

CHEMICAL VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain physical and chemical changes which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs.

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are too small to produce appreciable effects even under the worst conditions of normal human occupancy. Only in such unusually air-tight enclosures as submarines and some air raid shelters need the increase in carbon dioxide and the reduction in oxygen be considered. The amount of carbon dioxide in air is often used as an index of odors of human origin, but the information it affords rarely justifies the labor involved in making the observation[2, 3]. Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a substance has not been demonstrated. The only certain fact is that expired air may be odorous, offensive, and capable of producing loss of appetite and a disinclination for physical activity. 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 really 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 10.) 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 33 on Air Cleaning Devices.

When the only source of contamination is the human occupant, 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. Perception of odor has been found to vary as the logarithmic function of the odor intensity, 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* [3] and the A.S.H.V.E. Research Laboratory [2]. The findings from the Harvard study are given in Table 1. Outdoor air requirements 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.

The total quantity of outside air to be circulated through an enclosure is governed by both chemical and physical considerations. The physical requirements for controlling temperature, air distribution and air velocity usually predominate. 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 rather than by outside air.

It will be noted that, with adequate air space, the rate of air change indicated in Table 1 is from 10 to 30 cfm per person. In rooms occupied by only a few persons such a rate of air change will be automatically attained in cold weather by normal leakage around doors and windows, while it 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 an air change 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

TABLE 1. MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS [4]

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
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	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

such conditions, the control of air temperature is the major factor to be considered.

In more crowded rooms (large offices, large workrooms, auditoriums), the whole picture changes. Cubic space per person is less and it is usually impossible to admit untempered outside air without creating drafts. Here, mechanical ventilation is essential, but as will be noted in a later paragraph, it is even more essential for thermal than for chemical reasons. It is control of the thermal properties of the air in order to effect the removal of the heat produced by human bodies, rather than control of chemical properties, which must govern practice.

The present data regarding the effect of cubic space on ventilation requirements are not universally accepted. It is anticipated that studies now in progress under the auspices of the armed forces may yield additional information on the subject.

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 provisions of the code aim to provide *minimum*, rather than *adequate*, requirements.

Notwithstanding the rapid advance in the field of air conditioning during the past few years, there still remain those who believe in a superior, stimulating quality of outdoor air (particularly country, mountain and seashore air) under ideal weather conditions, as compared with

properly conditioned air. While this point of view is usually held by persons not intimately acquainted with the complicated factors involved, they nevertheless carry some weight. It is apparent, however, to anyone acquainted with the factors involved and the conditions effecting comfort, that in modern air conditioning, as in most other branches of engineering, modern science makes it possible to control the phenomena of nature for the service and comfort of man beyond any possibilities found in nature itself. When the requirements for optimum comfort as determined by the atmospheric environment are known (and the comprehensive studies to date indicate that they are known at least to a high degree), the air conditioning engineer can supply these requirements indoors to the same perfection as may accidentally be found at times outdoors, and keep them under control. The freedom of movement, action and thought, together with the variability of stimulae experienced by persons under ideal conditions in the country, mountains or seashore, and the psychological effect of these *wide open spaces* undoubtedly have some stimulating effect, which when compared with the monotony of confinement indoors, even in the most favorable atmospheric environment, account for the contrast. Various experimenters have attempted to duplicate the invigorating qualities of outdoor air by the use of ozone, ionization, or ultra-violet light, but results to date have been inconclusive or negative [5].

Ozone has been used with success for the destruction of micro-organisms (molds) in meat packing establishments and the like; and where considerable amounts of organic effluvia are present it may be useful as a deodorant. For ordinary ventilation practice, however, neither of these purposes can be attained, since the concentration of ozone necessary for effectiveness would be likely to transcend the limit of comfort in ordinary occupied rooms. While ozone has been used in the treatment of certain diseases, there is no evidence that it has a tendency to increase comfort or to benefit health under conditions of normal human occupancy. The allowable concentrations in the breathing zone are very small, between 0.01 to 0.05 ppm parts of air. These are much too small to influence bacteria. Higher concentrations are associated with a pungent unpleasant odor and considerable discomfort to the occupants. One part per million causes respiratory discomfort, headaches, depression, and a lowering of the metabolic rate, and may even lead to coma [6].

PHYSICAL IMPURITIES IN AIR

Dust particles of various types, when present in considerable concentrations, produce an irritating effect upon the mucous membranes of nose and throat and may be associated with high prevalence of acute respiratory diseases such as bronchitis and pneumonia. Dust which contains free silica has special harmful effects, causing a primary disease of the lungs (silicosis) and predisposing the victim in a high degree to tuberculosis. These, however, are special problems of industrial hygiene which will not be discussed in detail in this chapter. (See Chapter 10.)

A certain part of the dissemination of disease in confined spaces is caused by the emission of pathogenic organisms from infected persons. Droplets sprayed into the air in talking, coughing, sneezing, etc., do not all fall immediately to the ground within a few feet from the source, as was formerly believed. The large droplets fall, but minute droplets less than 0.1 mm in diameter evaporate to dryness before they fall the height of a man. Nuclear residues from such sources, which may contain infective organisms drift long distances with the air currents and the

virus may remain alive long enough to be transmitted to other persons in the same room or building. Droplet nuclei have been recovered from cultures of resistant micro-organisms a week after introduction into a tight chamber of 3000 cu ft capacity, although the majority of disease germs died out within a few hours [7]. Practical epidemiological evidence indicates that the danger of such atmospheric transmission is slight with the bacterial diseases but may be appreciable with the diseases caused by the much smaller viruses. Avoidance of overcrowding is a major factor in avoiding such dangers. The microbic concentration in the atmosphere may be reduced by air change, but since the rate of contamination may be great at local points over short periods of time the hazardous concentration may not be eliminated quickly enough and may even be spread over larger areas by local drafts. The possibility of sterilizing the air supply at the source, or destroying the micro-organisms at their point of admission to the air by the use of ultra-violet light [8] or glycol vapors [9] is being studied and offers considerable promise.

While in some instances it may be possible to reduce the physical impurities of the air by dilution from a non-contaminated source, such non-contaminated sources are rarely available. Frequently the outside air contains a higher concentration of physical impurities than that within an enclosure. Therefore, it is usually desirable to reduce the concentration of physical impurities by air cleaning methods (see Chapter 33).

THERMAL INTERCHANGES BETWEEN THE BODY AND ITS ENVIRONMENT

The importance of the thermal factors arises from the profound influence which they exert upon body temperature, comfort and health. Body temperature depends upon the balance between heat production and heat loss. The heat resulting from the oxidation which occurs within 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 level of about 98.6 F, the heat production must be balanced by the heat loss.

In conditioning air for comfort and health it is necessary to know the rate of sensible and latent heat liberation from the human body, which in conjunction with other heat loads (see Chapters 6, 14 and 15) determines the capacity required for proper conditioning. The data in common use are those of the A.S.H.V.E. Research Laboratory [10].

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.
S = rate of storage.
E = rate of evaporative heat loss.
R = rate of radiative heat loss or gain.
C = rate of convective heat loss or gain.

Factor M, the rate of metabolism, is always positive. The storage, S, may be either positive or negative, depending upon whether heat is being

stored or given off, accompanied by 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 the surface temperature of the body is above that of the walls and air, and negative when it is cooler.

The human body possesses remarkable powers of adaptation to a narrow range of atmospheric conditions around an ideal optimum where storage is zero, and metabolism and skin and tissue temperature are at optimum values. Under these conditions, the body experiences a sensation of comfort. As skin temperature and body-tissue temperature rise or fall above or below an optimum, complex adaptive mechanisms come into play, chiefly associated with redistribution of blood supply between the skin and deeper tissues (in a cold environment) and with sweat secretion (in a hot environment). These reactions are governed by nervous or chemical stimuli from both skin and internal tissues. Nerves from the skin, for example, carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The reactions involved in cold and in hot environments are on the whole radically different in nature. The mechanisms of adjustment involved are extremely complex and, while they are receiving considerable study, a complete understanding of their operation is still lacking.

Some of the phenomena of body temperature control are shown graphically in Fig. 1. The dotted curves, from a study at the *John B. Pierce Laboratory of Hygiene* [11], 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 per cent relative humidity) and the metabolic rate, the rate of heat dissipation by radiation and convection combined, and the latent heat loss due to evaporation of perspiration and moisture from the respiratory tract. The smooth line curves from the work of the A.S.H.V.E. Research Laboratory [10] give the same relationships for healthy, male subjects (18 to 24 years of age), seated at rest and normally clothed for winter-heated and air conditioned occupancy. The 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, storage is given by the difference between the metabolism and total heat loss.

The small difference between the metabolic or heat production rates for the two types of subjects may be accounted for by the 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 considerable heat loss, and for very warm conditions where there is sensible transfer from the atmosphere to the body. The two curves for evaporative loss serve to show how physiological control uses evaporation of perspiration to maintain equilibrium, particularly at high temperatures. Below about 75 F for the normally clothed subject,

and below about 85 F for the lightly clothed subject, evaporation loss is minimal and probably due to uncontrolled evaporation from the relatively dry skin and from the respiratory tract. Above these temperatures control is obtained by the availability of perspiration for evaporation. The difference in the curves above 75 F is probably largely determined by the difference in clothing and activity. Above temperatures from 95 to 100 F (the probable average outside surface temperature of the clothed body) the combined effect of radiation and convection causes a change from positive to negative. Slightly above this temperature even the greatly increased latent heat loss ceases to take care of the rate of heat production plus the negative radiation and convection loss, and heat storage occurs with a consequent rise in body temperature. Above this range, even

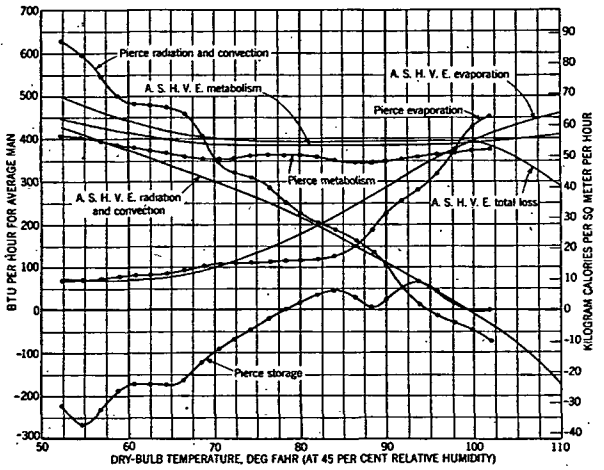


FIG. 1. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND OPERATIVE TEMPERATURE FOR THE CLOTHED SUBJECT

though there is inability to dissipate heat rapidly enough, metabolism actually increases. This may be accounted for by the predominance of the purely chemical laws of increased chemical reaction with rise in temperature, over physiological control, and indicates the point where a breakdown in thermal equilibrium begins. For higher temperatures life can only survive to the point where these accelerated processes will result in a rise in body temperature to the limiting level of from 106 to 108 F.

Air movement is an important factor in increasing heat loss by either convection or evaporation. The result is accomplished through removal of hot humid air from near the body surface and replacing it with cooler and relatively drier air. This is an important factor in maintaining thermal equilibrium either for persons at rest or at work in hot, humid conditions. For conditions in the comfort zone and below, excessive velocities (particularly localized drafts) should be avoided since differential cooling of one area of the body may produce surprisingly unpleasant reactions in quite different parts of the body. In one experiment [12] it was shown that the application of an ice pack to an area of 60 sq cm on the back of the neck for 15 min caused a drop of 17 F in the skin temperature

of the fingers and that this low temperature of the fingers persisted for one hour after the ice pack was removed.

ADAPTATION TO ENVIRONMENTAL CONDITIONS

As stated previously, the optimum environmental conditions for comfort are those under which the body can maintain complete thermal equilibrium with only minor adjustments in blood distribution. As conditions depart from the optimum, more drastic adjustments are made by the body in its attempt to maintain equilibrium, and the sensation of comfort gives way to one of increasing discomfort. If conditions are too severe, adequate adjustments may be impossible, and prolonged exposure may result in serious injury or death.

In a cold environment the blood vessels leading to the surface are constricted, thus decreasing the flow of blood and heat to the skin, and reducing the radiation and convection losses from it. If this adjustment is insufficient, an involuntary muscular activity (shivering) takes place which increases the metabolism. The range over which these physiological controls are effective is rather narrow, and man's existence in cold climates is dependent upon his ability to provide himself with heated living quarters and warm clothes.

During the last few years extensive research has been carried on by and for the armed services of the United Nations to determine the proper types and amounts of clothing for a wide range of conditions. The results of these studies are now being released and should add greatly to our knowledge on the subject.

As man's environment changes from cool or comfortable to warm, the body first attempts to adjust to the new conditions by a redistribution of the blood circulation. Blood capillaries near the surface become dilated, allowing more blood to flow to the skin, thus raising the skin temperature and increasing the radiation and convection losses from the body. If this correction proves inadequate, the stimulus is extended to the sweat glands which cause progressively greater portions of the body area to become moist or wet as conditions become more severe. The evaporation of this moisture from the body surface provides the body with its most effective method of cooling, but even this fails when the dew-point temperature of the air becomes so high that rapid evaporation is no longer possible. Although the body is able, by evaporative cooling, to maintain complete thermal balance under conditions of moderate excess, it does so only at the cost of considerable discomfort.

Studies at the *John B. Pierce Laboratory of Hygiene* [13] have indicated the relation between discomfort and the degree of wetting of the body surface by perspiration for lightly clothed subjects in a semi-reclining position, and the investigators there have designated this as the *zone of evaporative regulation*. The relation between sensible perspiration and the atmospheric environment for normal persons at rest and at work has been determined at the A.S.H.V.E. Research Laboratory [10].

HIGH TEMPERATURE HAZARDS

Studies at the A.S.H.V.E. Research Laboratory [14] and elsewhere [15] during the past two decades have made available a mass of information dealing with the physiological effects of hot atmospheres on workers and means of alleviating the distress and hazards associated therewith. This

interest has been termed *air conditioning in industry*, or *the effects of hot atmospheres in industrial hygiene*, and is a growing factor in air conditioning applications. Table 2 gives some of the physiological responses of men at rest and at work, to hot environments. Physiological studies [16] indicate that 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.

In hot environments the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is suggested that the feeling of lassitude and discomfort experienced is due in part to the anaemic condition of the brain. The stomach loses some of its power

TABLE 2. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp	Actual Creek 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 A.S.H.V.E. Research Laboratory.
[b]Computed value from exposures lasting less than one hour.

to act upon the food, owing to a diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract [17]. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may cause excessive fatigue, or may lead to attacks of cramps, unless the salts are replaced. In order to relieve both cramps and fatigue, it is recommended that salt tablets be taken, or that sufficient salt be added to the drinking water to provide a $^1/_{10}$ of 1 per cent saline solution [18].

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. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

Diseases due to heat are now classified as heat exhaustion, heat cramps,

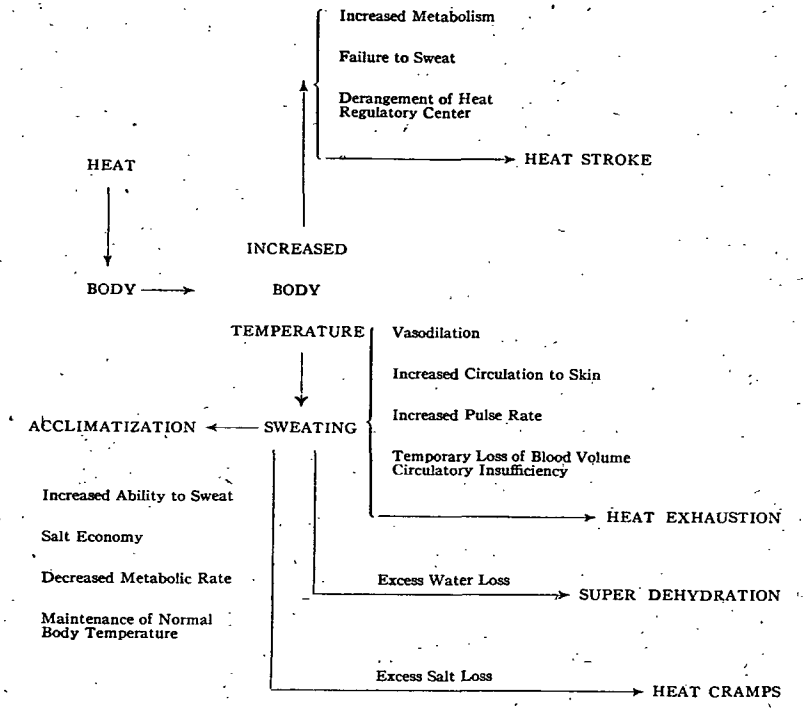


FIG. 2. PHYSICAL EFFECTS OF OVERHEATING AND ACCLIMATIZATION

and heat strokes [19]. Heat exhaustion is due to circulatory failure; heat cramps to excessive loss of body chlorides and heat stroke to an inadequacy of the heat dissipating mechanism which results in hyperthermia (high fever) (See Fig. 2). If the hyperthermia becomes excessive, the liver and the central nervous system may be seriously injured and this damage may prove fatal. The hazards of high temperature are not understood. It is difficult to say whether short exposures at high temperatures are more harmful than longer exposures at lower temperatures. A new concept is evident from the observation of an increase in leucocytes (white cells) of the blood in workers subjected to high temperatures. These leucocytes are defensive factors which are increased when infection invades the body. A rise in temperature and leucocyte count indicates a mobilization of the body defenses. Since both temperature and cell count are increased [20], the question arises as to whether long exposures to very high temperatures might not cause exhaustion of these defense mechanisms.

A study was made recently at the Armored Medical Research Laboratory [21] to determine the upper limits of environmental conditions under which a man can perform certain work. Thirteen enlisted men, who were first thoroughly acclimatized to the hot conditions, served as

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

subjects. During each test, the subjects were required to march for 4 hr at the rate of 3 mph, carrying 20 lb packs. Tests were made under a wide range of environmental conditions, and these environments were rated in 3 zones 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. Fig. 3 shows the limits of the three environmental zones.

Recognition of the need of air conditioning for workers in hot industries is growing rapidly and this should become an important field for the air conditioning engineer. The hot conditions may be remedied by any of the recognized comfort cooling applications. The choice of the type of system and cycle to be used in a given instance must be determined by the air conditioning engineer after a study of surrounding conditions.

In some hot industries where a small number of workers are engaged in spaces of large volumetric capacity the worker himself, rather than the

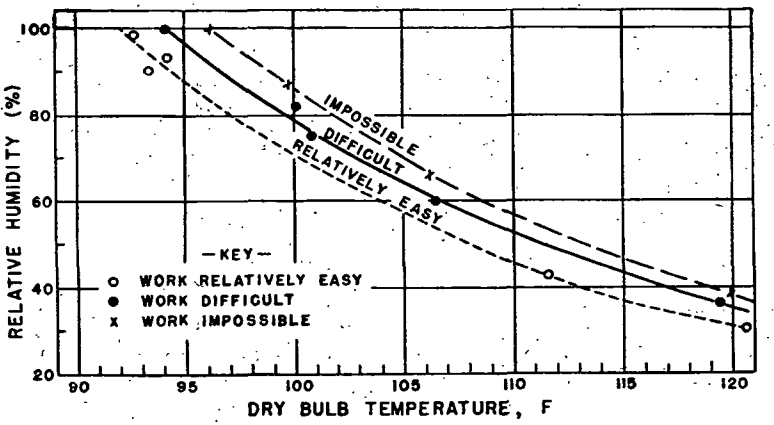


FIG. 3. THE ENDURANCE OF ENVIRONMENTAL CONDITIONS BY ACCLIMATIZED SUBJECTS WORKING AT A SPECIFIC RATE

From The Upper Limits of Environmental Heat and Humidity Tolerated by Acclimatized Men Working in Hot Environments (*Journal of Industrial Hygiene and Toxicology*, March, 1945, p. 70). Used by permission.

entire environment, can be cooled by placing him in a small cooled and ventilated booth, by blowing cooled air over him, or by circulating cooled air through a loose-fitting suit [22].

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue. They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature exceeding 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. An environment averaging 64 F for the 24-hour period has been indicated as associated with minimal mortality [23].

Within limits, however, there does occur a definite adaptation to external temperature level. People and animals raised under conditions of tropical moist heat stand chilling poorly as they are unable quickly to increase internal combustion to keep up the body temperature. For this reason they have trouble standing the cold, stormy weather of the temperate zones, and when exposed to it are very susceptible to respiratory infections. Likewise, people living in cool climates suffer greatly in the moist heat of the tropics until their adaptive mechanism has been adjusted. Within a few years, however, they find themselves reacting as natives to the new environment.

The adaptive level changes somewhat with the season [24]. 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 and skin temperature levels are higher for women. The thickness and insulating value of the clothing worn is also an important factor in the determination of the comfort level.

In recent tests made at the A.S.H.V.E. Research Laboratory [25], subjects were required to perform light work under very hot conditions for a four hour 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 two weeks. However, after acclimatization was completed, a recess of several days had no effect on the endurance of the subject.

EFFECTIVE TEMPERATURE INDEX

The primary purpose of comfort air conditioning is to provide an environment in which the body can easily maintain its thermal equilibrium, and thereby experience a feeling of comfort. It is important, therefore, that the designing engineer know what conditions are necessary for comfort.

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 considerably 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.

Combinations of temperature, humidity, and air movement which induce the same feeling of warmth are called thermo-equivalent conditions. A series of studies [26] at the A.S.H.V.E. Research Laboratory established the equivalent conditions met with in general air conditioning work. 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 deg saturated with moisture. The *effective temperature index* cannot be measured directly but is determined from dry- and wet-bulb temperature and air motion observations by reference to an Effective Temperature Chart (see Figs. 4, 5 and 6) or tables.

Fig. 4 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. 4 for air velocities of 300 and 500 fpm have been presented in some of the earlier editions of the GUIDE. Fig. 5 is another form of effective 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.

Fig. 6, commonly referred to as the Comfort Chart [27], is an effective temperature chart on which the summer and winter *comfort zones* have been indicated. These zones indicate the various combinations of conditions under which 50 per cent or more of the people are comfortable. Curves showing the percentage of subjects comfortable at each effective

temperature in summer and winter have been added to the chart. The summer comfort zone is indicated as extending from 66 ET to 75 ET with a maximum of 98 per cent comfortable at 71 ET. The winter comfort zone extends from 63 ET to 71 ET with a maximum of 97 per cent comfortable at 66 ET. The 71 ET and 66 ET lines are referred to, respectively, as the summer and winter comfort lines.

The comfort zones and lines as shown in Fig. 6 are based on research prior to 1932. Later studies [28] by the A.S.H.V.E. Research Laboratory

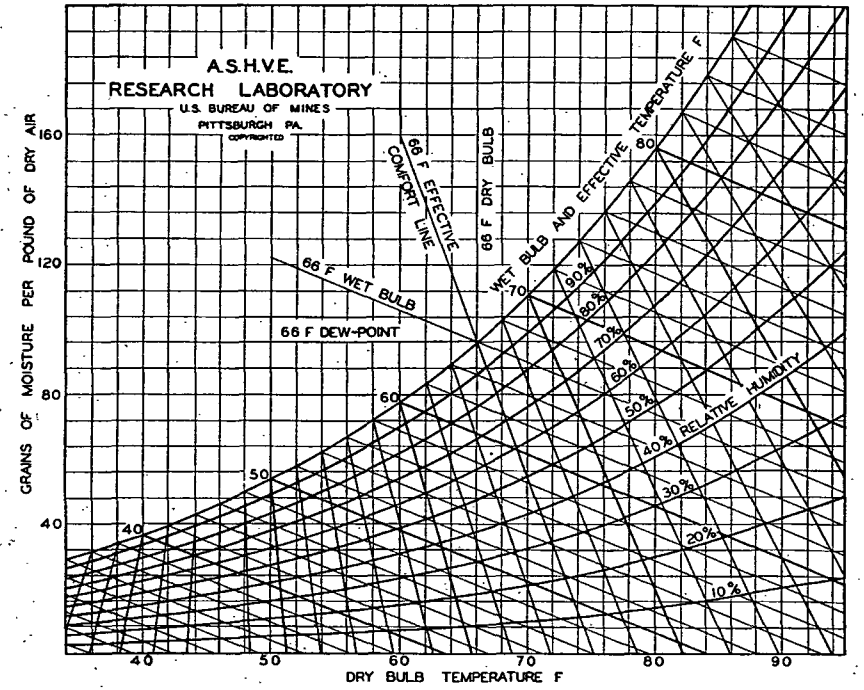


FIG. 4. PSYCHROMETRIC CHART, PERSONS AT REST, NORMALLY CLOTHED, IN STILL AIR

indicate a desirable winter effective temperature of 67 deg, and this finding is confirmed by current practice. The shape of the curve showing the per cent of subjects comfortable in winter also justifies this conclusion, since a drop of only one degree from the 66 ET line seriously reduced the percentage of subjects comfortable.

The comfort zones in Fig. 6 are located between the 30 per cent and 70 per cent relative humidity lines. There is some evidence that the zones could be extended somewhat beyond these limitations.

It should be emphasized that a satisfactory system will not necessarily result by designing for just any combination of conditions within the boundaries of the comfort zone. As pointed out, the comfort zone covers all conditions under which 50 per cent or more of the subjects were comfortable. An air conditioning system which leaves 50 per cent of the

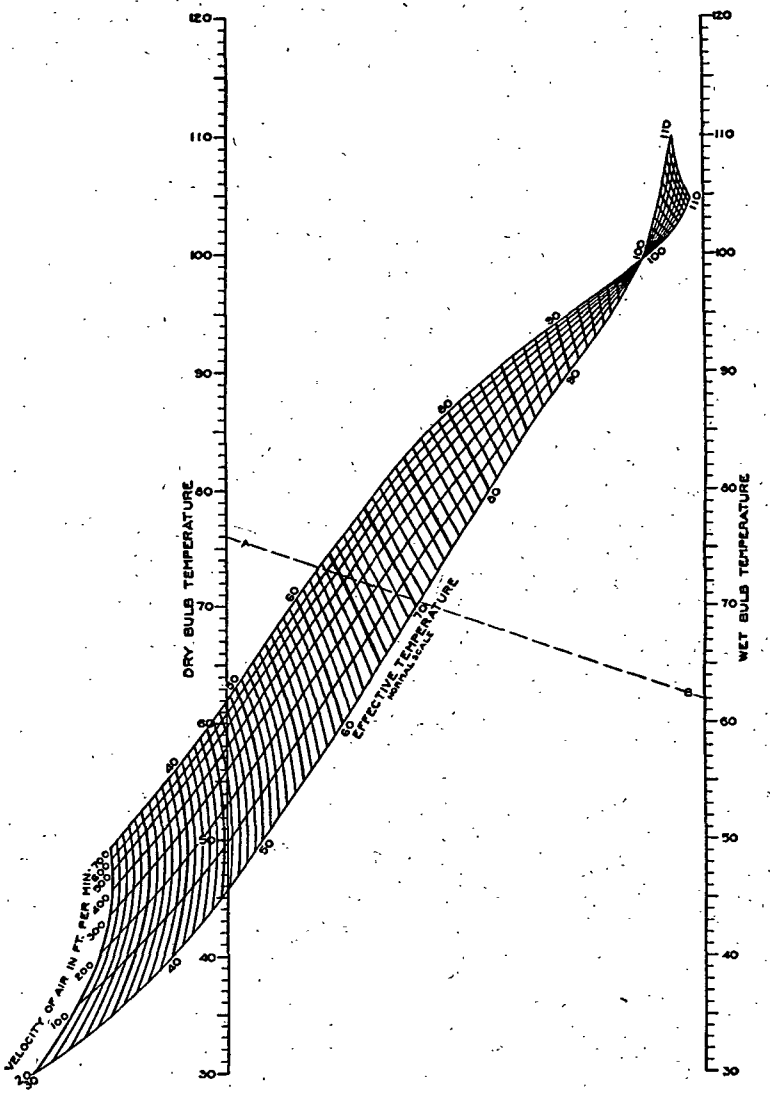


FIG. 5. EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

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.

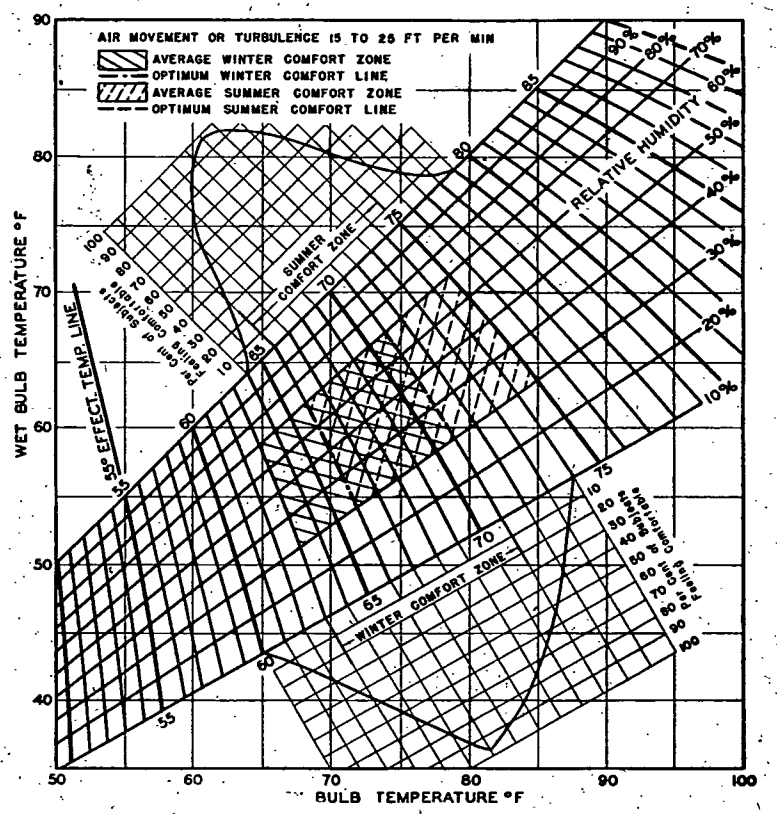


FIG. 6. A.S.H.V.E. COMFORT CHART FOR STILL AIR

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station 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 optimum 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.

people uncomfortable would not be acceptable. Systems should be designed to assure comfort for the maximum possible number.

The results of recent tests [25] made at the A.S.H.V.E. Research Laboratory in very hot conditions with subjects doing light work were in very close agreement with the effective temperature chart. Other recent work [21] 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.

Radiation between the occupant of an enclosure and the surfaces of the

room itself and objects within the room, including windows, heating and cooling equipment, and other occupants, has an important bearing on the feeling of warmth and may alter to some measurable degree the optimum conditions for comfort previously indicated. Fig. 7 [29] shows the elevation in the dry-bulb temperature of the air necessary to compensate for the lower temperature of three of four side-wall surfaces, and indicates that for this condition each degree reduction in the average of the three wall surface temperatures requires an elevation of 0.3 deg in the dry-bulb temperature of the air. Studies by the A.S.H.V.E. Research Laboratory [28] on the effect of radiation within an enclosure, including the effect of panel heating, indicate that each degree elevation or depression of the mean

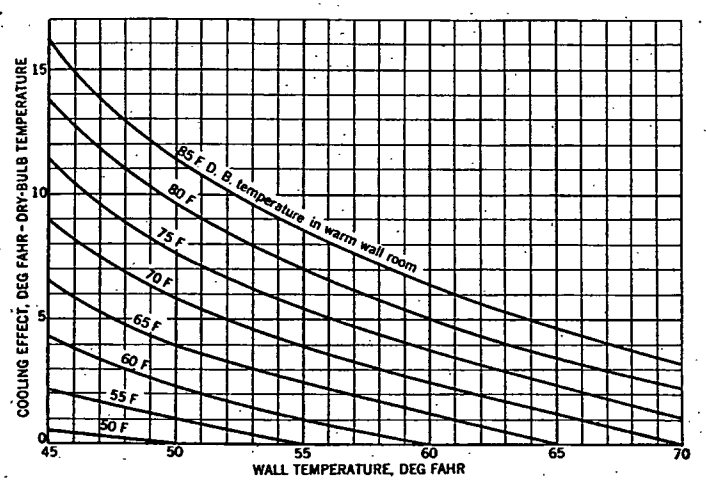


FIG. 7. COOLING EFFECT OF THREE COLD WALLS IN A SMALL EXPERIMENTAL ROOM, AS DETERMINED BY COMPARISON WITH SENSATIONS IN A ROOM OF UNIFORM WALL AND AIR TEMPERATURE

radiant temperature above or below the air temperature requires about 0.5 deg counterchange in effective temperature of the air. 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 imbedded 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.

More recent tests [25] made at the A.S.H.V.E. Research Laboratory indicate that for high effective temperatures the correction in ET necessary per degree rise in mean radiant temperature is not a constant, but varies with the values of effective temperature and mean radiant temperature. At effective temperatures near the upper limit of human endurance, the effect of radiant heat is very small. Additional studies on the physiological effects of radiant heating are contemplated.

A long series of studies have been made to determine the optimum effective temperature for comfort of normal persons in both winter and summer air conditioned space, in different geographical regions and for different age groups of men and women. A group of these studies[30] was made between 1935 and 1940 by the A.S.H.V.E. Research Laboratory in Pittsburgh, and in several metropolitan districts of the United States and Canada in cooperation with the managements of offices employing large numbers of workers. Some of the results are shown in Fig. 8. Taking all of these studies together, women of all age groups studied indicate an average effective temperature for comfort 1.1 deg higher than for men. All men and women, beyond the age of 40 years, show an average desire

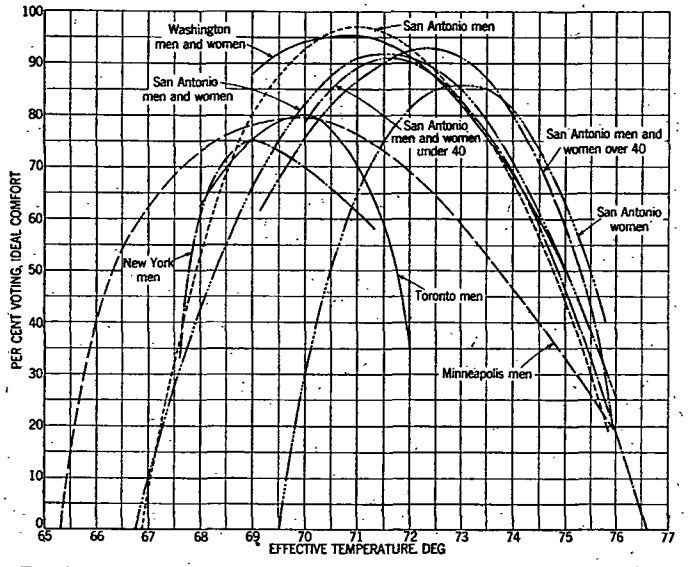


FIG. 8. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OBSERVATIONS INDICATING COMFORT

for 0.9 deg ET higher than those below this age, while the men and women, respectively, beyond 40 desired effective temperatures of 0.8 and 1.2 deg higher than those below 40. The persons serving in all of these studies were representative of office workers clothed for air conditioned space in the summer season and engaged in the customary sedentary activity of office workers.

On the basis of present knowledge, for different geographical regions and age groups, the total spread in optimum comfort conditions ranges from a low of 66 deg ET for winter heating and air conditioning, to a high of 73 deg ET for summer cooling and air conditioning.

The spread for summer cooling and air conditioning for optimum comfort is confined entirely to an effective temperature range of from 69 to 73 deg, and it may be presumed that for winter conditioning a like spread would exist; while for inter-seasonal conditions there will be a fluctuation between these two ranges.

Laboratory studies[28] indicate that for the average individual a temperature change of about 3 deg ET is required to change a person's sensation from *ideally comfortable* to *cool* or *warm*. From this it may be observed that necessary variations in the effective temperature of air conditioned space for optimum comfort for most persons need little differentiation for either the winter season or the summer season, and not more than about 4 deg on the average between seasons.

Acclimatization and habits of clothing and diet account for these variations. An analysis[31] of most of the material published up to 1942, made by the A.S.H.V.E. Technical Advisory Committee on Sensations of Comfort, indicates a spread of approximately 3 deg in the optimum effective temperature for summer cooling and air conditioning due to geographical location. However, it should be recognized that variations in sensation of comfort among individuals may be greater for any given location, as shown in Fig. 8, than variations due to a difference in geographical location. The available information indicates rather clearly that changes in weather conditions over a period of a few days do not acclimate people to a desire for different indoor conditions, but in general, people experiencing low temperatures over an extended period of time become acclimated to desiring lower indoor temperatures, while those experiencing higher temperatures become acclimated to a desire for higher indoor temperatures. It is obvious that a person spending a considerable portion of his time in space conditioned to his comfort will become acclimated to his indoor environment. While few people spend more than a small percentage of the total time within an air conditioned enclosure, there is some evidence that persons experiencing summer comfort air conditioning a large part of the time tend to become acclimated to about 70 or 71 deg ET.

The sudden sensation of coldness felt by persons entering a cooled and air conditioned space during the summer months, and often referred to as shock, may at times be important. It is due to the rapid evaporation of perspiration which accumulated on the skin during previous subjection to hot and humid outside conditions. While studies[32] have shown that for healthy individuals this *shock* is usually a pleasant experience, it is plausible, but not proven, that under some conditions it may result in unpleasant or even harmful cooling. Where a large number of occupants may enter for only a short time, 15 min or less, such occupants may be satisfied with less cooling. For long occupancy very little deviation from the optimum effective temperature is desirable.

An exit shock when leaving air conditioned space and entering a warm atmosphere is equally plausible. Experiments at the A.S.H.V.E. Research Laboratory[33] indicated no demonstrable harm to a healthy individual. Complete acclimatization occurred as soon as normal perspiration was established in 5 to 15 min and mild exercise shortened the time required.

Satisfactory comfort conditions for persons at work[34] are found to 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 optimum comfort. However, work by the A.S.H.V.E. Research Laboratory[35] indicates that under certain conditions moderate activity on the part of a person standing up and moving about may result in a slightly higher optimum effective temperature than for a person seated at rest, because of the larger body surface area exposed for heat elimination and the increase in effective air movement over his body. Where few workers occupy a large space in hot industries, work by the

A.S.H.V.E. Research Laboratory [22] shows that they may be made reasonably comfortable by blowing relatively small volumes of slightly cooled air over them or through their clothing.

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 per cent [36]. No data are yet available on the optimum air conditions for full term infants and

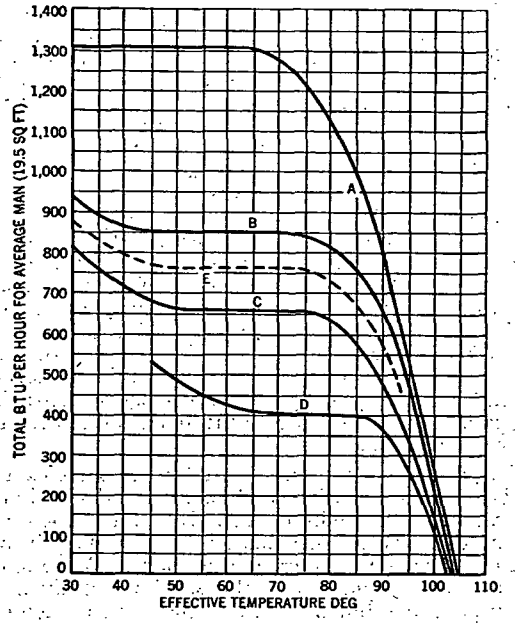


FIG. 9. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Persons working so as to have a metabolic rate of 1310 Btu per hour. Curve *B*—Persons working so as to have a metabolic rate of 850 Btu per hour. Curve *C*—Persons working so as to have a metabolic rate of 660 Btu per hour. Curve *D*—Persons seated at rest, or with a 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 deg and extrapolation of Curves *B* and *D*. All curves are averages of values for high and low relative humidities which apply with satisfactory accuracy for most considerations. For special problems requiring a higher degree of accuracy see more detailed A.S.H.V.E. Research Laboratory reports.

young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 to 68F with natural indoor humidities. For children (having high metabolism) at school, in winter clothes, 70 F has been considered correct, while in a gymnasium 55 F has been recommended. A great number of persons seem to be fairly content in summer with a higher plane of indoor temperature, particularly when the matter of first cost and operating cost of a cooling plant is given due consideration. Studies by the University of Illinois [37] in cooperation with the A.S.H.V.E. Committee on Research indicate that effective temperatures as high as 74.5 deg are acceptable in the living quarters of a residence, and while this condition is not representative of optimum comfort it

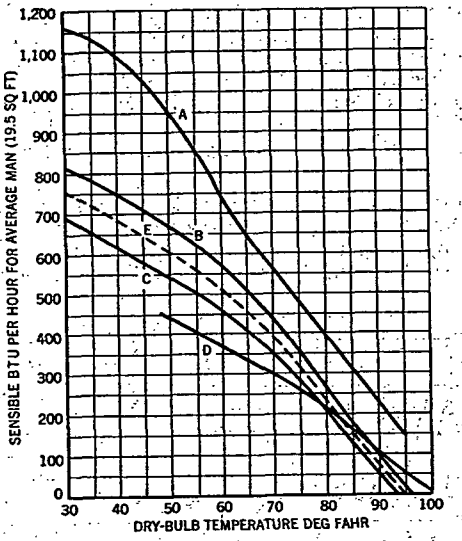


FIG. 10. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

[a]Loc. Cit. See footnote a, Fig. 9.

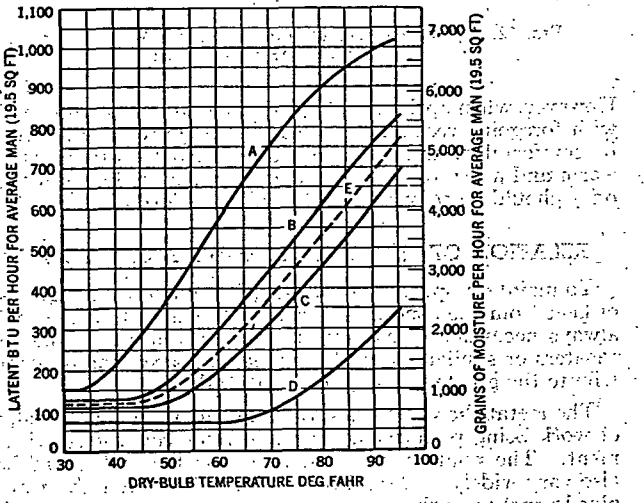


FIG. 11. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Loc. Cit. See footnote a, Fig. 9.

provides sufficient relief in hot weather to be acceptable to the majority of users. It should be emphasized, however, that these are borderline cases that may be acceptable largely in the interest of economy. Comprehensive studies by the A.S.H.V.E. Research Laboratory [30] in cooperation with office staffs in widely distributed regions, including San Antonio, Minneapolis, Washington, D. C., and New York City (see Fig. 7), show conclusively that lower effective temperatures are required for optimum comfort.

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated, it should not be expected that all the occupants of a room will feel perfectly comfortable.

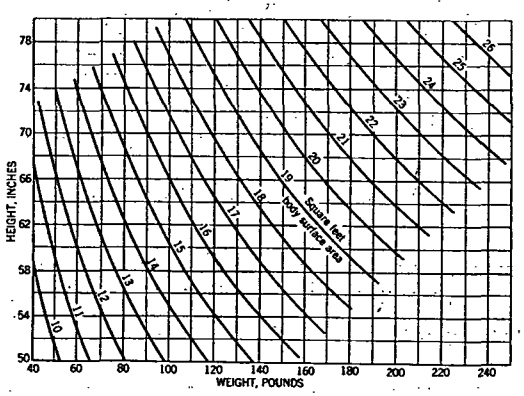


FIG. 12. CHART FOR DETERMINING SURFACE AREA OF INDIVIDUALS FOR HEIGHT AND WEIGHT GIVEN

However, when optimum comfort temperatures are applied in accordance with foregoing recommendations, the majority of the occupants should be comfortable, and it should be expected that there will be a few too warm and a few too cold. These individual differences among the minority should be counteracted by suitable clothing.

RELATION OF AIR CONDITIONING NEEDS TO METABOLISM

To maintain optimum conditions in summertime requires the removal of heat from the conditioned space. In calculating the cooling load it is always necessary to consider the heat given up by the occupants. In theaters or similarly densely populated spaces, the occupants may contribute the greater part of the total load.

The metabolic rate varies with the size of the individual, with the rate of work being performed, and at extreme conditions, with the environment. The relative proportions of sensible and latent heat given off also vary widely. The curves in Fig. 9 show the total heat loss (sensible plus latent) from the average man for four different rates of work. Heat loss in this figure is plotted against effective temperature. Figs. 10 and 11 give the sensible and latent losses for the same four work rates, plotted against dry-bulb temperature. By proper interpolation, it is possible

TABLE 4. RELATION BETWEEN METABOLIC RATE AND ACTIVITY [a]

ACTIVITY	HOURLY METABOLIC RATE FOR AVG PERSON OR TOTAL HEAT DISSIPATED, BTU PER HOUR	HOURLY SENSIBLE HEAT DISSIPATED, AT 79 F, BTU PER HOUR	HOURLY LATENT HEAT DISSIPATED, AT 79 F, BTU PER HOUR	MOISTURE DISSIPATED PER HOUR PER PERSON	
				Grains	Pounds
Basal	291	145	145	978	0.140
Seated at Rest	384	225	159	1072	0.153
Reading Aloud (Seated)	420	225	195	1315	0.188
Standing at Rest	431	225	206	1389	0.198
Hand Sewing (Seated)	441	225	216	1457	0.208
Knitting 23 stitches per minute on Sweater	462	225	237	1598	0.228
Dressing and Undressing	468	225	243	1639	0.234
Tailor	482	225	257	1733	0.248
Singing	486	225	261	1760	0.251
Office Worker Moderately Active	490	225	265	1787	0.255
Light Work Standing	549	225	324	2185	0.312
Typewriting Rapidly	558	225	333	2246	0.321
Ironing with 5 lb iron	570	225	345	2326	0.332
Dishwashing—Plates, Bowls, Cups and Saucers	600	225	375	2529	0.361
Clerk Moderately Active Standing at Counter	600	225	375	2529	0.361
Book Binder	626	225	401	2704	0.386
Shoemaker	661	225	436	2940	0.420
Sweeping Bare Floor 38 Strokes per Minute	672	229	443	2987	0.427
Pool Player	680	230	450	3055	0.434
Walking 2 mph, Light Dancing	761	250	511	3446	0.492
Light Metal Worker (at Bench)	862	277	585	3945	0.564
Painter of Furniture (at Bench)	876	280	596	4019	0.574
Carpenter	954	307	647	4363	0.623
Restaurant Serving	1000	325	675	4552	0.650
Pulling Weight	1041	335	708	4774	0.682
Walking 3 mph	1050	339	711	4795	0.685
Walking 4 mph, Active Dancing, Roller Skating	1390	452	938	6325	0.904
Walking Down Stairs	1444	467	977	6588	0.941
Stone Mason	1490	485	1005	6777	0.968
Bowling	1500	490	1010	6811	0.973
Man Sawing Wood	1800	590	1210	8160	1.166
Swimming	1986	-----	-----	-----	-----
Running 5.3 mph	2268	-----	-----	-----	-----
Walking 5 mph	2330	-----	-----	-----	-----
Walking Very Fast 5.3 mph	2580	-----	-----	-----	-----
Walking Upstairs	4365	-----	-----	-----	-----
Maximum Exertion Different People	3000-4800	-----	-----	-----	-----

[a] These metabolic rates were compiled by the A.S.H.V.E. Research Laboratory from actual tests, from other authoritative sources, and from estimates based upon various considerations. Division of the total heat dissipation into latent and sensible rates is based on actual test data and on various considerations for metabolic rates up to 1250 Btu per hour, and extrapolated for higher rates. Values for total heat dissipation for a person at rest apply for a dry-bulb temperature range from approximately 60 to 90 F; for other than rest conditions the values apply for a similar but lower temperature range. Below these temperature ranges metabolic rates and total rates of heat dissipation increase, while above these ranges metabolic rates increase slightly and total heat dissipation rates decrease rapidly. Division of total dissipation rates into sensible and latent heat holds only for a dry-bulb temperature of 79 F. For lower temperatures, sensible heat dissipation increases and latent heat decreases, while for higher temperatures the reverse is true.

from Figs. 9, 10 and 11, to determine the sensible and latent heat dissipation for any work rate and any environmental condition, provided the metabolic rate is known at one effective temperature. For example, if it is found that a certain type of work results in a metabolic rate of approximately 760 Btu per hour for an average person working in an atmosphere of 70 ET, then this total rate of heat dissipation to atmospheres of various temperatures will be approximately as given by the broken-line curve in Fig. 9. The broken line curves in Figs. 10 and 11 give the rate of sensible and latent heat dissipation of the person for different dry-bulb temperatures. The metabolic rates for a number of types of

work are given in Table 4, and may be plotted in this manner on the figures.

It should be noted that metabolic rates are proportioned to the body surface areas of the persons considered, and that the data given in Table 4 and Figs. 9, 10 and 11 are for persons having an average surface area of 19.5 sq ft, or that of the average American male 5 ft 8 in. high and weighing 150 lb. Therefore, these data will not apply to many audiences made up largely of women or younger persons. Fig. 12, taken from the work of DuBois [38], gives the relation of body surface area to height and weight, and may serve to correct the data for groups of various types.

REFERENCES

1—Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27).

2—A.S.H.V.E. RESEARCH REPORT No. 959—Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261).

3—A.S.H.V.E. RESEARCH REPORT No. 1031—Ventilation Requirements, by C. P. Yaglou, E. C. Riley and D. J. Coggins (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 133).

4—Loc. Cit. Note 3, p. 156.

5—A.S.H.V.E. RESEARCH REPORT No. 921—Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 191). A.S.H.V.E. RESEARCH REPORT No. 965—Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt and L. C. Benjamin (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 357). A.S.H.V.E. RESEARCH REPORT No. 985—Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 271). The Nature of Ions in Air and Their Possible Physiological Effects, by L. B. Loeb (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 101). The Influence of Ionized Air upon Normal Subjects, by L. P. Herrington (*Journal Clinical Investigation*, 14, January, 1935). The Effect of High Concentrations of Light Negative Atmospheric Ions on the Growth and Activity of the Albino Rat, by L. P. Herrington and Karl L. Smith (*Journal Industrial Hygiene*, 17, November, 1935). Subjective Reactions of Human Beings to Certain Outdoor Atmospheric Conditions, by C.-E. A. Winslow and L. P. Herrington (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 119).

6—*The British Medical Journal*, Editorial, June 25, 1932, p. 1182. See also Loc. Cit. Note 8.

7—Air-Borne Infection and Sanitary Air Control, by W. F. Wells (*Journal Industrial Hygiene*, November, 1935).

8—Sanitary Ventilation in Wards, by W. F. Wells (*Heating and Ventilating*, April, 1939, p. 26). Measurement of Sanitary Ventilation, by W. F. Wells (*American Journal of Public Health*, Vol. 28, 1938, p. 343).

Air Disinfection in Ventilation, by W. F. Wells (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 361).

Principles of Ultraviolet Disinfection of Enclosed Spaces, by L. J. Buttolph (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1945, p. 282).

Ultraviolet as a Protective Agent in Preparation of Pharmaceuticals, by A. R. Dennington (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1945, p. 398).

9—The Use of Glycol Vapors for Air Sterilization and the Control of Air Borne Infection, by B. H. Jennings, Edward Bigg, M.D., and F. C. W. Olson (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 343).

10—A.S.H.V.E. RESEARCH REPORT No. 830—Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 245). Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (*American Journal of Physiology*, Vol. 88, 1929, p. 386). A.S.H.V.E. RESEARCH REPORT No. 908—Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 541). Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (*American Journal of Hygiene*, Vol. XIII, 1931, No. 2, p. 415). A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59).

11—A.S.H.V.E. RESEARCH REPORT No. 1107—Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice, by C.-E. A. Winslow, T. Bedford, E. F. DuBois, R. W. Keeton, A. Missenard, R. R. Sayers and C. Tasker. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 111).

12—The Relative Influence of Radiation and Convection Upon the Temperature Regulation of the Clothed Body, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology*, Vol. 124, October, 1938, p. 51).

13—Relations Between Atmospheric Conditions, Physiological Reactions, and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Hygiene*, Vol. 26, July, 1937, p. 102). The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology*, Vol. 124, December, 1938, p. 692).

14—A.S.H.V.E. RESEARCH REPORT No. 654—Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 129). A.S.H.V.E. RESEARCH REPORT No. 672—Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 353). A.S.H.V.E. RESEARCH REPORT No. 690—Air Motion, High Temperatures and Various Humidities—Reactions on Human Beings, by W. J. McConnell, F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 167). A.S.H.V.E. RESEARCH REPORT No. 718—Work Tests Conducted in Atmospheres of High Temperatures and Various Humidities in Still and Moving Air, by W. J. McConnell and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 101). A.S.H.V.E. RESEARCH REPORT No. 719—Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by W. J. McConnell, C. P. Yaglou and W. B. Fulton (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 123). A.S.H.V.E. RESEARCH REPORT No. 908—Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 541). A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry—Physiological Reactions of Individual Workers to High Effective Temperatures, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59). A.S.H.V.E. RESEARCH REPORT No. 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 185). Physiologic Effects of Hot Atmospheres, by F. C. Houghten, M. B. Ferderber and A. A. Rosenberg (*Industrial Medicine*, January, 1940, p. 7).

15—A.S.H.V.E. RESEARCH REPORT No. 1151—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 157).

16—A.S.H.V.E. RESEARCH REPORT No. 1153—Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 185).

17—Influence of Effective Temperature upon Bacterial Action of Gastro-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp. Biol. Med.*, Vol. 24, 1927, p. 823).

18—Manual of Industrial Hygiene, 1943 (Prepared by U. S. Public Health Service, p. 328). Introduction to Industrial Medicine, 1944 (University of Pittsburgh, p. 167).

19—Heat Disease: Clinical and Laboratory Studies, by M. W. Heilman and E. S. Montgomery (*Journal of Industrial Disease and Toxicology*, 18:651, 1936).

20—A.S.H.V.E. RESEARCH REPORT No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 59).

21—The Upper Limits of Environmental Heat and Humidity Tolerated by Acclimatized Men Working in Hot Environments, by Ludwig W. Eichna, William F. Ashe, William B. Bean, and Walter B. Shelley (*The Journal of Industrial Hygiene and Toxicology*, Vol. 27, March, 1945, p. 59).

22—A.S.H.V.E. RESEARCH REPORT No. 1188—Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 403).

23—Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1928.

24—The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (*American Journal of Physiology*, Vol. 124, December, 1938, p. 692).

25—Physiological Response of Subjects Exposed to High Effective Temperatures and Elevated Mean Radiant Temperatures, a Report to the Air Conditioning Section, Design Division, Bureau of Ships, U. S. Navy. (A paper on the work will be presented early in 1946).

26—A.S.H.V.E. RESEARCH REPORT No. 673—Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361). A.S.H.V.E. RESEARCH REPORT No. 691—Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193). A.S.H.V.E. RESEARCH REPORT No. 717—Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 89). A.S.H.V.E. RESEARCH REPORT No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926, p. 315).

27—How to Use the Effective Temperature Index and Comfort Charts, by C. P. Yaglou, W. H. Carrier, Dr. E. V. Hill, F. C. Houghten and J. H. Walker (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 410).

28—A.S.H.V.E. RESEARCH REPORT No. 1172—Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst and J. Suciu, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 93).

29—A.S.H.V.E. RESEARCH REPORT No. 946—Cold Walls and Their Relation to Feeling of Warmth, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 83).

30—A.S.H.V.E. RESEARCH REPORT No. 1035—Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 215). A.S.H.V.E. RESEARCH REPORT No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145). A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337). Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 549). A.S.H.V.E. RESEARCH REPORT No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 459). A.S.H.V.E. RESEARCH REPORT No. 1136—Summer Cooling Requirements in Washington, D.C., and Other Metropolitan Districts, by F. C. Houghten, Carl Gutberlet and Albert A. Rosenberg (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 577). A.S.H.V.E. RESEARCH REPORT No. 1160—Reactions of 745 Clerks to Summer Air Conditioning, by W. J. McConnell and M. Spiegelman (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 291).

31—A.S.H.V.E. Research Paper—Comfort with Summer Air Conditioning, by 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. (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 107).

32—A.S.H.V.E. Research Report No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson (A.S.H.V.E. Transactions, Vol. 44, 1938, p. 571).

33—A.S.H.V.E. Research Report No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 145).

34—A.S.H.V.E. Research Report No. 755—Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. Transactions, Vol. 32, 1926, p. 315).

35—A.S.H.V.E. Research Report No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber (A.S.H.V.E. Transactions, Vol. 45, 1939, p. 59).

36—Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S.H.V.E. Transactions, Vol. 36, 1930, p. 383).

37—A.S.H.V.E. Research Report No. 1012—Study of Summer Cooling in the Research Residence for the Summer of 1934, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 207).

38—DuBois, D. and E. F. (*Archives of Internal Medicine*, 1916, Vol. 18, p. 865).

CHAPTER 13

Air Conditioning in the Prevention and Treatment of Disease

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, Evacuation of Sick and Wounded, General Hospital Air Conditioning

THE late war has caused an increase of interest in the preventive aspects of air conditioning. It has re-emphasized the importance of the control of airborne infection and has demonstrated the value of air cooling under tropical conditions for the prevention of heat rash, for proper rest and sleep, and in the convalescence of patients.

CONTROL OF AIRBORNE INFECTION

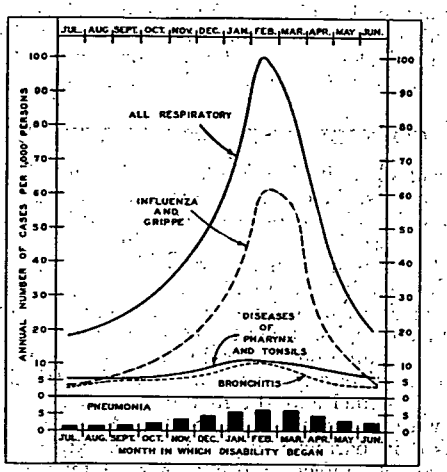
Any program of air sanitation is influenced by a number of factors[1, 2, 3, 4]. In the winter months, the closing of doors, windows and other means of access to the outside air to conserve warmth and further the crowding of persons indoors provides conditions conducive to a high incidence of contagion. This seasonal phenomenon, illustrated in Fig. 1 which represents a study made by the U. S. Public Health Service, will concern the ventilating engineer in so far as air quality, determined by temperature, humidity, air replenishment and type of air movement and by 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 in comparison with permanent personnel[5] as shown by studies of military personnel housed in barracks, Fig. 2. 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 microorganisms in droplet-nuclei and droplet-dust. Thus, control measures may eliminate consistently 90 per cent of airborne organisms in laboratory tests, but cannot effect a decrease in actual incidence of infection exceeding 30 per cent. Thirty per cent 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 and its effectiveness and applicability in actually reducing the incidence of airborne disease.

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 [6].

Intensive studies on air disinfection have indicated two distinct control measures, (*a*) suppression of dust and lint, and (*b*) disinfection of droplet-nuclei.

Well controlled, large scale tests of the various methods of air sterilization conducted in barracks [7, 8] 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 per cent in



Occurrence of diseases causing disability for 8 consecutive days or longer in a group of 100,000 wage earners (10 per cent women) in different industries.

[a]Graph obtained from Dean K. Brundage, U. S. Public Health Service.

FIG. 1. STUDY OF AVERAGE MONTHLY FREQUENCY (1921-1926 INCLUSIVE) OF SPECIFIED RESPIRATORY DISEASES[a]

Army barracks and station hospitals [8]. The incidence of acute respiratory infections was from 10 to 30 per cent lower in barracks with oiled floors and bedclothes than it was in control barracks which received no special treatment.

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 per cent [9, 10, 11]. Under practical conditions, however, 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.

Ultraviolet radiation of floors and upper air has proved more effective

than glycol aerosols in reducing respiratory illness. In barracks housing naval recruits, hospital admissions for respiratory infections (mostly *catarrhal fever*) were 25 per cent 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 [12]. A combination of ultraviolet radiation and dust control measures is believed to be more effective than when either one of the two is used alone, but the proof for this has yet to come.

VALUE OF AIR COOLING UNDER TROPICAL CONDITIONS

The commissioning of a class of naval hospital ships with all wards, laboratories and living spaces air cooled is a notable achievement to provide better treatment of patients, especially those suffering from extensive burns, by control of environmental factors. Although statistics

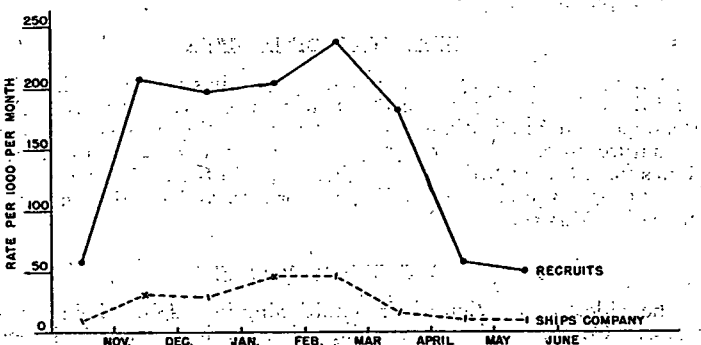


FIG. 2. MONTHLY INCIDENCE OF ACUTE RESPIRATORY ILLNESS AMONG NAVAL RECRUITS AND SHIP'S COMPANY (PERMANENT PERSONNEL) [5]

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 [13] 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.

Frequently from 50 to 75 per cent 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 [13] 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 which 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 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 [14, 15, 16].

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. It is not improbable that cooled houses in a tropical climate, if used consistently for one generation, might modify the whole character of a population [17]. The obvious advantages of part time cooling on 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, the control of allergic disorders, and for the physiological effects in industry.

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 of anesthetic gases; summer cooling with some dehumidification is needed to eliminate excessive fatigue and to protect the patient and operating personnel; and finally, filtering for the removal of allergens from the operating room air.

Reducing Explosion Hazard

Explosion hazards in operating rooms began with the introduction of modern 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 [18].

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 [19]. 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

per cent ethylene and 20 per cent 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 accumulate to any lower concentration, 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 [20] of 230 anesthetic explosions and fires, 70 per cent of the explosions and 60 per cent of the deaths were caused by igniting agents other then static sparks. In 1941 the *National Fire Protection Association* [21] made certain recommendations for safe practice based on available information. Some of these recommendations are:

Windows should be kept closed so that the air conditioning system can prevent pooling of explosive anesthetic gases. Twelve air changes per hour and a humidity of 55 per cent are advised. If a higher humidity were compatible with the well being of the patient and personnel, it should be maintained. All electrical installations should comply with the standards set by the *National Electrical Code* for use in explosive situations. Cautery equipment should not be used in hazardous locations. To prevent static sparks, all bodies in an operating room should be conductive or coupled. It is essential that adequate grounding be provided for the floor and every object in the operating room. Conductive rubber should be used on shoes, leg tips, operating table coverings and all rubber parts of the anesthesia equipment. All furniture in contact with the floor should be metal. In the absence of complete grounding facilities, the simple method of intercoupling patient, operating table, anesthetist and gas machine at ground potential may be used.

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.

It should be realized that when a room and the occupants have been completely grounded there is always the possibility that the patient or the operator might receive a dangerous shock if a short circuit developed in any of the electrical equipment.

A comprehensive study of the explosion problem and of the general causes and prevention of operating room hazards by the *University of Pittsburgh*, the A.S.H.V.E. 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 replacement 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 variation in the rectal temperature during the course of the operation[22]. 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. A comparison of surgeons' statements who operate in both air conditioned and non-air conditioned rooms strongly indicates that the recuperative power of the patient is greater when operated upon in air conditioned rooms[22].

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 per cent relative humidity and 72 to 80 F temperature. The work just cited reported that 68 to 70 F effective temperature not only furnished comfort for the operating room workers, but apparently prevented exhaustion of the patient as evidenced by rapid convalescence in the recovery ward. Additional heat may be furnished to the patient 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.

In an investigation recently conducted at the University of Pittsburgh, in a cooperative research program with the Society, comparative studies were made on bacterial content of conditioned and non-conditioned operating rooms. From these studies[23] it was concluded that the bacterial content of conditioned operating rooms was considerably less 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[24] without replenishment. The removal of bacteria by the process of air cooling and condensation of moisture out of air merits further study[25].

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[26].

Operations may be postponed on allergic patients during asthmatic manifestations through fear of complications. The removal of air-borne allergens, therefore, is in some cases an important function of the air conditioning system in preparing patients for operation.

Central system air conditioning plants and unit air conditioners prove satisfactory in operating rooms when producing between 8 and 12 air changes per hour of filtered and properly conditioned air without recirculation during the course of anesthesia. A separate exhaust fan system is usually necessary to confine and remove the gases and odors. Double windows are desirable and often necessary to prevent condensation and

frosting on the glass in cold weather and to minimize drafts. The air flow of 8 to 12 air changes in operating rooms should: (1) reduce the concentration of the anesthetic to well below the pharmacologic threshold in the vicinity of the operating personnel, (2) remove the great amounts of heat and sometimes moisture, from sterilizing equipment if inside the operating room, from the powerful surgical lights, from solar heat, and from the bodies of the operatives, and (3) provide extra capacity for quickly preparing the room for emergency operations. Much can be gained by thermal insulation of sterilizing equipment and by thorough exhaust ventilation of sterilizing rooms adjoining the operating rooms.

Too great a difference in temperature between the operating room and the final hospital destination of the patient, including corridors and elevators, is conducive to infections of the upper part of the respiratory tract and post-operative pneumonia. A suggested remedy is a recovery ward in which conditions closely approximate those of the operating room and in which the patients remain from one to four days. Satisfactory conditions in the recovery ward not only hasten convalescence, but dispel the fear frequently found in patients who must undergo operations during the hot seasons[27].

Experience has shown that a few hours after the operation the temperature of the post-operative room can be decreased a few degrees below that of an overheated operating room to stimulate recovery.

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 their heat regulating systems are not fully developed; the metabolism is low and the infants generally exhibit marked inability to maintain normal body temperatures. The resistance to infection is low and mortality rate high.

Air Conditioning Requirements

The optimum air conditions for the growth and development of these infants were determined by extensive research[28] 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 per cent, and the air movement less than 20 fpm.

A single nursery conditioned to 77 F and 65 per cent 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, and the temperature adjusted according to individual requirements. In this way 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

per cent 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 per cent 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 per cent of the birth weight; in the conditioned nurseries it was 8.9 per cent with 25 to 49 per cent relative humidity, and 6.0 per cent with 50 to 75 per cent 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 per cent) in infants weighing less than 5 lb. The gains were less under low humidity (25 to 50 per cent) in the same nurseries, and in the old nurseries prior to the installation 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 than under high humidity.

Summarizing, the best chances for life in premature infants are created by maintaining a relative humidity of 65 per cent 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, 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 the spread of respiratory cross-infections. Bactericidal ultraviolet light barriers and air conditioning or mechanical barriers and air conditioning were efficient [29].

FEVER THERAPY

Artificial production of fever in man is 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, which retard or neutralize the activity of pathogenic bacteria and their toxins.

Although the action may be direct and specific by destruction of the invading organisms within the safe limits of human tolerance, 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. Clinically, it has been shown that there is no change in the per cent of phagocytes which engulf bacteria in patients during fever therapy, although the action of the complement fixing antibodies may be temporarily diminished.

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 spasm, 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 from a few hours to 50 or more.

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, chorea, infectious arthritis (non-gonorrheal), encephalitis, and some forms of asthma. There are other conditions which show promise under this treatment; but the most striking results are seen in gonorrhea and syphilis, since the causative organisms can be destroyed at temperatures compatible with human life.

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 [30, 31]. 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 [32], temperatures were elevated more easily using saturated atmospheres. A fever therapy apparatus [33] 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, 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.

Some investigators employ short wave diathermy within the cabinet during the induction phase. When the optimum 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 mechanism of heat loss by evaporation is markedly decreased.

The chief physiologic effects exhibited during fever therapy are: (*a*) the cardio-vascular system is stimulated to increase the amount of blood to various parts of the body, (*b*) a mobilization of blood elements by the blood forming system, (*c*) general metabolism is increased, (*d*) only slight changes in blood chemistry except that the chlorides may be depleted if dehydration occurs.

The therapeutic value of fever therapy for certain diseases has been definitely proved. The combination of chemo-therapy and fever has proved to be more effective than when either is employed separately. This is especially true in regard to gonorrhea and early syphilis.

COLD THERAPY

In contrast to fever therapy the use of cold as a means of anesthesia and treatment is of established importance. Refrigeration anesthesia is being more widely used and the advantages of the method claimed by Allen have been completely confirmed [34].

It has been demonstrated that the cooling of limbs and other parts with ice-water or ice, cracked or pulverized, down to near the freezing point (5 C or 40 F) is harmless. Freezing must be avoided. There is a temporary retardation or suspension of life, with resumption of cellular activity as the temperature returns to normal. A human limb can remain bloodless and anesthetic below a tourniquet for at least 8 hr and perhaps up to 48 hr without injury, while the rest of the body remains warm. Where amputation is indicated it can thus be done without pain, loss of blood or strength and also without shock. There is no apparent interference with the subsequent healing of the stump. Refrigeration anesthesia is of importance not only in amputation but also in the control of hemorrhage, pain, infection and shock during the transportation of patients with traumatized limbs.

Apart from the advantages of cold in amputations, cryotherapy has been beneficial in the treatment of burns and arterial obstruction, and essential in the treatment of frostbite and immersion foot. It has been used successfully for dental anesthesia. The principle of hibernation in which the body as a whole is cooled for as much as five days in air temperatures between 50 to 60 F and applied to such conditions as morphine addiction, leukemia, and schizophrenia continues to be experimental. Prolonged cold therapy which keeps the body temperature below 95 F depresses the vital processes and is fraught with danger.

The methods used for refrigeration depending upon available facilities are as follows [34]:

(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

Although there is some division of opinion over the ultimate cause of allergy, the prevailing belief is that it is due to an inherited or acquired hypersensitiveness to pollen or other foreign proteins in certain individuals who react abnormally to the offending substance. The reaction may be induced by inhalation, eating, or absorption (through the skin) of the allergens. Some of the clinical manifestations are hay fever, asthma, eczema, and contact dermatitis.

Symptoms of Hay Fever and Asthma

The respiratory tract is the usual site of allergic manifestations, *e.g.* hay fever and asthma. In hay fever, the nose and eyes are red and itchy, and there is considerable discharge. Nasal obstruction is the most common and distressing symptom. The severity of the symptoms varies widely from day to day depending chiefly on the amount of pollen in the air.

Seasonal asthma comes in attacks. The most popular theory concerning the mechanism of action is that the offending substance irritates the nerve endings in mucous membranes of the respiratory tract, causing spasmodic contraction of the small bronchioles of the lungs, which interferes with breathing, particularly with expiration. Non-seasonal allergic disturbances are sometimes attributed to house or street dusts, fungi, odors, animal dander, irritating gases, and heat or cold, particularly sudden temperature changes. It is often stated in the literature that heat regulation in asthmatic individuals tends to be unstable, with a tendency toward the subnormal. Many allergic cases who are apparently well develop their attacks when cold weather appears, or upon changing from warm to cool outdoor air.

Air Conditioning Apparatus

In recent years considerable effort has been directed toward the elimination of the principal cause of allergy from the air of enclosures by filtration or other air conditioning processes capable of removing pollens, in the hope of providing relief to individuals who fail to respond to medical treatment (desensitization or immunization).

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 [35].

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 per cent proved satisfactory [36]. 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. But while specific desensitization is preventive and in a few instances curative, 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 days. 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 air-borne 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 [37]. 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 the principal measure employed for preventing and relieving the distressing symptoms of anoxemia, which is a deficiency in the oxygen content of the blood. Some of the more important conditions in which oxygen treatment is believed to be beneficial are pneumonias, anemia, heart affections, post-operative pulmonary disturbances, certain mental disturbances, asphyxia, asthma and atelectasis in new-born infants.

The effectiveness of oxygen therapy depends upon the manner of administration. Three common methods, catheter, face mask or tent [38, 39] are employed.

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. The oxygen rich atmosphere in these enclosures is therefore reconditioned in a closed circuit by removal of excess heat, moisture, and carbon dioxide given off from the occupants being treated.

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 hour 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 per cent 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.

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 [40] prescribed conditions should be an effective temperature of 66 to 68 F, humidity of 50 per cent, air movement of not less than 50 linear feet per minute, oxygen concentration of 50 per cent, and carbon dioxide of less than one per cent.

Oxygen in Aviation

An important application of the principle of oxygen therapy is in aviation. At the present time all high altitude military airplanes in this country are 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. The use of oxygen in commercial aviation will depend on the height and duration of the flights as well as the state of health of the passengers. The necessity for portable, comfortable equipment, the possible fire hazards due to smoking, and the use of oxygen on sleeper planes are some of the difficulties facing civil airline operators. The pressure cabin airplane is a solution to the problem.

AIR EVACUATION OF THE SICK AND WOUNDED

Between November 1942 and June 1945, some 510,000 wounded or sick soldiers were transported by air. So efficient was the service that in the Normandy campaign men wounded in France in the morning could be given operative treatment in English hospitals the same evening. A major problem encountered in this type of service is the removal of odors in the hospital space. The ideal deodorant method for use in aircraft should effect complete removal of unpleasant odors without vitiating cabin air with other odors or irritants in the form of vapor sprays or

perfumes. It should not only be light weight and portable but capable of utilization in pressure type cabins without imposing an additional burden upon the limited ventilation facilities.

GENERAL HOSPITAL AIR CONDITIONING

Complete conditioning of a large hospital involves a capital investment and running expenses which may not be justified. Air conditioning has important applications in certain hospital wards such as nurseries for premature infants, oxygen therapy chambers, heat therapy rooms or cabinets, allergy wards [27], operating rooms and recovery wards. 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; it 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 window 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.

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, and in a variety of other ailments which often accompany summer heat waves.

Considerable research is in progress on the influence of air conditioning upon a wide variety of diseases such as pneumonia, upper respiratory diseases, tuberculosis, arthritis, nervous instability, hyper-thyroidism, essential hypertension, skin diseases, and vascular disorders.

Problem of Odors

The evacuation of battle casualties in aircraft and their subsequent hospitalization have stimulated efforts to minimize odors arising from draining wounds, old odorous casts, and gangrenous wounds. For aircraft, chemical sprays and vapors, perfumes, oxidizing gases and ventilation methods are unsatisfactory. An ideal deodorant would purify the air by means of odor adsorption so that subsequently the air can 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 question of superiority of adsorption methods for elimination of odors over other methods remains to be answered. The present status of the problem is that the commercial aspect is highly controversial.

REFERENCES

1—Confined Air as a Vehicle of Infection; Current Progress in Air Sanitation, by Stuart Mudd (*Medical Clinics of North America*, 28:1293, 1944).

2—The Seasonal Patterns of Measles and Chicken Pox, by Mildred W. Wells (*American Journal of Hygiene*, 40:279, 1944).

3—Aerobiology (Publication No. 17 of the *American Association for the Advancement of Science*, 1942).

4—Scarlet Fever as an Air-Borne Infection, by H. L. Hodes, F. F. Schwentker, B. M. Chenowith, Jr., and J. L. Peck (*American Journal of Medical Sciences*, 209:64, 1945).

5—Factors in the Control of the Spread of Acute Respiratory Infections with Reference to Streptococcal Illness and Acute Rheumatic Fever, by S. M. Wheeler and T. D. Jones (*American Journal of the Medical Sciences*, 209:58, 1945).

6—Laboratory and Field Studies of Glycols and Floor-Oiling in the Control of Air-Borne Bacteria, by A. P. Krueger *et al* (*U. S. Naval Medical Bulletin*, 42:1288, 1944).

7—Bacterial Content of Air in Army Barracks, by A. M. Lemon, H. Wise and M. Hamburger (*War Medicine*, 6:92, 1944).

8—A Study of the Nature and Control of Air-Borne Infection in Army Camps, by O. H. Robertson, M. Hamburger, C. G. Loosli, T. T. Puck, and H. M. Lemon (*Journal of the American Medical Association*, 126:993, 1944).

9—Lethal Effects of Triethylene Glycol Vapor on Air Borne Bacteria and Influenza Virus, by O. H. Robertson *et al* (*Science* 97:142, 1943).

10—The Present Status of Glycol Vapors in Air Sterilization, by M. Hamburger, Jr., O. H. Robertson, and T. T. Puck (*American Journal of the Medical Sciences*, 209:162, 1945).

11—Summary of a 3-Year Study of the Clinical Applications of the Disinfection of Air by Glycol Vapors, by T. N. Harris and J. Stokes, Jr. (*American Journal of the Medical Sciences*, 209:152, 1945).

12—Ultra-Violet Light Control of Air-Borne Infections in a Naval Training Center, by S. M. Wheeler, H. S. Ingraham, A. Hollaender, N. D. Lill, J. Gershon-Cohen, and E. W. Brown (*American Journal of Public Health*, 35:457, 1945).

13—A Comparative Study of the Effect on Men of Continuous Versus Intermittent Exposure to a Tropical Environment, by N. Pace, M. B. Fisher, J. E. Birren, G. C. Pitts, W. A. White, Jr., W. V. Consolazio, and L. J. Pecora (Research Project X-205, Report No. 2, *Naval Medical Research Institute*, Bethesda, Md., May, 1945).

14—Physiological Observations Made on Men Aboard Ship During a Shakedown Cruise in Tropical Waters, by N. Pace, W. V. Consolazio, and A. R. Behnke (Research Project X-205, Report No. 3, *Naval Medical Research Institute*, Bethesda, Md., August, 1945).

15—Environmental and Physiologic Studies Aboard an Air-Cooled Hospital Ship Enroute from Norfolk, Virginia to Canal Zone (U. S. S. TRANQUILITY (AH-14), 6-13 June 1945), by A. R. Behnke (Research Project X-205, Report No. 4, *Naval Medical Research Institute*, Bethesda, Md., September 1945).

16—Environmental and Physiologic Studies aboard an Air-Cooled Hospital Ship En Route from Norfolk, Virginia to Canal Zone (U. S. S. CONSOLATION (AH-15), 14-20 July 1945), by G. J. Duffner and M. Ross (Research Project X-205, Report No. 5, *Naval Medical Research Institute*, Bethesda, Md., September 1945).

17—Book Review by H. C. Bazett (*Science*, 102:74, 1945) of Climate and the Energy of Nations, by S. F. Markham (Oxford University Press, 1944).

18—Safeguarding the Operating Room Against Explosions, by Victor B. Phillips (*Modern Hospital*, 46, April and May, 1936).

See also Explosion Hazards of Combustible Anesthetics, by G. W. Jones, R. E. Kennedy and G. J. Thomas (*U. S. Bureau of Mines, Technical Paper* No. 653, 1943).

19—Fundamentals of Anesthesia (American Medical Association Press, Chicago, Ill., 2nd Edition, 1944, p. 204).

20—The Hazard of Fire and Explosion in Anesthesia, by B. A. Green (*Anesthesiology* 2:144, 1941).

21—Control of Physical Hazards of Anesthesia, by R. M. Tobell and A. W. Friend (*Canadian Medical Association Journal* 46:560, 1942).

22—A.S.H.V.E. RESEARCH REPORT No. 1111—Air Conditioning Requirements of an Operating Room and Recovery Ward, by F. C. Houghten and W. Leigh Cook, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 161).

23—Report on Air Conditioning in Surgery, by W. Leigh Cook, Jr. (*Department of Industrial Hygiene, School of Medicine, University of Pittsburgh*, 1940).

24—Unpublished Naval Studies by A. R. Behnke and O. Schneider (1940).

25—Disinfection of Air by Air Conditioning Processes, by C. P. Yaglou and Ursula Wilson (*American Association for the Advancement of Science*, Publication No. 17, p. 129).

26—The Control of Cross-Contamination by the Use of Mechanical Barriers, by J. A. Reyniers (*Aerobiology, American Association for the Advancement of Science, Symposium*, 17:254, 1942).

27—Report of the Committee on Air Conditioning (*The American Hospital Association*, p. 2, 1937).

28—The Premature Infant: A Study of the Effect of Atmospheric Conditions on Growth and on Development, by K. D. Blackfan, C. P. Yaglou and K. McKenzie (*American Journal Diseases of Children*, 46: 1175, 1933).

29—Observations on the Control of Respiratory Contagion in the Cradle, by I. Rosenstern (*Aerobiology, American Association for the Advancement of Science, Symposium* 17:242, 1942).

30—Fever Therapy by Physical Means, by Frank H. Krusen and E. C. Elkins (*Journal American Medical Association*, 112:1689, 1939).

31—Physical Medicine, by F. H. Krusen (W.B. Saunders Co., Philadelphia and London, 1941).

32—A.S.H.V.E. Research Report No. 654—Some Physiological Reactions of High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten (A.S.H.V.E. Transactions, Vol. 29, 1923, p. 129).

33—A.S.H.V.E. Research Report No. 1054—Fever Therapy Induced by Conditioned Air, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 131). A.S.H.V.E. Research Report No. 1161—Fever Therapy Locally Induced by Conditioned Air, by M. B. Ferderber, F. C. Houghten and Carl Gutberlet (A.S.H.V.E. Transactions, Vol. 46, 1940, p. 307).

34—Refrigeration for Anesthesia and Therapy, by L. W. Crossman and S. K. Safford (*The Modern Hospital*, 64:86, 1945).

35—Air Cleaning as an Aid in the Treatment of Hay Fever and Bronchial Asthma, by Leo H. Criep and M. A. Green (*Journal of Allergy*, 7:120, 1936).

36—The Effect of Low Relative Humidity and Constant Temperature on Pollen Asthma, by B. Z. Rappaport, T. Nelson and W. H. Welker (*Journal of Allergy*, 6:111, 1935).

37—Hospital Air Conditioning, by C. P. Yaglou (The Environment and Its Effect Upon Man, *Harvard School of Public Health*, p. 244, 1939).

38—Manual of Oxygen Therapy Techniques, by A. H. Andrews, Jr. (The Year Book Publishers, Inc., 1943).

39—Principles and Practices of Inhalational Therapy, by A. L. Barach (J. B. Lippincott Co., Philadelphia, London, Montreal, 1944).

40—The Management of Pneumonia, by J. G. M. Bullowa, (*Oxford University Press*, p. 260, 1937).

CHAPTER 14

Heating Load

General Procedure, Outside Temperatures, Inside Temperatures, Attic Temperatures, Temperatures in Unheated Spaces, Ground Temperatures, Basement Temperatures and Heat Loss, Transmission Heat Loss, Heat Loss Through Ceilings and Roofs, Infiltration Loss, Selection of Wind Velocities, Auxiliary Heat Sources, Intermittently Heated Buildings, Residence Heat Loss Problems

IN the design of 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 specified inside air temperature during periods of minimum selected design weather conditions. The heat losses may be divided into two groups, namely (1) the transmission losses or heat losses through the confining walls, floor, ceiling, glass or other surfaces and (2) the infiltration losses or heat losses due to air leakage through cracks and crevices, around doors and windows, opening of doors and other sources of interchange of air between the inside and outside.

GENERAL PROCEDURE

The general procedure for calculating heat losses of a structure is:

1. Select the outside design temperature. The data on climatic conditions given in Table 1 and the Design Temperature Zone Map, Fig. 1, will be useful but should be applied with judgment as suggested in the section Outside Design Temperatures.

2. Select the inside air temperature, at the 60-in. breathing line or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 2).

3. Estimate temperatures in adjacent unheated spaces and the attic. 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 6).

5. Measure amount of net 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 inside and outside 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 and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 8).

8. The sum of the heat losses by transmission (Item 6) through the outside wall 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.

OUTSIDE DESIGN TEMPERATURES

There are no hard and fast rules for selecting the outside design temperature to be used for a given locality or type of building or heating system, and the problem is to some extent a matter of judgment and experience. The outside design temperature is seldom taken as the lowest temperature, or even the lowest daily mean temperature ever recorded in a given locality. Such temperatures are rarely repeated in successive years. A temperature somewhat higher than the minimum or the lowest daily mean on record may properly be assumed in making the heat loss

TABLE 1. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]

Col. A	Col. B	Col. C	Col. D	Col. E	Col. F	Col. G
State	City	Normal Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Average Annual Minimum Temperature[b]	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
Alabama	Birmingham	53.8	−10	12	8.0	N
	Mobile	58.9	−1	22	10.2	N
Arizona	Flagstaff	35.8	−30	−15	7.8	SW
	Phoenix	59.5	16	26	5.3	E
Arkansas	Fort Smith	50.4	−15	6	8.1	E
	Little Rock	51.6	−12	10	8.3	NW
California	Los Angeles	58.5	28	37	6.3	NE
	San Francisco	54.2	27	37	7.5	N
Colorado	Denver	38.9	−29	−11	7.5	S
	Grand Junction	38.9	−21	−2	4.3	NW
Connecticut	New Haven	38.4	−15	−1	9.5	N
District of Col.	Washington	43.4	−15	−1	7.9	NW
Florida	Jacksonville	62.0	10	29	9.1	NE
Georgia	Atlanta	51.5	−8	12	11.4	NW
	Savannah	58.5	8	22	9.5	NW
Idaho	Lewiston	42.3	−23	1	5.3	E
	Pocatello	35.7	−28	−12	9.3	SE
Illinois	Chicago	36.4	−23	−8	12.5	W
	Springfield	39.8	−24	−7	12.1	NW
Indiana	Evansville	45.1	−16	1	9.8	S
	Indianapolis	40.3	−25	−6	11.5	S
Iowa	Dubuque	33.9	−32	−17	7.1	NW
	Sioux City	32.6	−35	−20	11.6	NW
Kansas	Concordia	39.8	−25	−13	7.7	N
	Dodge City	41.4	−26	−10	10.3	NW
Kentucky	Louisville	45.3	−20	−5	9.9	SW
Louisiana	New Orleans	61.6	7	26	8.7	NE
	Shreveport	56.2	−5	16	8.1	SE
Maine	Eastport	31.5	−23	−15	12.6	NW
	Portland	33.8	−21	−6	9.2	NW
Maryland	Baltimore	43.8	−7	8	8.1	SW
Massachusetts	Boston	38.1	−18	−3	11.2	W
Michigan	Alpena	29.6	−28	−12	11.0	W
	Detroit	35.8	−24	−11	12.7	SW
	Marquette	28.3	−27	−13	10.8	NW
Minnesota	Duluth	24.3	−41	−28	13.5	SW
	Minneapolis	29.4	−34	−23	11.3	NW
Mississippi	Vicksburg	56.8	−1	18	8.3	SE
Missouri	St. Joseph	39.5	−24	−12	9.3	W
	St. Louis	43.6	−22	−2	11.7	S
	Springfield	44.3	−29	−5	10.9	SE
Montana	Billings	34.3	−49	−30	11.9	SW
	Havre	27.6	−57	−36	9.5	SW
Nebraska	Lincoln	37.0	−29	−13	10.7	S
	North Platte	35.4	−35	−17	8.3	W
Nevada	Tonopah	39.7	−15	−2	10.0	SE
	Winnemucca	37.9	−36	−10	8.2	NE
New Hampshire	Concord	33.3	−35	−15	6.6	NW
New Jersey	Atlantic City	41.6	−9	6	15.6	NW
New York	Albany	35.2	−24	−11	8.4	S
	Buffalo	34.8	−20	−4	17.2	W
	New York	40.7	−14	−3	16.7	NW
New Mexico	Santa Fe	38.3	−13	0	7.8	NE
North Carolina	Raleigh	50.0	−2	13	7.9	SW
	Wilmington	54.3	5	18	8.1	W
North Dakota	Bismarck	24.6	−45	−31	9.0	NW
	Devils Lake	20.3	−46	−33	10.4	W
Ohio	Cleveland	37.2	−17	−2	15.0	SW
	Columbus	39.9	−20	−3	11.6	SW

TABLE 1. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS[a]—(Concluded)

Col. A	Col. B	Col. C	Col. D	Col. E	Col. F	Col. G
State or Province	City	Normal Temperature, Oct. 1-May 1	Lowest Temperature Ever Reported	Average Annual Minimum Temperature[b]	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
Oklahoma	Oklahoma City	47.9	−17	2	11.8	NW
Oregon	Baker	35.2	−25	−17	6.8	SE
	Portland	46.1	−2	18	7.4	S
Pennsylvania	Philadelphia	42.7	−11	6	11.0	NW
	Pittsburgh	41.0	−20	−2	11.7	W
Rhode Island	Providence	37.2	−17	1	12.6	NW
South Carolina	Charleston	57.4	7	22	10.6	SW
	Columbia	54.0	−2	19	8.1	NE
South Dakota	Huron	28.2	−43	−26	10.5	NW
	Rapid City	33.4	−34	−21	7.9	W
Tennessee	Knoxville	47.9	−16	2	7.1	SW
	Memphis	51.1	−9	9	9.4	S
Texas	El Paso	53.5	−5	16	9.0	NW
	Ft. Worth	55.2	−8	12	10.3	NW
	San Antonio	60.6	4	21	8.3	NE
Utah	Modena	36.3	−32	−15	9.3	W
	Salt Lake City	40.0	−20	2	6.7	SE
Vermont	Burlington	31.5	−29	−17	11.7	S
Virginia	Lynchburg	46.8	−7	8	8.1	NW
	Norfolk	49.3	2	15	12.4	N
	Richmond	47.0	−3	10	8.0	SW
Washington	Seattle	46.3	3	20	10.1	SE
	Spokane	37.7	−30	−5	6.3	SW
West Virginia	Elkins	39.4	−28	−8	5.8	W
	Parkersburg	42.6	−27	−1	7.3	SW
Wisconsin	Green Bay	30.0	−36	−18	10.6	SW
	LaCrosse	31.7	−43	−21	5.6	S
	Milwaukee	33.4	−25	−12	12.0	W
Wyoming	Lander	30.0	−40	−12	3.9	SW
	Sheridan	31.0	−41	−26	5.1	NW
Alta.	Edmonton	22.8	−57	−41.3	7.5	SW
B. C.	Vancouver	42.6	2	13.1	4.5	E
	Victoria	44.0	−2	19.4	12.6	N
Man.	Winnipeg	17.2	−54	−37.7	10.1	NW
N. B.	Fredericton	27.5	−35	−25.0	9.1	NW
N. S.	Yarmouth	34.8	−12	−0.1	14.3	NW
Ont.	London	32.6	−27	−13.7	10.3	W
	Ottawa	26.4	−35	−24.1	8.4	W
	Port Arthur	22.0	−40	−29.5	8.0	NW
	Toronto	32.6	−26	−11.2	13.6	SW
P. E. I.	Charlottetown	29.4	−27	−13.2	9.8	NW
Que.	Montreal	28.1	−29	−18.4	11.3	SW
	Quebec	24.5	−34	−23.2	13.3	SW
Sask.	Prince Albert	16.0	−70	−47.3	5.1	W
Yukon	Dawson	1.9	−68	−54.3	3.7	S
Newfoundland	St. Johns	31.4	−21	−5.3	12	SW

[a]United States data from U. S. Weather Bureau, and Canadian data from Meteorological Service of Canada, corrected to 1943.

[b]Obtained by averaging the lowest temperatures (one for each year) recorded for the period of years the local Weather Bureau has been in operation.

computations. Column D, Table 1 lists the lowest dry-bulb temperatures ever reported in the places listed. Column E, Table 1 shows the average of the lowest annual dry-bulb temperatures for the same localities. Temperatures for other cities may be obtained from local weather bureau records[1]. The design temperatures shown in Fig. 1 are generally representative of the practice in various sections of the United States, although

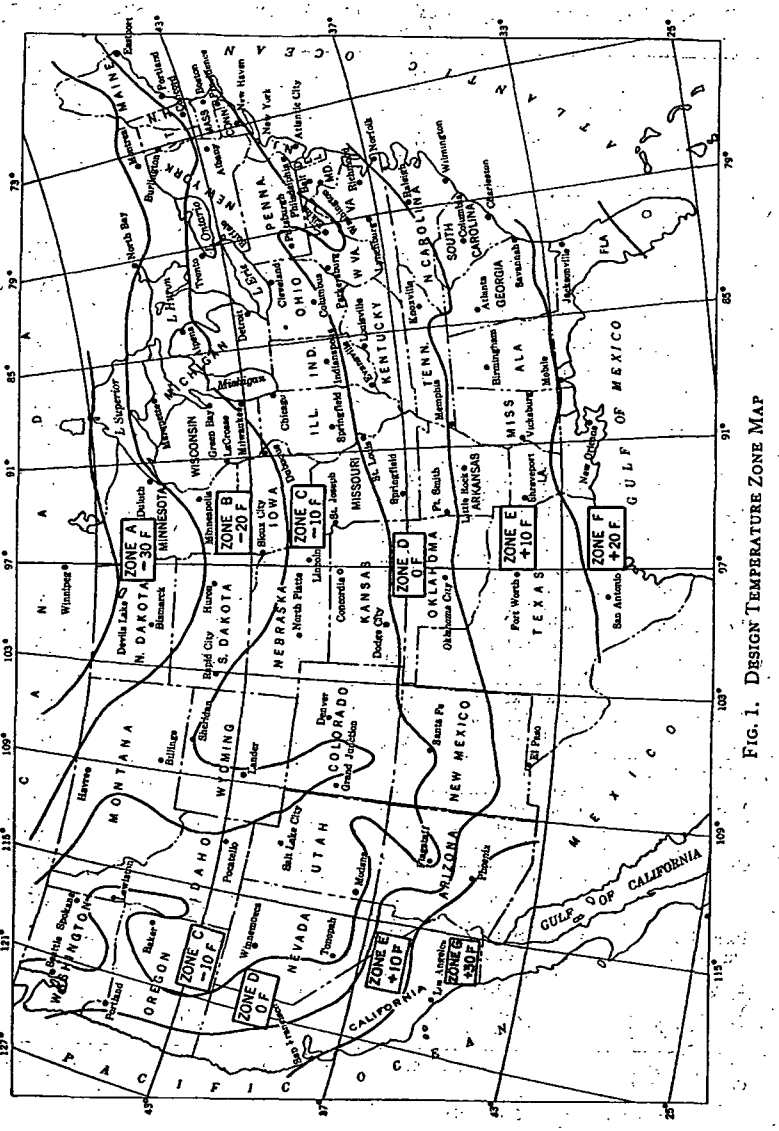


FIG. 1. DESIGN TEMPERATURE ZONE MAP

in some instances, due to local conditions of altitude or exposure, the design temperatures may vary somewhat from those indicated.

The A.S.H.V.E. Technical Advisory Committee on Weather Design Conditions has recommended the adoption for heating of an outside design temperature which is equalled or exceeded during 97½ per cent

of the hours in December, January, February, and March, but the work of compiling the various local temperatures is not yet completed.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 2 presents values which conform to good practice.

TABLE 2. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED[a]

TYPE OF BUILDING	DEG F	TYPE OF BUILDING	DEG F
SCHOOLS—		THEATERS—	
Class rooms	70–72	Seating space	68–72
Assembly rooms	68–72	Lounge rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and baths	70		
Wardrobe and locker rooms	65–68	HOTELS—	
Kitchens	66	Bedrooms and baths	70
Dining and lunch rooms	65–70	Dining rooms	70
Playrooms	60–65	Kitchens and laundries	66
Natatoriums	75	Ballrooms	65–68
		Toilets and service rooms	68
HOSPITALS—			
Private rooms	70–72	HOMES	70–72
Private rooms (surgical)	70–80	STORES	65–68
Operating rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	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 12.) When relative humidity is not controlled separately, optimum dry-bulb temperature for comfort will be slightly higher than shown in Table 2.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 12. 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 *effective temperature* for sedentary persons, as determined at the A.S.H.V.E. Research Laboratory, is 66 deg.

As explained in Chapter 12 for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures listed in Table 2, consideration should be given to whether provision is to be made for humidification and, if so, the actual relative humidity to be maintained.

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 2 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

TABLE 3. 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 per cent per foot of height above the breathing level (5 ft) up to 15 ft and 1/10 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 per cent to 100 per cent.

somewhat difficult to determine as it depends on many factors, including (1) the type of heating system, (2) ceiling height, and (3) the inside-outside 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 direct radiation, unit heaters or warm air is used, and in the latter case, whether the circulation is by gravity, auxiliary fan or forced air. Although with properly adjusted air flow the temperature differential with unit heaters can be reduced to a minimum, it is possible with improper adjustment that it may be increased over that which would normally result without mechanical circulation of the air if the air flow is not properly adjusted.

It would be difficult from present available information to establish rules for determining the temperature difference to use in all cases. However, for residences and other structures having ceiling heights under 10 ft, the comparatively small temperature differential between the

breathing level and ceiling may generally be neglected without serious error. For higher ceilings where specific test data are not available, an allowance of approximately 1 per cent per foot of height above the breathing level may be made for ceiling heights up to 15 ft and approximately 1/10 of 1 deg per foot of height above this level. The values in Table 3 are calculated on this basis. For direct radiation and gravity warm air systems, the allowance should be increased from 50 per cent to 100 per cent over those given in Table 3. These rules should, however, be used with considerable discretion.

Temperature at Floor Level: According to the University of Illinois Research Residence tests[2], the temperature at the floor level ranged from about 2½ 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[3] 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-floor differential may also be neglected as the two are somewhat compensating, especially where both floor and ceiling heat losses are calculated for the same space. In other cases, the 10 ft temperature differentials in Table 3 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.

$$t_a = \frac{A_c U_c t_i + t_o (A_r U_r + A_w U_w + A_g U_g)}{A_r U_r + A_w U_w + A_g U_g + A_c U_c} \qquad (1)$$

where

t_a = attic temperature, Fahrenheit degrees.
t_i = inside temperature near top floor ceiling, Fahrenheit degrees.
t_o = outside 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 6). 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 6).
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: $t_i = 70$; $t_o = 10$; $A_c = 1000$; $A_r = 1200$; $A_w = 100$; $A_g = 10$; $U_r = 0.50$; $U_c = 0.40$; $U_w = 0.30$; $U_g = 1.13$.

Solution: Substituting these values in Equation 1:

$$t_a = \frac{(1000 \times 0.40 \times 70) + 10\,[(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)]}{(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13) + (1000 \times 0.40)}$$

$$t_a = \frac{34{,}413}{1041} = 33.1 \text{ F.}$$

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. However, according to tests [4], such venting of attics by means of louvers or other small openings does not appreciably reduce the attic temperature and may be neglected without serious error. 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 outside temperature.

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 generally range between the inside and outside temperatures, depending on the relative areas of the surfaces adjacent to the heated room and exposed to the outside. If the respective surface areas adjacent to the heated room and exposed to the outside 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 inside and outside 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.

$$t_u = \frac{t(A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.}) + t_o\,(A_aU_a + A_bU_b + A_cU_c + \text{etc.})}{A_1U_1 + A_2U_2 + A_3U_3 + \text{etc.} + A_aU_a + A_bU_b + A_cU_c + \text{etc.}} \quad (2)$$

where

t_u = temperature in unheated space, Fahrenheit degrees.

t = inside design temperature of heated room, Fahrenheit degrees.

t_o = outside design temperature, Fahrenheit degrees.

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 outside, 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 outside (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 $t = 70$ and $t_o = -10$.

Solution. Substituting in Equation 2:

$$t_u = \frac{70[(100\times0.15)+(120\times0.20)+(140\times0.25)]+-10[(100\times0.10)+(140\times0.30)]}{(100\times0.15)+(120\times0.20)+(140\times0.25)+(100\times0.10)+(140\times0.30)}$$

$$t_u = \frac{4660}{126} = 37 \text{ F.}$$

The temperature in unheated spaces having large glass areas and with two or more surfaces exposed to the outside (such as sleeping porches and sun parlors), are generally assumed to be the same as the outside temperature.

GROUND TEMPERATURES

Ground temperatures to be assumed for estimating basement heat losses will usually differ in the case of basement walls and floors, the temperatures under the floors being generally higher than those adjacent to walls.

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 the 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. Consequently ground temperatures near the surface may be higher in cold climates where the snow remains on the ground for a greater length of time than in more moderate climates where the snow melts away periodically during the winter.

Complete data for various localities are not as yet available but in estimating heat losses through vertical walls below grade, it is advisable not to assume average ground temperatures above 32 F in northern climates when estimating heat losses from heated basements. This is for the mean height of the basement wall. Since the recommended wall coefficient for basement walls in contact with the soil is only 0.10, any small variation in the assumed ground temperature will not materially affect the calculated heat loss.

Temperatures Under Basement Floors

The temperature under basement floors [5] is influenced by the heat from the basement or protected from the influence of atmospheric conditions by the basement. In computing losses through basement floors the ground temperatures may be assumed the same as the approximate water temperature at depths of 30 to 60 ft given in Fig. 3, Chapter 37.

Test results indicate that the heat losses through basement floors are frequently over estimated [6].

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 a basement is completely below grade and is *not heated*, the temperature in the basement will normally range between that in the rooms above and the ground temperature. Basement windows will of course lower the basement temperature when it is colder outside and any heat given off by the heating plant will increase the basement temperature. In any case, the exact basement temperature is likely to be a somewhat indeterminate quantity, if the basement is not heated. Since the basement temperature will generally be lower than that of the rooms above, an allowance should theoretically be made for the loss from the rooms above through the floor over the basement.

If the basement is *heated* and a specified temperature is to be maintained, the heat loss should be estimated in the usual manner, based on the proper wall and floor coefficients (see Chapter 6) and the outside air and the ground temperatures. Heat loss through windows and walls

above grade should be based on outside 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.

TRANSMISSION HEAT LOSS

The basic formula for the loss of heat by transmission through any surface is given in Equation 3.

$$H_t = A\,U\,(t - t_o) \qquad (3)$$

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 surfaces, square feet.
U = coefficient of transmission, air to air, Btu per (hour) (square foot) (Fahrenheit degree temperature difference) (Chapter 6).
t = inside temperature near surface involved which may not necessarily be the so-called breathing line temperature, Fahrenheit degrees.
t_o = outside temperature, or temperature of adjacent unheated space or of the ground, Fahrenheit degrees.

Example 3. Calculate the transmission loss through an 8 in. brick wall having an area of 150 sq ft if the inside temperature (t) is 70 F and the outside temperature (t_o) is −10 F.

Solution. The coefficient of transmission (U) of a plain 8 in. brick wall is 0.50 (Chapter 6, Table 7). The area (A) is 150 sq ft. Substituting in Equation 3:

$$H_t = 150 \times 0.50 \times [70 - (-10)] = 6000 \text{ Btu per hour.}$$

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 3 the ceiling area (A), the inside-outside temperature difference ($t - t_o$) and the proper value of (U):

   a. *Flat roofs.* Select the coefficient of transmission of the ceiling and roof from Tables 14 or 15, Chapter 6, 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 17, Chapter 6 or calculate the combined roof and ceiling coefficient by means of Equation 5, Chapter 6, where this formula is applicable as explained in Chapter 6.

2. By estimating the attic temperature (based on the inside and outside design temperature) by means of Equation 1, and substituting for t_o in Equation 3, the value of t_a 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 per paragraph 1a.

INFILTRATION HEAT LOSS

The infiltration heat loss includes (1) the *sensible heat loss* or the heat required to warm the outside 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 outside air which enters a room by infiltration to the temperature of the room, is given in Equation 4.

$$H_s = 0.24\,Qd\,(t - t_o) \qquad (4)$$

where

H_s = heat required to raise temperature of air leaking into building from t_o to t, Btu per hour.
0.24 = specific heat of air.
Q = volume of outside air entering building, cubic feet per hour (see Chapter 8).
d = density of air at temperature t_o, pounds per cubic foot.

It is sufficiently accurate to use $d = 0.075$ in which case Equation 4 reduces to

$$H_s = 0.018\,Q\,(t - t_o) \qquad (4a)$$

The volume of outside air entering per hour (Q) depends on the wind velocity and direction, the width of crack or size of openings, the type of openings and other factors, as explained in Chapter 8. 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:

$$H_s = 0.018\,Q\,L\,(t - t_o) = B\,L\,(t - t_o) \qquad (4b)$$

where

B = air leakage per (hour) (foot of crack) (Chapter 8) 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 4. What is the infiltration heat loss per hour through the crack of a 3 x 5 ft average, double-hung, non-weatherstripped, wood window, based on a wind velocity of 15 mph? Assume inside and outside temperatures to be 70 F and zero respectively.

Solution. According to Table 2, Chapter 8, 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 foot of crack per hour. Therefore, $B = 39 \times 0.018 = 0.70$. The length of crack ($L$) is $(2 \times 5) + (3 \times 3)$, or 19 ft; $t = 70$ and $t_o = 0$. Substituting in Equation 4b,

$$H_s = 0.70 \times 19 \times (70 - 0) = 931 \text{ Btu per hour.}$$

Crack Length to be Used for Computations

The amount of crack used for computing the infiltration heat loss should not be less than half of the total crack in the outside walls of the room. For a building having no partitions, whatever wind enters 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.

The total infiltration loss of a building having partitions will not be equal to the sum of the infiltration losses of the various rooms, since at any given time infiltration will take place only on the windward side or sides and not on the leeward side. Therefore, if a building has more than one room which is divided by interior walls or partitions, it is sufficiently accurate to use half of the total infiltration losses for determining the total heat requirements.

Latent Heat Loss

When it is intended to add moisture to air leaking into a room for the maintenance of proper winter comfort conditions, it is necessary to

determine the heat equivalent to evaporate the required amount of water vapor, which may be calculated by the equation:

$$H_l = Q\,d\left(\frac{m_i - m_o}{7000}\right) h_{fg} \qquad (5)$$

where

H_l = heat required to increase moisture content of air leaking into building from m_o to m_i, Btu per hour.
Q = volume of outside air entering building, cubic feet per hour.
d = density of air at temperature t_i, pounds per cubic foot.
m_i = vapor density of inside air, grains per pound of dry air.
m_o = vapor density of outside air, grains per pound of dry air.
h_{fg} = latent heat of vapor at m_i, Btu per pound.

If the latent heat of vapor (h_{fg}) is assumed to be 1060 Btu per pound, Equation 5 reduces to

$$H_l = 0.0114\,Q\,(m_i - m_o) \qquad (5a)$$

Equations 4a, 4b and 5a may also be used for determining the sensible and latent heat gains due to infiltration in cooling load computations.

SELECTION OF WIND VELOCITIES

The effect of wind on the heating requirements of any building should be given consideration under two heads:

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 movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

Theoretically as a basis for design, the most unfavorable combination of temperature and wind velocity should be chosen. It is entirely possible that a building might require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. However, the combination of wind and temperature which is the worst would differ with different buildings, because wind velocity has a greater effect on buildings which have relatively high infiltration losses. It would be possible to work out the heating load for a building for several different combinations of temperature and wind velocity 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.

It has been the practice for many years in estimating air leakage by the crack method to use the *average* wind velocity during the months of December, January and February. This *average* wind velocity may not necessarily correspond with that occurring during periods when the outside design temperature prevails, the latter being not an *average* but rather a near extreme, that is, a specified number of degrees above the lowest temperature recorded in the locality involved. Therefore instead of using the aforementioned *average* wind velocity, it is the practice of some designers to use in all cases a wind velocity of 15 mph together with the proper design temperature. Although a 15 mph wind velocity is higher than the general average wind velocity during December, January and February in various United States cities, this and higher wind velocities frequently occur during periods of outside temperature corresponding to the design temperature. It should be added that this wind velocity

also corresponds with that on which the heat loss coefficients in Chapter 6 are based, although the effect of variations in wind velocity on the infiltration losses is generally much greater than the effect of wind velocity on the heat loss by transmission through walls. Therefore, pending further investigation of this subject, either the average during December, January and February or a 15 mph wind velocity may be used at the discretion of the designer. Where the air change method is used for estimating infiltration losses, the wind velocity is not considered.

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, according to a survey made in 1943, many GUIDE users have found that the use of exposure factors is not necessary as the GUIDE method of calculating heat losses provides an ample heat loss allowance. 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 multistory buildings. Although the exposure allowance is frequently assumed to be 15 per cent, 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.

As stated previously, the value of U in the tables of Chapter 6 is based on a wind velocity of 15 mph and the surface resistance for this wind velocity (0.17) is sufficiently low that higher wind velocities will decrease the surface resistance to a negligible degree and therefore have only a slight effect on the average over-all coefficient. On the other hand, infiltration losses vary almost directly as the wind velocity, as will be apparent from the factors in Table 2 of Chapter 8. The more exact method therefore would be to differentiate among the various exposures more accurately by calculating the infiltration and transmission losses separately for the different sides of the building, using different assumed wind velocities for the infiltration losses on the various sides of the building.

AUXILIARY HEAT SOURCES

The heat supplied by persons, lights, motors and machinery should always 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 be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. 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, which is

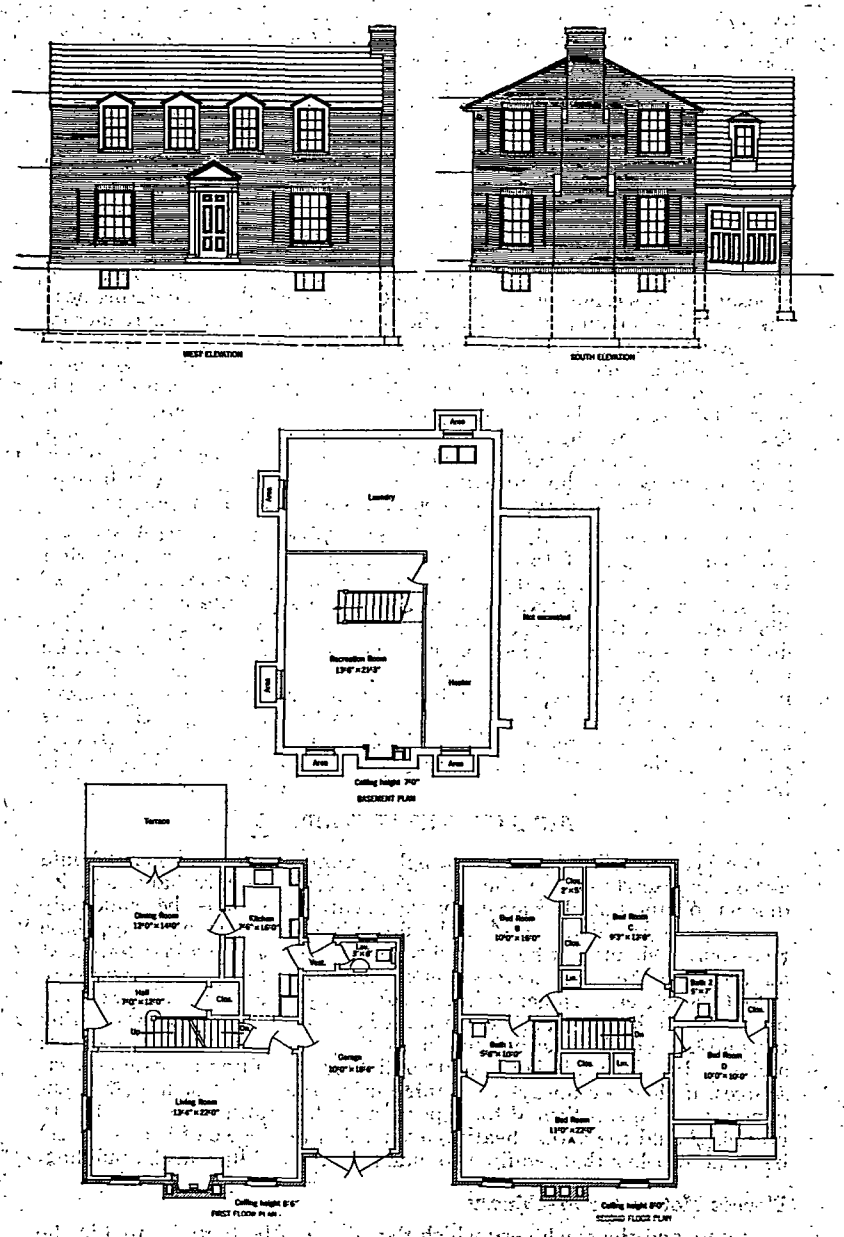


FIG. 2. ELEVATIONS AND FLOOR PLANS OF RESIDENCE

retained in the room if the product being 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 the first case the Btu supplied per hour $= \frac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2546$, and in the second case Btu per hour = bhp $\times$ 2546, in which 2546 is the Btu equivalent of 1 hp-hr. 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.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.413. One cubic foot of producer gas gives off about 150 Btu; one cubic foot of manufactured gas about 535 Btu; and one cubic foot of natural gas about 1000 Btu. A Welsbach burner averages 3 cu ft of gas per hour and a fish-tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, refer to data given in Chapter 12.

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 inside 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[7].

This additional heat may be figured 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 per cent 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 is usually made except in the size of boilers or furnaces. For churches, auditoriums and other intermittently heated buildings, additional capacity should be provided.

RESIDENCE HEAT LOSS PROBLEMS

Example 5. Calculate the heat loss of residence shown in Fig. 2 located in the vicinity of Chicago. Assume inside and outside design temperatures to be 70 F and −10 F respectively. The attic is unheated. Assume ground temperature to be 50 F under basement and garage floors and 32 F adjoining basement walls. Estimate infiltration by crack method, assuming average wind velocity to be 12.5 mph during December, January and February. No wall, ceiling or roof insulation is to be figured in this problem, but all first and second floor windows are to have storm sash. The building is constructed as follows (transmission coefficients (U) in parentheses):

Walls: Brick veneer, building paper, wood sheathing, studding, metal lath and plaster (0.28). 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 (0.53).

Ceiling (Second floor): Metal lath and plaster (0.69).

Windows: Double-hung wood windows with storm sash (0.45). Steel casement sash in basement (1.13).

Floor (Bedroom D): Maple finish flooring on yellow pine sub-flooring; metal lath and plaster ceiling below (0.25).

TABLE 4. HEAT LOSS CALCULATION SHEET FOR UNINSULATED RESIDENCE (FIG. 2)

A	B	C	D	E	F	G
Room or Space	Part of Structure	Net Area or Crack Length	Coefficient	Temp. Diff.[a]	Heat Loss (Btu per hour)	Totals (Btu per hour)
Bedroom A	Walls	238 sq ft	0.28	80	5330	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[b]	0.35[c]	80	1010	
	Ceiling[d]	242 sq ft	0.69	39.8	6660	14,440
Bedroom B and Closet	Walls	156 sq ft	0.28	80	3490	
	Glass	40 sq ft	0.45	80	1440	
	Infiltration	36 lin ft[e]	0.35	80	1010	
	Ceiling[d]	160 sq ft	0.69	39.8	4400	10,340
Bedroom C	Walls	114 sq ft	0.28	80	2560	
	Glass	27 sq ft	0.45	80	970	
	Infiltration	18 lin ft[f]	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	7,330
Bedroom D and Closet	Walls	118 sq ft	0.28	80	2650	
	Glass	20 sq ft	0.45	80	720	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	120 sq ft	0.69	39.8	3300	
	Floor over Garage	110 sq ft	0.25	35[g]	960[m]	8,130
Bathroom 1	Walls	30 sq ft	0.28	80	670	
	Glass	14 sq ft	0.45	80	500	
	Infiltration	18 lin ft	0.35	80	500	
	Ceiling[d]	55 sq ft	0.69	39.8	1510	3,180
Bathroom 2	Walls	79 sq ft	0.26	80	1770	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	15 lin ft	0.35	80	420	
	Ceiling[d]	35 sq ft	0.69	39.8	960	
	Floor over Garage	35 sq ft	0.25	35	310[m]	3,780
Living Room	Walls	267 sq ft	0.28	80	5980	
	Walls (adjoining garage)	94 sq ft	0.39[h]	35	1280[m]	
	Glass	50 sq ft	0.45	80	1800	
	Infiltration	40 lin ft	0.35	80	1120	10,180
Dining Room	Walls	166 sq ft	0.28	80	3720	
	Glass (doors)	35 sq ft	1.13	80	3160	
	Glass (window)	20 sq ft	0.45	80	720	
	Infiltration[i]	31 lin ft	0.35	80	870	8,470
Kitchen and Entrance to Garage	Walls (outside)	96 sq ft	0.28	80	2150	
	Walls (adjoining garage)	51 sq ft	0.39[h]	35	700[m]	
	Infiltration	27 lin ft	0.35	80	760	
	Glass	18 sq ft	0.45	80	650	
	Door to garage	17 sq ft	0.51	35	300[m]	4,560
Lavette and Vestibule	Walls (outside)	82 sq ft	0.28	80	1840	
	Walls (adjoining garage)	85 sq ft	0.39[h]	35	1160[m]	
	Door	19 sq ft	0.51	80	780	
	Glass	9 sq ft	0.45	80	320	
	Infiltration	19 lin ft	0.35	80	530	4,630
Entrance Hall	Walls	39 sq ft	0.28	80	870	
	Door	21 sq ft	0.38	80	640	
	Infiltration	20 lin ft	0.35	80	560	
	Ceiling[d, p]	87 sq ft	0.69	39.8	2490	4,560
Garage	Walls	167 sq ft	0.28	45	2110	
	Glass	53 sq ft	1.13	45	2700	
	Doors	44 sq ft	0.51	45	1010	
	Infiltration	37 lin ft	1.62[j]	45	2700	
	Floor (heat gain)	185 sq ft	0.10[k]	−15[k]	−280	
	Heat gain				−4710[m]	3,530
Recreation Room[n]	Floor	287 sq ft	0.10	20	570	
	Walls	220 sq ft	0.10	38	840	
	Glass	8 sq ft	1.13	80	720	
	Infiltration	8 lin ft	0.76	80	490	2,620
					Total	85,750

NOTES FOR TABLE 4.

[a] The inside-outside temperature difference is 70 − (−10) or 80 F, except where otherwise noted.

[b] Only the south windows are used for arriving at the window crack for this room, on the assumption that whatever air enters through the south window cracks will leave through the west windows or elsewhere.

[c] Double-hung wood windows with storm sash are assumed to have the same leakage per foot of crack as weatherstripped windows. The air leakage per foot of crack is about 19.5 cu ft per foot of crack for a wind velocity of 12.5 mph. (See Table 2, Chapter 8.) The heat equivalent of the air leakage per (hour) (degree temperature difference per foot of crack) is obtained by multiplying this value by 0.018, or 19.5 × 0.018 = 0.35.

[d] In this problem 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 the winter months. The attic temperature is estimated from Equation 4 to be 30.2 F when the outside temperature is −10 F and the room temperature is 70 F. The temperature difference is therefore 70 − 30.2 or 39.8 F.

[e] The window crack in the west wall having two windows is used.

[f] One-half the total crack is used in these rooms.

[g] Temperature in garage assumed to be 35 F.

[h] Coefficient for wall adjoining garage calculated on basis of metal lath and plaster on both sides of studs. (U = 0.39.)

[i] The door crack is used for estimating the infiltration in this room and as the French doors are weatherstripped the infiltration coefficient is assumed to be the same as in Note b.

[j] The leakage for the garage doors is assumed to be twice that for poorly-fitted double-hung wood windows or about 90 cu ft per foot of crack for a wind velocity of 12.5 mph. The infiltration coefficient is therefore 0.018 × 90 or 1.62.

[k] The ground temperature is assumed to be 50 F and, as the garage temperature is 35 F, the heat transfer will be from the ground to the garage, and this heat gain should therefore be subtracted from the heat loss.

[m] The heat losses from various rooms into the garage are heat gains for the garage.

[n] Heat is to be provided for the recreation room and this space is therefore figured on the basis of a 70 F temperature. Heat loss into the basement from recreation room is neglected, the calculations being based only on losses through the outside walls, glass and floor.

[p] The upstairs hall ceiling is included with the downstairs entrance hall because these are connected by means of the stairway. The heat should be provided downstairs.

Example 5 (continued)

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 4, and a summary of the results in Table 5. The values in column F of Table 4 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 4 for further explanation of data.

Attention is called to the summary of heat losses (Table 5) of the uninsulated residence (Fig. 2). As storm windows are used in this instance the *glass* and *door* trans-

TABLE 5. SUMMARY OF HEAT LOSSES OF UNINSULATED RESIDENCE

Heat losses given in Btu per hour

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	5330	6660	.....	1440	1010	14,440
Bedroom B	3490	4400	.....	1440	1010	10,340
Bedroom C	2560	3300	.....	970	500	7,330
Bedroom D	2650	3300	960	720	500	8,130
Bathroom 1	670	1510	.....	500	500	3,180
Bathroom 2	1770	960	310	320	420	3,780
Living Room	7260	.....	.....	1800	1120	10,180
Dining Room	3720	.....	.....	3880	870	8,470
Kitchen	2850	.....	.....	950	760	4,560
Lavette	3000	.....	.....	1100	530	4,630
Entrance Hall	870	2490	.....	640	560	4,560
Garage	−1030[a]	.....	−1550[b]	3410	2700	3,530
Recreation	840	.....	570	720	490	2,620
Totals	33,980	22,620	290	17,890	10,970	85,750
Percentages	39.6	26.4	0.3	20.9	12.8	100.0

[a] Wall heat loss of 2110 Btu minus wall heat gain of 3140 Btu.

[b] Heat gains; 960, 310 and 280 Btu.

mission heat losses of 20.9 per cent are relatively small. The infiltration losses (12.8 per cent) are also comparatively small in this case because the storm windows serve substantially the same purpose as weatherstripping. In this problem, the wall, ceiling and floor transmission losses comprise 66.3 per cent of the total.

Example 6. Calculate the heat loss of residence shown in Fig. 2 based on the same conditions as in *Example 5* 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 5*).

Floor (*Bedroom D*), 0.18.

Solution: The procedure for calculating the heat losses is similar to that for *Example 5*. A summary of the results is given in Table 6.

TABLE 6. SUMMARY OF HEAT LOSSES OF INSULATED RESIDENCE

Heat losses given in Btu per hour

Room or Space	Walls	Ceiling and Roof	Floor	Glass and Door	Infiltration	Totals
Bedroom A	2670	2370	.....	1440	1010	7,490
Bedroom B	1750	1570	.....	1440	1010	5,770
Bedroom C	1280	1170	.....	970	500	3,920
Bedroom D	1320	1170	690	720	500	4,400
Bathroom 1	340	540	.....	500	500	1,880
Bathroom 2	820	340	220	320	420	2,120
Living Room	3580	.....	.....	1800	1120	6,500
Dining Room	1860	.....	.....	3880	870	6,610
Kitchen	1400	.....	.....	950	760	3,110
Lavette	1460	.....	.....	1100	530	3,090
Entrance Hall	440	850	.....	640	560	2,490
Garage	−400[a]	.....	−1190[b]	3410	2700	4,520
Recreation	840	.....	570	720	490	2,620
Totals	17,360	8,010	290	17,890	10,970	54,520
Percentages	31.9	14.7	0.5	32.8	20.1	100.0

[a]Wall loss of 1050 Btu minus gains of 590, 320 and 540 Btu. [b]Heat gains; 690, 220 and 280 Btu.

REFERENCES

1—An Analysis of Winter Temperatures for One Hundred and Twenty Cities, by Clark M. Humphreys (*Carnegie Institute of Technology Bulletin*).

2—Investigation of Oil-Fired Forced Air Furnace Systems in the Research Residence, by A. P. Kratz and S. Konzo (*University of Illinois Engineering Experiment Station Bulletin* No. 318).

3—A.S.H.V.E. RESEARCH REPORT No. 1011—Tests of Three Heating Systems in an Industrial Type of Building, by G. L. Larson, D. W. Nelson, and John James (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

4—Methods of Moisture Control and Their Application to Building Construction, by F. B. Rowley, A. B. Algren and C. E. Lund. *University of Minnesota, Engineering Experiment Station Bulletin*, No. 17).

5—A.S.H.V.E. RESEARCH REPORT No. 1213—Heat Loss Through Basement Walls and Floors, by F. C. Houghten, S. I. Taimuty, Carl Gutberlet and C. J. Brown (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 369).

6—Measurements of Heat Losses from Slab Floor, by R. S. Dill, W. C. Robinson and H. E. Robinson (*U. S. Department of Commerce, National Bureau of Standards*, Building Materials and Structures Report BMS 103).

7—Heat Requirement Tables for Intermittently Heated Buildings, (*Engineering Experiment Station Bulletin*, No. 60, *A. and M. College of Texas*, College Station, Texas), contains a set of tables applicable to either intermittent heating or cooling. Further information may be found in a paper, A Method of Compiling Tables for Intermittent Heating, by Elmer G. Smith (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, June, 1942, p. 386).

CHAPTER 15

Cooling Load

Design Outside Temperatures, Components of Heat Gain, Normal Heat Transmission, Solar Heat Transmission, Solar Radiation Through Glass, Heat Introduced by Outside Air, Heat Emission of Appliances, Moisture Through Walls

LOAD calculations for summer air conditioning are more complicated than heating load calculations. Due to the variable nature of some of the contributing load components and the fact that they do not necessarily impose their maximum effect simultaneously, considerable care must be used in determining their phase relationship.

The conditions to be maintained in an enclosure depend upon several factors, especially the outside design conditions, duration of occupancy, and relationship between air motion, dry-bulb and wet-bulb temperatures. Information concerning the proper indoor effective temperature to be maintained is given in Chapter 12, for different geographical locations and for various age groups of individuals. Typical commercial design room conditions for the summer average peak load are shown in Table 1.

Summer dry-bulb and wet-bulb temperatures of various cities are given in Table 2. The temperatures are not the maximums but the design temperatures which should be used in air conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 to 4 per cent of the time, and because they are of such short duration it is not practicable to design a cooling system for them. The temperatures shown in Table 2 are based on available design conditions known to be applied successfully.

COMPONENTS OF HEAT GAIN

A cooling load determination is composed of five components which are classified in the following manner:

1. Normal heat transfer through windows, walls, partitions, doors, floors, ceilings, etc.
2. Transfer of solar radiation through windows, walls, doors, skylights, and roof.
3. Heat emission of occupants within enclosures.
4. Heat introduced by infiltration of outside air and controlled ventilation.
5. Heat emission of mechanical, chemical, gas, steam, hot water, and electrical appliances located within enclosures.

The components of heat gain, classified by source, are further classified as sensible and latent heat gain.

The first two components fall into the classification of sensible heat gain, that is, they tend to raise the temperature of the air within the structure. The last three components not only produce sensible heat gain but they may also tend to increase the moisture content of the air within the structure.

Normal Heat Transmission

By normal heat transmission, as distinguished from solar heat transmission, is meant the transmission of heat through windows, walls, partitions, etc. from without to interior of enclosure by virtue of difference between outside and inside air temperatures. Since the daily range of

TABLE 1. TYPICAL COMMERCIAL DESIGN ROOM CONDITIONS FOR SUMMER AVERAGE PEAK LOAD[a]

TYPE OF INSTALLATION	DRY-BULB TEMP	WET-BULB TEMP	RELATIVE HUMIDITY PER CENT	GRAINS PER LB	EFFECTIVE TEMP
Deluxe Application	78	65.25	50	72.7	72.2
Normal Application	80	67	51	78.5	74.0
15 to 40 min Occupancy	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 76 F and 50 per cent relative humidity at other than peak load.

outside air temperature is usually in excess of 20 F, the *effective outside temperature* for normal heat transmission through walls and floors having substantial heat capacity is usually taken 5 to 10 deg below the design maximum outside temperature. For windows, however, the instantaneous value of outside temperature is used. This load is calculated in a manner similar to that described in Chapter 14 (except that flow of heat is reversed) by means of the formula:

$$H_t = A\,U\,(t_e - t) \qquad (1)$$

where

H_t = heat transmitted through the material of wall, glass, floor, etc., Btu per hour.
A = net inside area of wall, glass, floor, etc., square feet.
t = inside temperature, Fahrenheit degrees.
t_e = effective outside temperature, Fahrenheit degrees.
U = coefficient of transmission of wall, glass, floor, etc. from Tables 4 to 18, Chapter 6, Btu per (hour) (square foot) (Fahrenheit degree difference in temperature).

Solar Heat Transmission

Calculations of the solar heat transmitted through walls and roofs are difficult because of periodic character of heat flow and time lag due to heat capacity of construction.

An illustration of the variation in radiation intensity on differently oriented surfaces is given in Fig. 1. The greater part of the radiation intensity is always direct radiation from the sun. However, during the time when the sun is shining, any surface receives radiation of a lower

TABLE 2. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Ala.	Birmingham	95	78	5.2	S
	Mobile	95	80	8.6	SW
Ariz.	Phoenix	105	76	6.0	W
Ark.	Little Rock	95	78	7.0	NE
Calif.	Los Angeles	90	70	6.0	SW
	San Francisco	90	65	11.0	SW
Colo.	Denver	95	64	6.8	S
Conn.	New Haven	95	75	7.3	S
Del.	Wilmington	95	78	9.7	SW
D. C.	Washington	95	78	6.2	S
Fla.	Jacksonville	95	78	8.7	SW
	Tampa	94	79	7.0	E

TABLE 2. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Concluded)

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Ga.	Atlanta	95	76	7.3	NW
	Savannah	95	78	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	95	76	8.2	S
Ind.	Indianapolis	95	76	9.0	SW
Iowa	Des Moines	95	77	6.6	SW
Kansas	Wichita	100	75	11.0	S
Ky.	Louisville	95	76	8.0	SW
La.	New Orleans	95	79	7.0	SW
	Shreveport	100	78	6.2	S
Maine	Portland	90	73	7.3	S
Md.	Baltimore	95	78	6.9	SW
Mass.	Boston	92	75	9.2	SW
Mich.	Detroit	95	75	10.3	SW
Minn.	Minneapolis	95	75	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	100	76	9.5	S
	St. Louis	95	78	9.4	SW
Mont.	Helena	95	67	7.3	SW
Nebr.	Lincoln	95	75	9.3	S
Nev.	Reno	95	65	7.4	W
N. H.	Manchester	90	73	5.6	NW
N. J.	Trenton	95	78	10.0	SW
N. Y.	Albany	92	75	7.1	S
	Buffalo	93	75	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	90	65	6.5	SE
N. C.	Asheville	90	75	5.6	SE
	Wilmington	95	79	7.8	SW
N. Dak.	Bismarck	95	73	8.8	NW
Ohio	Cincinnati	95	78	6.6	SW
	Cleveland	95	75	9.9	S
Okla.	Oklahoma City	101	76	10.1	S
Ore.	Portland	90	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	95	75	9.0	NW
R. I.	Providence	93	75	10.0	NW
S. C.	Charleston	95	80	9.9	SW
	Greenville	95	76	6.8	NE
S. Dak.	Sioux Falls	95	75	7.6	S
Tenn.	Chattanooga	95	77	6.5	SW
	Memphis	95	78	7.5	SW
Texas	Dallas	100	78	9.4	S
	El Paso	100	69	6.9	E
	Galveston	95	80	9.7	S
	Houston	95	78	7.7	S
	San Antonio	100	78	7.4	SE
Utah	Salt Lake City	92	63	8.2	SE
Vt.	Burlington	90	73	8.9	S
Va.	Norfolk	95	78	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	85	65	7.9	S
	Spokane	90	65	6.5	SW
W. Va.	Parkersburg	95	75	5.3	SE
Wis.	Madison	95	75	8.1	SW
	Milwaukee	95	75	10.4	S
Wyo.	Cheyenne	95	65	9.2	S

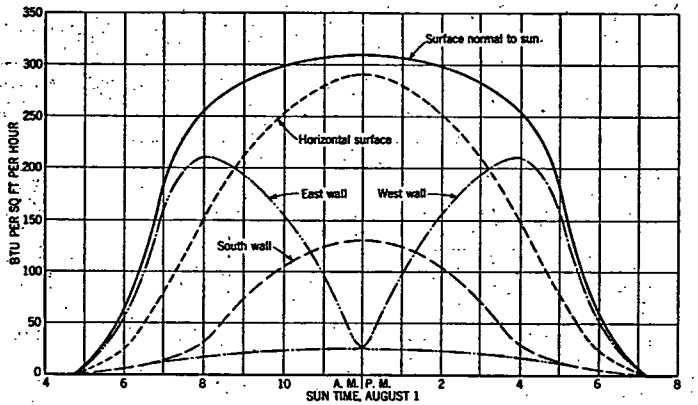


FIG. 1. SOLAR INTENSITY NORMAL TO SUN ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1 AT 40 DEG NORTH LATITUDE

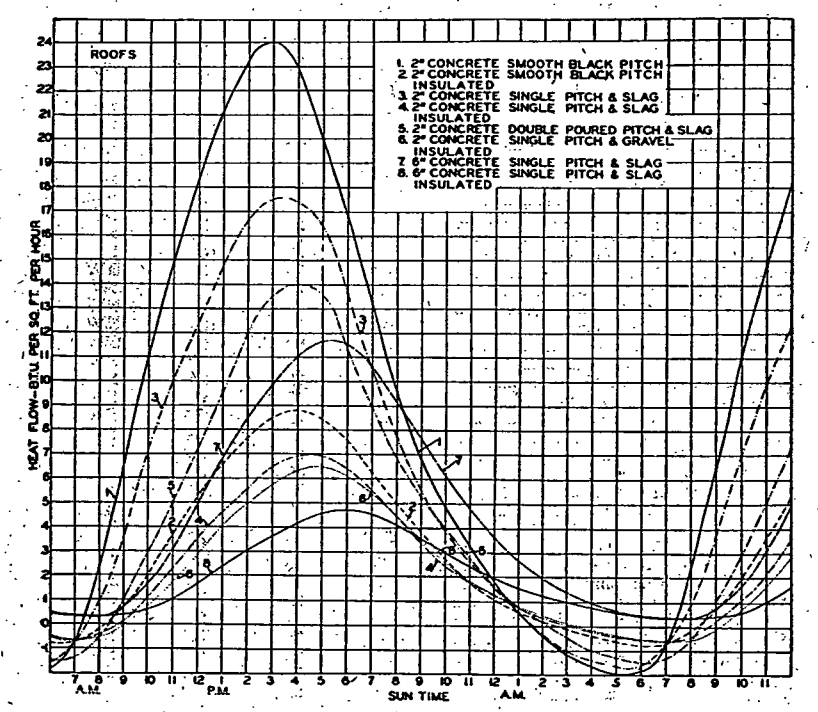


FIG. 2. HEAT FLOW-TIME RELATIONSHIP FOR HORIZONTAL ROOFS

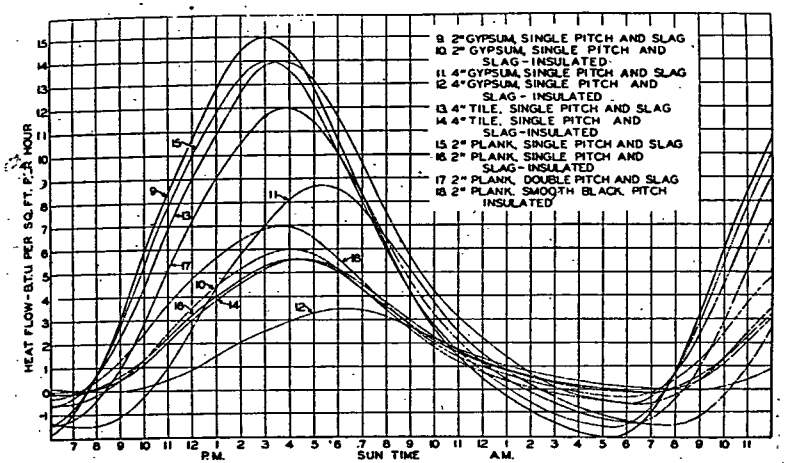


FIG. 3. HEAT FLOW-TIME RELATIONSHIP FOR HORIZONTAL ROOFS

intensity coming from all parts of the sky due to reflection and refraction. This scattered radiation intensity has been found to vary from a very low value to values as high as 20 per cent of the total radiation observed on certain days in Pittsburgh. The values shown on the curves in Fig. 1 are for combined direct solar and scattered sky radiation, and are given to represent expected design radiation intensity for August 1 for Pittsburgh. They were prepared by the A.S.H.V.E. Laboratory from data [1] obtained by pyrheliometer observations.

A study of the curves discloses the periodic relationship and wide variation in solar intensity on various surfaces. It will be observed that both the roof (horizontal surface) and south wall radiation curves are in exact phase relationship with each other, while those for the east and west walls overlap each other due to scattered sky radiation on the west wall during the forenoon and on the east wall during the afternoon. This phase relationship has an important bearing on the cooling load. Failure to consider the periodic character of heat flow resulting from diurnal movement of the sun and the lag due to heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall, may result in a large error in load calculations.

The values of solar intensity appearing in Fig. 1 must not be confused with the actual heat transmission through the wall, for much of the solar radiation impinging against the outer surface fails to pass through the wall. Instead it is delivered to the outside air by reflection, radiation, and convection. A mathematical solution for the determination of solar heat transmission has been developed but the equations involved are too complex for practical application [2]. A simpler method embodying an approximate determination of the maximum contribution to the cooling load due to heat transfer from the inside surface of a wall exposed to solar radiation has been developed [3].

The heat flow in summer through various types of roofs and walls has been measured [4,5] by the A.S.H.V.E. Laboratory. The curves in Figs. 2

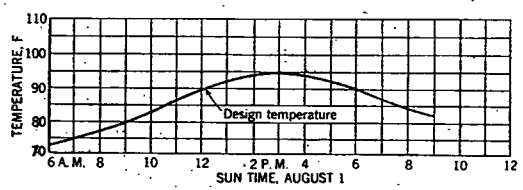


FIG. 4. OUTSIDE DESIGN TEMPERATURE BASIS FOR FIGS. 2 AND 3

and 3 give the heat flow through the inside surface of roofs [5] with details of the construction of the roofs tested. The conditions for which these results are given are: solar radiation for 40 deg north latitude on August 1 as given in Fig. 1, outdoor design temperature reaching a maximum of 95 F as shown in Fig. 4, and an indoor temperature of 75 F. Due to deviations in the tests from design temperature, solar conditions, and typical construction it is believed that the values from Figs. 2 and 3, as well as from Figs. 5, 6, 7, and 8 for walls, may be as much as 20 to 30 per cent below normal design expectations.

Curves in Figs. 5, 6, 7 and 8 were prepared by the A.S.H.V.E. Laboratory from tests made there and show the heat flow through the inside surface of three types of walls for various orientations [5]. The results are given for the following conditions: 90 per cent of the solar radiation given in Fig. 1 for 40 deg north latitude on August 1; outdoor design temperature reaching a maximum of 95 F, as shown in Fig. 4, and an indoor temperature of 78 F and 50 per cent relative humidity.

The heat flow shown in Figs. 2, 3, 5, 6, 7 and 8 is a combination of normal transmission and solar radiation transmission and is the total heat flow through the wall or roof. Due to the heat capacity of walls and roofs there is a time lag [2] in the transmission of heat through them as shown by the curves. For the types of construction covered in these figures and for the conditions indicated, the heat flow through the inside surface at any given time can be read directly. For other types of construction, the curves may be used as a guide in estimating the heat flow.

TABLE 3. TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

TYPE AND THICKNESS OF WALL OR ROOF	TIME LAG, HOURS
1-in. yellow pine horizontal roof, water proofing, smooth black finish	1
2-in. yellow pine horizontal roof, water proofing, smooth black finish	1¾
4-in. reinforced clay tile horizontal roof, water proofing, slag finish	2¼
2-in. gypsum horizontal roof, water proofing, slag finish	2¼
Slate and slaters felt on 2¼ in. tongue and grooved yellow pine, sloped roof	2½
4-in. gypsum horizontal roof, water proofing, slag finish	4½
6-in. concrete horizontal roof, water proofing, slag finish	5
1-in. concrete, 4-in. cinders, 1½-in. concrete, water proofing, smooth black finish	8
Wood siding, 1-in. sheathing, 2 x 4 studs, lath and plaster	2
Wood siding, 1-in. sheathing, 2 x 4 studs (studding space filled with insulation) lath and plaster	5
4-in. brick, 1-in. sheathing, 2 x 4 studs, lath and plaster	7
4-in. brick, 8-in. tile and plaster	10½
13-in. brick, plastered	12
9-in. brick, 3¾-in. tile, 5¼-in. air space, 3¾-in. tile and 1¼-in. plaster	16

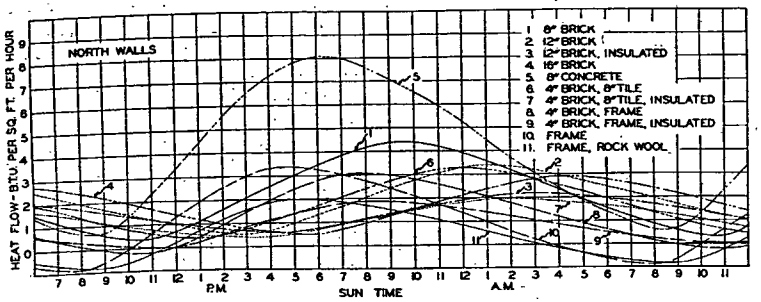


FIG. 5. HEAT FLOW-TIME RELATIONSHIP FOR NORTHERN EXPOSED WALLS

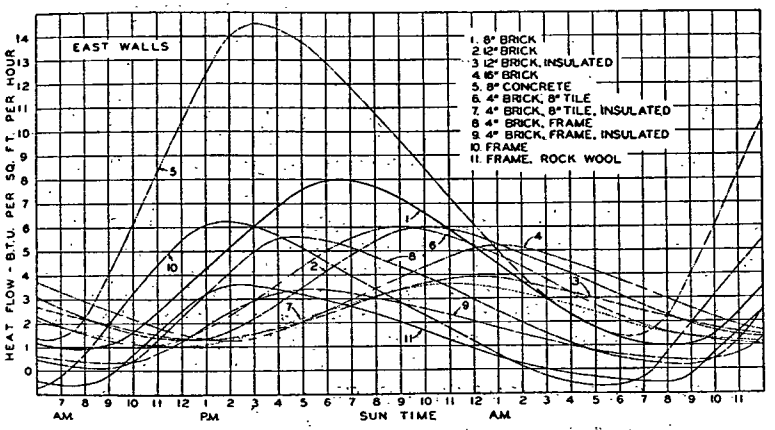


FIG. 6. HEAT FLOW-TIME RELATIONSHIP FOR EASTERN EXPOSED WALLS

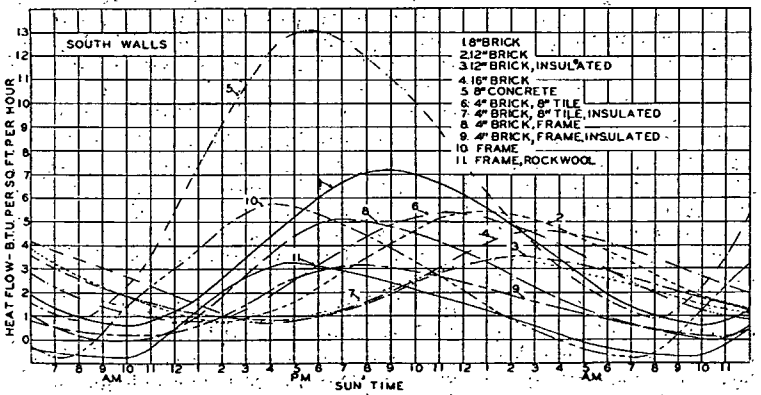


FIG. 7. HEAT FLOW-TIME RELATIONSHIP FOR SOUTHERN EXPOSED WALLS

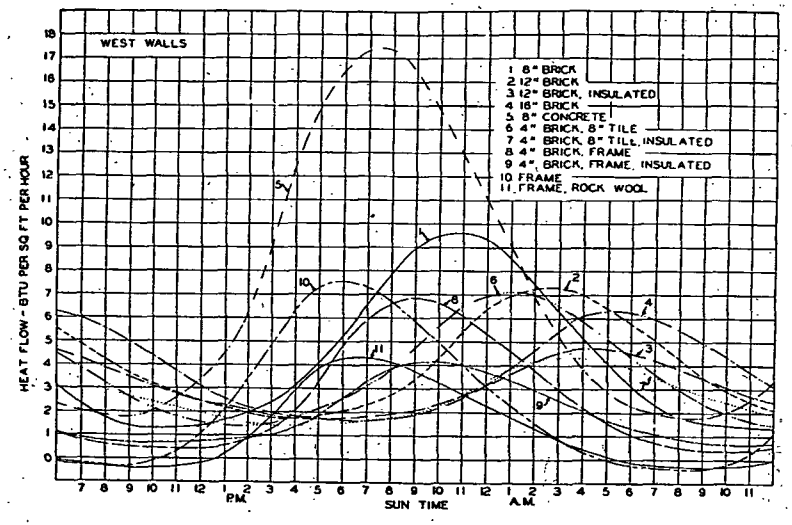


FIG. 8. HEAT FLOW-TIME RELATIONSHIP FOR WESTERN EXPOSED WALLS

The heat gain through southeast and southwest walls can be considered approximately the same as that through east and west walls respectively. The gain through northeast and northwest walls is less than half that through east and west walls respectively. The time lag for other types of construction is included in Table 3 which was prepared by the A.S.H.V.E. Laboratory from data collected by it and by other authorities.

Solar Radiation Transmitted Through Glass

Windows present a problem somewhat different from that of opaque walls, because they permit a large percentage of the solar energy to pass through. A small portion is reflected and an additional portion is absorbed by the glass. The reflected portion increases with the index of refraction and with the decrease of the angle of the sun with the glass. The absorbed portion increases with the glass thickness and with decreasing angle of the sun with the glass. It also depends on the character of glass; some *heat absorbing* glass having a very high absorptive value.

The absorbed heat warms the glass, causing an increase in the inward flow of heat due to temperature difference. In effect, therefore, a portion of the absorbed heat passes inward and can be considered as a part of the room load due to solar radiation. Particularly where *heat absorbent* glass or glass block is used without shades, the increase in temperature of the inner surface may be sufficiently great to cause serious added discomfort due to direct radiation of heat.

The amount of heat delivered to a room due to solar radiation may be obtained from Tables 4 to 9. These values should be used only for the net glass area on which the sun shines. In addition, to obtain the total heat gain, computation should be made of the heat transmission due to temperature difference according to Equation 1. The solar radiation data

TABLE 4. SOLAR RADIATION TRANSMITTED BY SINGLE UNSHADED GLASS FOR VARIOUSLY ORIENTED VERTICAL SURFACES AND A HORIZONTAL SURFACE

For 25 Deg North Latitude on August 1[a, b, c]

SUN TIME	SOLAR ALTITUDE	SOLAR RADIATION TRANSMITTED, BTU PER (HOUR) (SQUARE FOOT)							
		Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal
6 A.M.	7.5	77	84	41	3	3	3	3	20
7	20.5	145	173	98	10	10	10	10	90
8	34.0	148	193	121	13	13	13	13	155
9	47.5	123	174	123	15	15	15	15	230
10	61.5	67	129	98	23	16	16	16	279
11	74.5	28	60	60	31	16	16	16	312
12	83.0	16	16	25	31	25	16	16	324
1 P.M.	74.5	16	16	16	31	60	60	28	312
2	61.5	16	16	16	23	98	129	67	279
3	47.5	15	15	15	16	123	174	123	230
4	34.0	13	13	13	13	121	193	148	155
5	20.5	10	10	10	10	98	173	145	90
6	7.5	3	3	3	3	41	84	77	20

[a]The values of solar radiation are for a relatively clear atmosphere. In heavily industrialized and hazy atmosphere it may be necessary to reduce values as much as 10 per cent.

[b]Values are for sea level. Approximate correction for altitude can be made by increasing the values 1 per cent per thousand feet of altitude.

[c]Values in the tables may be used approximately for times of the year other than August 1 by choosing a table for a latitude differing from the geographical latitude by the difference of declination of the sun at the desired date and at August 1 (17.5 deg).

TABLE 5. SOLAR RADIATION TRANSMITTED BY SINGLE UNSHADED GLASS FOR VARIOUSLY ORIENTED VERTICAL SURFACES AND A HORIZONTAL SURFACE

For 30 Deg North Latitude on August 1[a, b, c]

SUN TIME	SOLAR ALTITUDE	SOLAR RADIATION TRANSMITTED, BTU PER (HOUR) (SQUARE FOOT)							
		Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal
6 A.M.	9.0	89	98	48	4	4	4	4	25
7	21.5	146	178	107	11	11	11	11	95
8	34.5	142	194	133	14	14	14	14	152
9	47.5	106	174	133	23	15	15	15	227
10	60.0	53	129	112	36	16	16	16	277
11	72.0	20	59	80	45	16	16	16	307
12	78.0	16	16	36	45	36	16	16	318
1 P.M.	72.0	16	16	16	45	80	59	20	307
2	60	16	16	16	36	112	129	53	277
3	47.5	15	15	15	23	133	174	106	227
4	34.5	14	14	14	14	133	194	142	152
5	21.5	11	11	11	11	107	178	146	95
6	9.0	4	4	4	4	48	98	89	25

[a]The values of solar radiation are for a relatively clear atmosphere. In heavily industrialized and hazy atmosphere it may be necessary to reduce values as much as 10 per cent.

[b]Values are for sea level. Approximate correction for altitude can be made by increasing the values 1 per cent per thousand feet of altitude.

[c]Values in the tables may be used approximately for times of the year other than August 1 by choosing a table for a latitude differing from the geographical latitude by the difference of declination of the sun at the desired date and at August 1 (17.5 deg).

TABLE 6. SOLAR RADIATION TRANSMITTED BY SINGLE UNSHADED GLASS FOR VARIOUSLY ORIENTED VERTICAL SURFACES AND A HORIZONTAL SURFACE

For 35 Deg North Latitude on August 1[a, b, c]

SUN TIME	SOLAR ALTITUDE	SOLAR RADIATION TRANSMITTED, BTU PER (HOUR) (SQUARE FOOT)							
		Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal
6 A.M.	10.0	98	109	54	4	4	4	4	31
7	22.5	143	179	110	11	11	11	11	100
8	34.5	133	194	140	39	14	14	14	167
9	46.5	93	173	144	31	15	15	15	225
10	58.5	41	128	131	50	16	16	16	270
11	68.5	16	59	94	62	45	16	16	299
12	73.0	16	16	45	68	45	16	16	309
1 P.M.	68.5	16	16	45	62	94	59	16	299
2	58.5	16	16	16	50	131	128	41	270
3	46.5	15	15	15	31	144	173	93	225
4	34.5	14	14	14	39	140	194	133	167
5	22.5	11	11	11	11	110	179	143	100
6	10.0	4	4	4	4	54	109	98	31

[a]The values of solar radiation are for a relatively clear atmosphere. In heavily industrialized and hazy atmosphere it may be necessary to reduce values as much as 10 per cent.

[b]Values are for sea level. Approximate correction for altitude can be made by increasing the values 1 per cent per thousand feet of altitude.

[c]Values in the tables may be used approximately for times of the year other than August 1 by choosing a table for a latitude differing from the geographical latitude by the difference of declination of the sun at the desired date and at August 1 (17.5 deg).

TABLE 7. SOLAR RADIATION TRANSMITTED BY SINGLE UNSHADED GLASS FOR VARIOUSLY ORIENTED VERTICAL SURFACES AND A HORIZONTAL SURFACE

For 40 Deg North Latitude on August 1[a, b, c]

SUN TIME	SOLAR ALTITUDE	SOLAR RADIATION TRANSMITTED, BTU PER (HOUR) (SQUARE FOOT)							
		Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal
5 A.M.	1.5	18	17	6	1	1	1	1	7
6	11.5	106	121	62	5	5	5	5	38
7	23.0	141	181	118	11	11	11	11	102
8	34.5	122	194	147	19	14	14	14	167
9	45.5	76	170	156	42	15	15	15	220
10	56.0	30	125	144	66	16	16	16	262
11	64.5	16	53	110	87	25	16	16	290
12	68.0	16	16	59	94	59	16	16	298
1 P.M.	64.5	16	16	25	87	110	53	16	290
2	56.0	16	16	16	66	144	125	30	262
3	45.5	15	15	15	42	156	170	76	220
4	34.5	14	14	14	18	147	193	122	167
5	23.0	11	11	11	11	118	181	141	102
6	11.5	5	5	5	5	62	121	106	38
7	1.5	1	1	1	1	6	17	18	7

[a]The values of solar radiation are for a relatively clear atmosphere. In heavily industrialized and hazy atmosphere it may be necessary to reduce values as much as 10 per cent.

[b]Values are for sea level. Approximate correction for altitude can be made by increasing the values 1 per cent per thousand feet of altitude.

[c]Values in the tables may be used approximately for times of the year other than August 1 by choosing a table for a latitude differing from the geographical latitude by the difference of declination of the sun at the desired date and at August 1 (17.5 deg).

TABLE 8. SOLAR RADIATION TRANSMITTED BY SINGLE UNSHADED GLASS FOR VARIOUSLY ORIENTED VERTICAL SURFACES AND A HORIZONTAL SURFACE

For 45 Deg North Latitude on August 1[a, b, c]

SUN TIME	SOLAR ALTITUDE	SOLAR RADIATION TRANSMITTED, BTU PER (HOUR) (SQUARE FOOT)							
		Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal
5 A.M.	2.0	23	23	17	1	1	1	1	11
6	12.5	111	129	68	6	6	6	6	44
7	23.0	135	182	144	11	11	11	11	103
8	33.5	115	190	151	24	14	14	14	163
9	44.0	63	168	163	55	15	15	15	214
10	53.0	24	123	154	88	16	16	16	251
11	60.0	16	56	127	113	30	16	16	277
12	63.0	16	16	76	119	76	16	16	285
1 P.M.	60.0	16	16	30	113	127	56	16	277
2	53.0	16	16	16	88	154	123	24	251
3	44.0	15	15	15	55	163	168	63	214
4	33.5	14	14	14	24	151	190	115	163
5	23.0	11	11	11	11	144	182	135	103
6	12.5	6	6	6	6	68	129	111	44
7	2.0	1	1	1	1	17	23	23	11

[a]The values of solar radiation are for a relatively clear atmosphere. In heavily industrialized and hazy atmosphere it may be necessary to reduce values as much as 10 per cent.

[b]Values are for sea level. Approximate correction for altitude can be made by increasing the values 1 per cent per thousand feet of altitude.

[c]Values in the tables may be used approximately for times of the year other than August 1 by choosing a table for a latitude differing from the geographical latitude by the difference of declination of the sun at the desired date and at August 1 (17.5 deg).

TABLE 9. SOLAR RADIATION TRANSMITTED BY SINGLE UNSHADED GLASS FOR VARIOUSLY ORIENTED VERTICAL SURFACES AND A HORIZONTAL SURFACE

For 50 Deg North Latitude on August 1[a, b, c]

SUN TIME	SOLAR ALTITUDE	SOLAR RADIATION TRANSMITTED, BTU PER (HOUR) (SQUARE FOOT)							
		Northeast	East	Southeast	South	Southwest	West	Northwest	Horizontal
5 A.M.	4.5	48	48	18	2	2	2	2	8
6	13.5	119	139	78	6	6	6	6	49
7	23.5	131	183	127	11	11	11	11	104
8	33.0	105	190	161	32	13	13	13	159
9	42.0	52	168	173	71	14	14	14	204
10	50.0	19	125	166	109	16	16	16	240
11	56.0	16	58	144	136	41	16	16	261
12	58	16	16	92	140	92	16	16	269
1 P.M.	56	16	16	41	136	144	58	16	261
2	50	16	16	16	109	166	125	19	240
3	42	14	14	14	71	173	168	52	204
4	33	13	13	13	32	161	190	105	159
5	23.5	11	11	11	11	127	183	131	104
6	13.5	6	6	6	6	78	139	119	49
7	4.5	2	2	2	2	18	48	48	8

[a]The values of solar radiation are for a relatively clear atmosphere. In heavily industrialized and hazy atmosphere it may be necessary to reduce values as much as 10 per cent.

[b]Values are for sea level. Approximate correction for altitude can be made by increasing the values 1 per cent per thousand feet of altitude.

[c]Values in the tables may be used approximately for times of the year other than August 1 by choosing a table for a latitude differing from the geographical latitude by the difference of declination of the sun at the desired date and at August 1 (17.5 deg).

TABLE 10. SOLAR RADIATION TRANSMITTED THROUGH SHADED WINDOWS

TYPE OF SHADING	FINISH	PORTION OF THAT TRANSMITTED BY UNSHADED WINDOW
Canvas awning	Dark	0.25–0.35
Inside venetian blind, slats at 45 deg, fully covering window	Aluminum	0.65–0.80
Inside roller shade, half drawn	Dark	0.90–0.95
Outside venetian blind, slats at 45 deg, extended as an awning without sides to cover approximately ⅔ of window	Light	0.35–0.50
Inside roller shade, fully drawn	Aluminum	Approx. 0.45
Inside roller shade, half drawn	Buff	Approx. 0.70
Outside venetian blind, slats at 45 deg, fully covering window	Aluminum	Approx. 0.60

in Tables 4 to 9 were prepared from solar radiation transmission data developed by the A.S.H.V.E. Research Laboratory [6] and direct radiation intensity values of Moon [7]. Included in these values are also an allowance for sky radiation and a reduction factor to take account of the radiation which is prevented from entering the room due to the glass. The notes accompanying the table give suggested factors of modification due to haze, elevation, time of year, etc. Table 10 gives the portion of the solar radiation transmitted to a room by an unshaded window for different indoor and outdoor shading fixtures according to tests [8] at the A.S.H.V.E. Research Laboratory. It is obvious that there are a number of factors which influence the magnitude of the values, including color, fit, elevation and angle of the sun to the wall and so forth. These values, therefore, can be considered approximate only, and will have to be used with considerable judgment.

Table 11 gives factors by which the solar radiation through unshaded glass should be multiplied to take into account the effect of difference from the usual transmissivity of glass (1 minus absorptivity minus reflectivity) for perpendicularly incident radiation. These tables are for use primarily with *heat absorbent glass*. The values given are only approximate, being based upon radiation perpendicular to the glass; and *are for use with single glass only*.

The total heat transmission through sunlit windows may be expressed by the equation:

$$H_g = A_g [fI + U_g (t_o - t_i)] \qquad (2)$$

where

H_g = heat transmitted through a window, Btu per hour.

A_g = net area of glass, square feet.

TABLE 11. MULTIPLYING FACTORS FOR GLASSES HAVING VARIOUS TOTAL ENERGY TRANSMISSIVITIES

TRANSMISSIVITY	FACTOR
0.900	1.02
0.875	1.00
0.800	0.93
0.700	0.85
0.600	0.76
0.500	0.68
0.400	0.60
0.300	0.52
0.200	0.45

f = factor (from Table 10) for adjustment of radiation transmitted through bare windows due to shading or heat absorbent glass (bare window = 1.0).

I = solar heat gain to room through bare glass, Btu per (hour) (square foot) (Tables 5-9).

U_g = coefficient of transmission of glass, Btu per (hour) (square foot) (Fahrenheit degree).

t_o = outside temperature, Fahrenheit degrees.

t_i = inside temperature, Fahrenheit degrees.

The maximum solar intensity on any surface is of limited duration as shown in Fig. 1. In the case of windows the total energy impinging on the glass before and after the time of maximum intensity is further reduced by increased shading of the glass from the frame, or wall. Another point which should be noted is that the maximum solar radiation load on the east wall occurs early in the morning when the outside temperature is low. Tests [9] have shown that where sunlight from bare windows impinges upon floors and walls of high heat capacity, the effect of the radiation is not immediately felt in the room. On the other hand, where shades or blinds are used, the heat may go directly into the air or it may rise to the top of the room and not be immediately reflected in the room load. For these reasons it is customary to assume a load somewhat below the maximum value; the amount of this reduction being largely a matter of individual judgment depending upon the particular problem in hand, but possibly being of the order of 10 to 20 per cent reduction from the maximum.

TABLE 12. HEAT GAIN THROUGH GLASS BLOCKS[a]

		SOLAR RADIATION HEAT GAIN (DIRECT PLUS SKY) BTU PER (SQ FT) (HOUR)						TOTAL HEAT GAIN[b] (SOLAR RADIATION PLUS NORMAL TRANSMISSION) BTU PER (SQ FT) (HOUR)					
SIDE		EAST[c]	WEST[c]	SOUTH				EAST[c]	WEST[c]	SOUTH			
N. LATITUDE DEGREES		40	40	30	35	40	45	40	40	30	35	40	45
Sun Time	Outside Temp F												
7:00	74	65.0		1.0	2.8	3.0	5.0	61.0		−4.5	−2.0	−0.5	1.0
8:00	76	63.0	0.0	3.0	4.4	6.5	11.0	77.5		0.0	2.0	4.0	5.0
9:00	79	40.0	5.0	5.5	7.1	10.2	13.4	73.5	5.0	5.0	7.0	10.0	12.0
10:00	83	24.0	6.0	8.5	11.3	14.7	17.1	57.5	6.5	11.0	15.0	18.0	20.8
11:00	87	15.5	7.0	12.0	15.2	18.7	21.8	45.0	7.5	16.5	22.0	25.5	32.0
12:00	90	10.0	10.0	14.0	17.4	21.0	24.8	36.5	10.5	21.5	28.0	33.8	40.8
1:00	93	7.0	15.5	12.0	15.2	18.7	21.8	30.0	22.0	25.0	31.8	38.5	46.0
2:00	94	6.0	24.0	8.5	11.3	14.7	17.1	24.0	35.0	26.0	32.0	39.0	47.0
3:00	95	5.0	40.0	5.5	7.1	10.2	13.4	19.5	55.0	24.0	29.8	36.5	45.0
4:00	95	4.5	65.0	3.0	4.4	6.5	11.0	15.5	77.0	20.0	25.5	31.5	40.5
5:00	93	4.0	63.0	1.0	2.8	3.0	5.0	12.5	85.5	15.0	20.0	25.2	33.5
6:00	91	2.5	23.5	0.0	0.7	0.7	3.0	10.5	55.0	9.5	13.5	18.0	25.5
7:00	89	1.5	0.0			0.0	0.7	8.0	18.5	3.5	7.0	11.0	18.0

[a] For August 1.

[b] Inside temperature, 78 F.

[c] For east and west walls these values can be applied to all latitudes between 30 and 45 deg N without excessive errors.

The direct solar and scattered sky radiation penetration through glass block panels is given in Table 12 for various times of the day for south, east and west exposures for different latitudes on August 1. This table

also gives the total heat gain into an air conditioned space when 78 F is maintained indoors, resulting from the effect of both radiation and air to air transmission. These values result from A.S.H.V.E. Laboratory data [1] and apply for expected design radiation intensity, and for a design day having a maximum temperature of 95 F. The resulting heat gains are averages for four typical glass block designs, two having smooth exterior faces, and the other two having exterior ribbed faces.

Heat Emission of Occupants

The heat and moisture given off by human beings under different states of activity are shown in various tables and figures of Chapter 12 which covers the physical and physiological principles of air conditioning. It will be observed from these data that the rate of sensible and latent heat emission by human beings varies greatly depending upon state of activity. In many applications this component becomes a large percentage of total load. Metabolic rates are markedly variable for some extreme environmental conditions and this is another important factor which must be considered in cooling load computations. Consideration should be given to the typical age and sex of the occupants (whether men, women, or children) and the duration of the occupancy (since for short occupancy applications the extra heat and moisture brought in by people may be an important factor in the load).

Heat Introduced by Outside Air

Heat and moisture are added by the outside air introduced for ventilation purposes and also by infiltration air entering the building through cracks, openings, and doors.

The volume of air entering due to infiltration may be estimated from data given in Chapters 8 and 14. Information on the amount of outside air required for ventilation will be found in Chapter 12. In general, ventilation air is supplied on a per person basis at a rate between 5 and 30 cfm per person (10 to 20 cfm being normal good practice), its specific amount depending upon the amount of contamination by smoke, food odors, etc.

As outside air is introduced through the air conditioning apparatus it tends to decrease infiltration by building up the pressure within the building. Usually, however, some infiltration still occurs unless the amount of outside air is very large. For instance, it is necessary to supply between 2,000 and 3,000 cfm of outside air through the apparatus to prevent infiltration through a single swinging door. As no demonstrated factors are generally available for the effect (on infiltration) of outside air introduced through the apparatus, it is necessary to use judgment in assigning an effect.

In determining the amount of outside air which must be drawn through the apparatus the amount of air needed for ventilation is established and it is normally possible to subtract from this the assumed amount of infiltration. The division between the ventilation provided by infiltration and by outside air is important to the designer, since the infiltration adds directly to the room sensible and moisture loads, whereas the outside air brought through the apparatus affects only the refrigeration load and the dehumidifier performance (except where shallow coils are used, in which case a portion of the outside air is effectively "by-passed" through the coil).

TABLE 13. HEAT GAIN FROM VARIOUS SOURCES[a]

	Source	Per Cent Name-plate Rating	Percentage or Btu per Hour		
			Sensible	Latent	Total
	Electrically Heated Equipment				
1.	Electric Oven-Baking	70	80%	20%	100%
2.	Baker's Oven		80%	20%	100%
3.	Coffee Urn—per Gallon Capacity	20	1025	1025	2050
4.	Glass Coffee Maker—per Section	30	90%	10%	100%
5.	Warming Receptacle	50	80%	20%	100%
6.	Plate Warmer	80	100%	0%	100%
7.	Hot Plates	100	65%	35%	100%
8.	Steam Table—Water Bath Type	75	90%	10%	100%
9.	Frying Griddles	70			
10.	Fry Kettle	40	75%	25%	100%
11.	Waffle Baker	60	90%	10%	100%
12.	Sandwich Grille	50	90%	10%	100%
13.	Toaster—Intermittent or Timed Control	100	90%	10%	100%
14.	Toaster—Continuous		2050		2050
15.	Hair Dryer in Beauty Parlor—600 w		2050		2050
16.	Permanent Wave Machine in Beauty Parlor—24-25 w Units				
	Gas Burning Equipment[b]				
17.	Gas Heated Oven—Baking—*no vent connection*	70	72%	28%	100%
	gravity vent connection	35	100%	0%	100%
18.	Open Top Burner—per Hole	50	45%	55%	100%
19.	Hot or Closed Top—per Burner—*no vent connection*	100	45%	55%	100%
	gravity vent connection	70	50%	50%	100%
20.	Coffee Urn—Large 18 in. Diam.—Single Drum		5000	5000	10,000
21.	Coffee Urn—Small 12 in. Diam.—Single Drum		3000	3000	6000
22.	Coffee Urn—per Gallon of Rated Capacity		500	500	1000
23.	Plate Warmer—*no vent connection*	60	90%	10%	100%
	gravity vent connection	10	100%	0%	100%
24.	Steam Table—Water Bath Type—*no vent connection*	60	55%	45%	100%
	gravity vent connection	42	50%	50%	100%
25.	Frying Griddles—*no vent connection*	75	81%	19%	100%
	gravity vent connection	52	90%	10%	100%
26.	Fry Kettle—*no vent connection*	70	72%	28%	100%
	gravity vent connection	49	80%	20%	100%
27.	Egg Boiler—per Egg Compartment		2500	2500	5000
28.	Cigar Lighter—Continuous Flame Type		2250	250	2500
29.	Curling Iron Heater		2250	250	2500
30.	Pilot Light		180	20	200
	Steam Heated Equipment[c]				
31.	Steam Heated Surface Not Polished—per Square Foot of Surface		330	0	330
32.	Steam Heated Surface Polished—per Square Foot of Surface		130	0	130
33.	Insulated Surface—per Square Foot		80	0	80
34.	Bare Pipes, Not Polished—per Square Foot of Surface		400	0	400
35.	Bare Pipes, Polished—per Square Foot of Surface		220	0	220
36.	Insulated Pipes—per Square Foot		110	0	110
37.	Coffee Urn—Large, 18 in. Diam.—Single Drum		2000	2000	4000
38.	Coffee Urn—Small, 12 in. Diam.—Single Drum		1200	1200	2400
39.	Egg Boiler—per Egg Compartment		2500	2500	5000
40.	Steam Table—per Square Foot of Top Surface		300	800	1100
	Miscellaneous				
41.	Heat Liberated By Food per Person, as in a Restaurant		30	30	60
42.	Heat Liberated from Hot Water used direct and on towels per hour—Barber Shops		100	200	300

[a] Heat gain from electric or gas residential ranges or cooking stoves depends on size of the family, socio-economic status of the individual, time of day for principal meal, and whether the equipment is manually or automatically controlled. Total heat gain will probably not exceed 40 per cent name-plate rating. Per cent sensible and latent heat will depend upon use of equipment; dry heat; baking or boiling.

[b] Name-plate ratings of gas burning equipment can be obtained from a Directory of Approved Gas Appliances and Listed Accessories, obtainable from *American Gas Association*.

[c] Steam Requirements of Process Equipment, Report of the Commercial Relations Committee, *National District Heating Association* (*Heating, Piping and Air Conditioning*, November, 1942, p. 675).

The total heat gain resulting from outside air introduced may be determined by Equation 3:

$$H = \frac{Q}{v}(h_o - h_i) \qquad (3)$$

where

H = heat to be removed from outside air entering the building above inside conditions, Btu per hour.

Q = volume of outside air entering building, cubic feet per hour.

v = cubic feet of outside air per pound of dry air.

h_o = enthalpy of outside air, Btu per pound of dry air.

h_i = enthalpy of inside air, Btu per pound of dry air.

The moisture gain resulting from outside air and infiltration may be determined by Equation 4:

$$Q_w = \frac{Q}{v}(W_o - W_i) \qquad (4)$$

where

Q_w = weight of water to be removed from outside air above inside conditions, pounds per hour.

W_o = humidity ratio of outside air, pounds water per pound dry air.

W_i = humidity ratio of inside air, pounds water per pound dry air.

Heat Emission of Appliances

Heat generating appliances which give off either sensible heat or both sensible and latent heat in an air conditioned enclosure may be divided into three general classes of equipment or devices: (1) electrical appliances, (2) gas appliances, and (3) steam heating appliances.

In the first group may be found such devices as lights[10], fans, motors, toasters, waffle irons, etc. The heat load caused by such devices may normally be obtained by multiplying the nameplate rating in watts by an appropriate load factor and by 3.4 (Btu per watt hour). In some cases it may also be possible to remove some of the heat of such appliances as lights and motors with exhaust air without involving it in the room load.

Electric motors are usually rated in units of horsepower *output*. To determine the corresponding input, which is the rate at which heat is added to the conditioned space by full-load operation of such motors, some idea of motor efficiency is necessary. The aggregate input in horsepower should then be multiplied by 2546 (Btu per horsepower hour).

Motor efficiencies can be assumed about as follows: Motor efficiencies vary from 50 to 60 per cent at the ⅛ hp level to 80 per cent at 1 hp and 88 per cent at 10 hp and above. Where the motor is outside of the conditioned space the heat equivalent of the motor output only is used, but where the motor is inside of the space the heat equivalent of the output divided by the efficiency is used.

In the second group belong such appliances as coffee urns, gas ranges, steam tables, broilers, hot plates, etc. For heat generating capacities of such appliances refer to Table 13.

Judgment must be used in the application of data given in Table 13. 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 allowing 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 per cent of the heat will be carried away and 50 per cent dissipated in the space to be conditioned. The same effectiveness of the hood should be figured for both latent and sensible heat.

TABLE 14. PERMEABILITY OF VARIOUS MATERIALS TO WATER VAPOR

MATERIAL	PERMEABILITY GRAINS PER (SQ FT) (HR) (INCH HG)
	Group 1[a]
Plaster base and plaster, ¾ in.	14.7
Fir sheathing, ¾ in.	2.9
Waterproof paper[c]	49.1
Pine lap siding	4.9
Paint film	3.4
Sugar cane fiberboard, ¾ in.	12.5
Brick masonry, 4 in.	1.1
	Group 2[b]
Foil-surfaced reflective insulation, double-faced	0.08 to 0.13
Roll roofing—smooth, 40 to 65 lb/roll 108 sq ft	0.13 to 0.17
Duplex or laminated papers, 30-30-30	1.37 to 2.58
Plaster, wood lath	11.00
Plaster, 3 coats of lead and oil	3.68 to 3.84
Plaster, 2 coats of Aluminum paint	1.15
Plaster, fiberboard or gypsum lath	19.73 to 20.57
Plywood, ½ in., 5-ply Douglas fir	2.67 to 2.74
Insulating lath and sheathing, board type	25.68 to 34.27
Insulating sheathing, surface-coated	3.03 to 4.36
Insulating cork blocks, 1 in.	6.19
Mineral wool, unprotected, 4 in.	29.07

[a]Calculating Vapor and Heat Transfer Through Walls, by L. G. Miller (*Heating and Ventilating* 35, No. 11, 56 November, 1938).

[b]How to Overcome Condensation in Building Walls and Attics, by L. V. Teesdale (*Heating and Ventilating*, Vol. 36, No. 4, April, 1939).

[c]Light weight slaters felt used to keep rain from drifting through. Not used as a vapor barrier.

Moisture Through Walls

In some applications walls of the conditioned space may be in contact with other spaces which have in them a higher water vapor pressure than that in the conditioned space. It is known that water vapor will flow through the building materials in proportion to the vapor pressure difference on the two sides of the material. The total amount of water vapor transmitted is dependent on the permeability which is usually expressed in grains of moisture per (square foot) (hour) (inch of mercury vapor pressure difference). The values for permeability in Table 14 are quoted from a publication of the *National Bureau of Standards*[11]. The water vapor entering the conditioned space must be added to the latent cooling load.

Vapor barriers, to be effective in reducing entrance of moisture, must seal completely the walls, ceilings, and floors that are exposed to space

having excessive vapor pressure and all doors must have gaskets applied to them to make the barrier effective.

ILLUSTRATION

From the foregoing discussion it is obvious that the determination of the maximum cooling load is rather complicated by reason of the variable nature of contributing load components. An illustrative example will explain the method presented in the foregoing text.

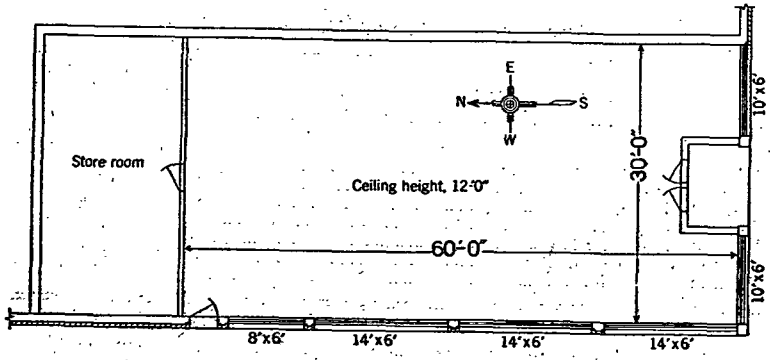


FIG. 9. PLAN DIAGRAM OF CLOTHING STORE

Example 1. Determine cooling load requirements for a clothing store illustrated in Fig. 9 and located in Pittsburgh, Pa., Latitude 40 deg. This is a one-story building located on a corner and it faces south and west. Assume building on east and north sides conditioned.

Wall construction, 8 in. concrete block, 4 in. brick veneer, plaster on walls, $U = 0.33$ (Table 8, Chapter 6, No. 93 B).

Roof construction, 2 in. concrete, ½ in. insulating board, metal lath and plaster ceiling, $U = 0.26$ (Table 15, Chapter 6, No. 14 B).

Floor, maple flooring on yellow pine, no ceiling below, $U = 0.34$ (Table 11, Chapter 6, No. 1-N).

Partition, wood lath and plaster on both sides of studding, $U = 0.34$ (Table 9, Chapter 6, No. 3 B).

Windows, provided with awnings.

Front doors, 2 ft 6 in. x 7 ft (glass paneled).

Side door, 3 ft x 7 ft (glass paneled), $U = 1.13$ (Table 18 A, Chapter 6).

Occupancy, 10 clerks, 40 patrons.

Lights, 4200 w.

Outside design conditions, dry-bulb 95 F; wet-bulb 75 F.

Inside design conditions, dry-bulb 80 F; wet-bulb 67 F.

Basement temperature, 85 F.

Store room temperature, 88 F.

Solution. It is obvious from the shape and exposure of this store and the large glass area on the west side that the maximum cooling load will occur during the afternoon when the sun is shining on the west wall. From Fig. 1, the peak load may be expected at 4:00 p.m.

The combined normal transmission and solar radiation transmission through the roof at 4:00 p.m. is obtained from Fig. 2. While none of the roofs in Fig. 2 is exactly like this one, roof 4 is similar. A heat flow of 8½ Btu per (square foot) (hour) was assumed, 20 per cent more than for roof 4. The combined normal transmission and solar radiation transmission through the south and west walls at 4:00 p.m. is obtained from curves in Figs. 7 and 8. Note that the curve values for combined wall transmission have been increased by a judgment factor of 20 per cent. Also, that the values are still only about half what would be obtained due to normal transmission taking the instantaneous temperature difference.

The normal heat transmission through the south and west glass, floor and partition is determined by application of Equation 1. The solar radiation effect passing through bare south and west glass is determined from Table 7 as being 18 and 193 Btu per (square foot) (hour) respectively. This is multiplied by a haze factor of 0.9 (10 per cent reduction) and by a factor of 0.25 to take account of the awning (Table 10). It will be seen that the sun effect on the south glass might well have been neglected.

To determine the load imposed by occupants, it will be found from Chapter 12 that the average person standing at rest will dissipate 431 Btu per hour, and that the moisture dissipated is 0.198 lb per hour.

The amount of infiltration through a swinging door is determined from Table 3, Chapter 8 as being 3½ cfm per person for 40 patrons or 140 cfm total. Since the windows are sealed it can be assumed that this is the only infiltration which will occur. The infiltration affects both the sensible and latent load of the room and, therefore, must be considered separately from the outside air passed through the apparatus for ventilation purposes which affects the room conditions little or none. (See Chapter 7 and 43 for treatment of "by-passed" outside air as a part of the room load.)

The ventilation requirements as given in Table 1, Chapter 12 would indicate the need of about 9 cfm per person based on a volume of the store of 21,600 cu ft and 50 persons making a total volume of 450 cu ft. However, the ventilation requirements in the summer are somewhat greater than those in the winter and, therefore, 10 cfm per person or 500 cfm total will be chosen. From this should be subtracted the estimated infiltration of 140 cfm, making a net amount of outside air to be drawn through the apparatus of 360 cfm.

The total heat addition due to the infiltration and outside air is determined from Equation 3, and the moisture addition is determined from Equation 4.

The room total load is then made up of normal transmission, combined normal and sun radiation transmission, solar radiation through glass, heat from occupants, total heat addition due to infiltrations, and heat from lights.

It is necessary to determine only the amount of water condensed from the air (not the latent heat). In considering the room load this consists of the water evaporated from the people plus the excess water in the infiltration air.

The excess heat and moisture load due to the outside air tends to raise the return air temperature and moisture content, but if the dehumidifier is 100 per cent effective and fully saturates the leaving air, the outside air load has no direct effect upon the room. It, therefore, should be determined separately and added only in connection with the determination of the refrigeration requirements.

NORMAL TRANSMISSION:

SURFACE	DIMENSIONS	AREA SQ FT	U	TEMP DIFF F DEG	BTU PER HOUR
S Glass	2 (2 ft 6 in. x 7 ft) + 2 (10 ft x 6 ft)	155	1.13	15	2,627
W Glass	3 (14 ft x 6 ft) + (8 ft x 6 ft) + (3 ft x 7 ft)	321	1.13	15	5,445
Floor	26 ft x 54 ft	1404	0.34	5	2,387
N Partition	30 ft x 12 ft	360	0.34	8	979
Total					11,438

COMBINED NORMAL AND SOLAR RADIATION TRANSMISSION:

SURFACE	DIMENSIONS	AREA SQ FT	BTU PER (HOUR) (SQ FT)	BTU PER HOUR
S Wall	(30 ft x 12 ft) − 155	205	2 x 1.2	490
W Wall	(60 ft x 12 ft) − 321	399	2 x 1.2	960
Roof	60 ft x 30 ft	1800	8½	15,300
Total				16,750

SOLAR RADIATION THROUGH GLASS:

South $H_g = 155 \times (18 \times 0.9) \times 0.25 = 625$
West $H_g = 321 \times (193 \times 0.9) \times 0.25 = 13{,}930$ (Equation 2)

OCCUPANTS:

$50 \times 431 = 21{,}550$ Btu per hour.

$50 \times 0.198 = 9.90$ lb water evaporated per hour.

INFILTRATION AND OUTSIDE AIR:

$$H = \frac{Q}{v}(h_o - h_i) \quad \text{(Equation 3)}$$

v = 13.33 cu ft per pound dry air (infiltration, and outside air required for ventilation may be considered in terms of Standard Air).

μ = per cent saturation at 95 F dry-bulb and 75 F wet-bulb = 38.4 per cent (by calculation, Chapter 3).

$h_o = h_a + \mu h_{as}$ (Equation 5, Chapter 3).

h_a = specific enthalpy of dry air at 95 F = 22.827 Btu per pound (Table 1, Chapter 3).

h_{as} = difference between enthalpy of saturated mixture and specific enthalpy of dry air at 95 F = 40.49 Btu per pound (Table 1, Chapter 3).

$h_o = 22.80 + (0.384 \times 40.49) = 38.26$ Btu per pound dry air.

μ at 80 F dry-bulb and 67 F wet-bulb = 50.2 per cent (by calculation, Chapter 3).

$h_i = h_a + \mu h_{as} = 19.221 + (0.502 \times 24.47) = 31.50$ Btu per pound dry air (Table 1, Chapter 3).

$$H \text{ (infiltration)} = \frac{140 \times 60}{13.33}(38.26 - 31.50) = 4{,}270 \text{ Btu per hour.}$$

$$H \text{ (outside)} = \frac{(500 - 140)60}{13.33}(38.26 - 31.50) = 10{,}950 \text{ Btu per hour.}$$

W_o = humidity ratio of outside air at 95 F dry-bulb and 75 F wet-bulb = 0.384 × 0.03673 = 0.01410 lb water per pound dry air. (Equation 3, Chapter 3.)

W_i = humidity ratio of inside air at 80 F dry-bulb and 67 F wet-bulb = 0.502 × 0.02233 = 0.01121 lb water per pound dry air. (Equation 3, Chapter 3.)

Weight of water to be removed from infiltration air $= \frac{Q}{v}(W_o - W_i)$ (Equation 4)

$$= \frac{140 \times 60}{13.33}(0.01410 - 0.01121) = 1.82 \text{ lb per hour.}$$

LIGHTS: $4200 \times 3.413 = 14{,}335$ Btu per hour.

SUMMARY:

COMPONENTS OF LOAD	BTU PER HOUR
Normal Transmission	11,438
Combined Normal and Solar Radiation Transmission	16,750
Solar Radiation Through Glass	14,555
Occupants	21,550
Infiltration Air	4,270
Lights plus Heat Equivalent of that Proportion of the Power Load Used in Air Conditioning System Which Adds Heat to the Air	14,335
Total Room Load (Sensible and Latent Heat Gain)	82,898
Outside Air (Through Apparatus)	10,950
Grand Total Load	93,848

DETERMINATION OF APPARATUS DEW-POINT:

First, determine the water added as a part of the room load as follows:

Occupants	9.9 lb water per hour
Infiltration	1.82 lb water per hour
	11.72 lb water per hour

The condition line for this application may be determined from Equation 11 in Chapter 3 by taking the ratio of room total load to pounds water per hour released in the room = 82,898 ÷ 11.72 = 7075 Btu per pound of water. In this case, it crosses the saturation curve at a temperature for which the enthalpy h_s and the humidity ratio W_s satisfy the equation,

$$\frac{31.50 - h_s}{0.01121 - W_s} = 7075$$

The required apparatus dew-point may be determined graphically from the Mollier chart or by cut and try from Table 1, Chapter 3, as 57.9 F.

AIR QUANTITY AND REFRIGERATION LOAD:

Assuming 100 per cent saturation efficiency for the air conditioning apparatus, the air will leave the dehumidifier saturated. The thermodynamic properties involved in calculating the air quantity are:

	INSIDE AIR	AFTER COOLING
t	80.0	57.9
h	31.50	25.06
W	0.01121	(0.01030 Vapor + 0.00091 Liquid)

The cooling effect is 31.50 − 25.06 = 6.372 Btu per pound dry air. The total outside and recirculated dry air through the air conditioning apparatus is:

$$\frac{82{,}898 \times 13.33}{6.44 \times 60} = 2860 \text{ cfm (Standard Air)}$$

The refrigeration may be assumed equal to the grand total load; that is,

$$\frac{93{,}848}{12{,}000} = 7.82 \text{ tons}$$

REFERENCES

1—A.S.H.V.E. RESEARCH REPORT No. 1147—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson and Burt Gunst (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 83).

2—A.S.H.V.E. RESEARCH REPORT No. 923—Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 231). Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 369). Periodic Heat Flow in Building Walls Determined by Electrical Analogy Method, by Victor Paschkis (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 75). Summer Comfort Factors as Influenced by the Thermal Properties of Building Materials, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 148).

3—Periodic Heat Flow—Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 293).

4—A.S.H.V.E. RESEARCH REPORT No. 1157—Summer Cooling Load as Affected by Heat Gain Through Dry, Sprinkled and Water Covered Roofs, by F. C. Houghten, H. T. Olson and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 231).

5—A.S.H.V.E. RESEARCH REPORT No. 1195—Heat Gain Through Walls and Roofs as Affected by Solar Radiation, by F. C. Houghten, E. C. Hach, S. I. Taimuty and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 91).

6—The Transmission of Solar Radiation Through Flat Glass Under Summer Conditions, by G. V. Parmelee (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October-November, 1945, p. 562).

7—Proposed Standard Solar Radiation Curves for Engineering Use, by P. Moon (*Journal of the Franklin Institute*, November, 1940, Vol. 230, No. 5, p. 583-617).

8—A.S.H.V.E. RESEARCH REPORT No. 975—Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 101). A.S.H.V.E. RESEARCH REPORT No. 1180—Heat Gain Through Western Windows With and Without Shading, by F. C. Houghten, and David Shore (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 251).

9—A.S.H.V.E. RESEARCH REPORT No. 1002—Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet, and Albert J. Wahl (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 53).

10—Cooler Footcandles for Air Conditioning, by W. G. Darley (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 367). Lighting and Air Conditioning Design Factors, Report of *I.E.S.*—A.S.H.V.E. Joint Committee on Lighting in Air Conditioning (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1941, p. 605). Lighting and Air Conditioning, by Howard M. Sharp (*Heating and Ventilating*, November, 1942, p. 35).

11—Moisture Condensation in Building Walls, by Harold W. Woolley (*U. S. Department of Commerce, National Bureau of Standards*, Building Materials and Structures Report BMS63).

CHAPTER 16

Fuels and Combustion

Combustion Principles, Heat and Flames; Air Required, Excess Air and Secondary Air; Draft Requirements, Classification of Coal and Coke, Firing Methods, Classification and Combustion of Oil and Gas; Dustless Treatment of Coal

THE data given in the first part of this chapter are of general application to the various fuels used in domestic heating which are coal, coke, oil and gas. The choice of fuel is a question of dependability, cleanliness, fuel availability, economy, operating requirements and control.

FUNDAMENTAL PRINCIPLES OF COMBUSTION

Combustion may be defined as the chemical combination of a substance with oxygen with a resultant evolution of heat. The rate of combustion depends partly upon the specific rate of reaction of the combustible substance with oxygen, partly upon the rate at which oxygen is supplied, and upon the temperature obtained due to surrounding conditions.

Complete combustion is obtained when all of the combustible elements in the fuel are oxidized with all of the oxygen with which they can combine. All of the oxygen supplied may not be utilized.

Perfect combustion is defined as the result of supplying the required amount of oxygen for combination with all of the combustible elements of the fuel and utilizing all of the oxygen so supplied.

The oxygen required for the process of combustion is obtained from air which is a mechanical mixture of oxygen, nitrogen and small amounts of carbon dioxide, water vapor and inert gases. These inert gases are generally included with the nitrogen, and for engineering purposes the values given herewith may be used.

	By Volume, Per Cent	By Weight, Per Cent
Oxygen, O_2	20.9	23.15
Nitrogen, N_2	79.1	76.85

The combination of oxygen with the combustible elements and compounds of a fuel is in accordance with fixed laws. In the case of perfect combustion the reactions and resultant combinations are shown in Table 1.

The most important condition governing the process of combustion is temperature. It is necessary to bring a combustible substance to its ignition temperature before it will unite in chemical combination with oxygen to produce combustion. The ignition temperatures for several of the combustible constituents of fuels are presented in Table 1.

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.

TABLE 1. GENERAL DATA OF COMBUSTIBLE ELEMENTS AND COMPOUNDS [a]

Substance	Molecular Symbol	Chemical Reaction of Combustion	Ignition Temperature [a] F Deg	Calorific Value, Btu per Lb		Calorific Value, Btu per Cu Ft [b]	Theoretical Oxygen and Air Requirements, Lb per Lb		Theoretical Oxygen and Air Requirements, Cu Ft per Cu Ft,	
				Higher	Lower	Higher	O_2	Air	O_2	Air
Carbon (to CO)	—	$2C + O_2 = 2CO$	—	3950	—	—	1.332	5.763	—	—
Carbon (to CO_2)	—	$2C + 2O_2 = 2CO_2$	—	14093	—	—	2.664	11.527	—	—
Sulphur (to SO_2)	—	$S + O_2 = SO_2$	—	3983	—	—	0.998	4.285	—	—
Sulphur (to SO_3)	—	$2S + 3O_2 = 2SO_3$	—	5940	—	—	1.497	6.428	—	—
Carbon Monoxide	CO	$2CO + O_2 = 2CO_2$	1166-1319	4347	—	321.8	0.571	2.471	0.5	2.382
Methane	CH_4	$CH_4 + 2O_2 = CO_2 + 2H_2O$	1260-1380	23879	21520	1013.2	3.990	17.265	2.0	9.528
Acetylene	C_2H_2	$2C_2H_2 + 5O_2 = 4CO_2 + 2H_2O$	763-824	21500	20776	1499	3.073	13.297	2.5	11.911
Ethylene	C_2H_4	$C_2H_4 + 3O_2 = 2CO_2 + 2H_2O$	986-1123	21644	20295	1613.8	3.422	14.807	3.0	14.293
Ethane	C_2H_6	$2C_2H_6 + 7O_2 = 4CO_2 + 6H_2O$	990-1120	22320	20432	1792	3.725	16.119	3.5	16.675
Hydrogen	H_2	$2H_2 + O_2 = 2H_2O$	1063-1166	60958 [c]	51571 [c]	325	7.937	34.344	0.5	2.382
Hydrogen Sulphide	H_2S	$2H_2S + 3O_2 = 2H_2O + 2SO_2$	599-608	7100	6545	647	1.409	6.097	1.5	7.146
Propane	C_3H_8	$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O$	950-1080	21661	19944	2590	3.629	15.703	5.0	23.821
n-Butane	C_4H_{10}	$2C_4H_{10} + 13O_2 = 8CO_2 + 10H_2O$	890-1020	21308	19680	3370	3.579	15.487	6.5	30.967
Commercial Propane	—	—	—	21650	—	2500	—	—	—	—
Commercial Butane	—	—	—	21400	—	3200	—	—	—	—

[a] Values in table taken chiefly from page 51 of Fuel Flue Gases published by *American Gas Association*.
[b] Value from *National Bureau of Standards*.
[c] Gas measured at 60 F and 30 in. Hg.

The heat of combustion of the several fuel elements and compounds in their *pure* state is given in Table 1.

The reaction of the carbon in the fuel with oxygen may result in the formation of carbon monoxide or carbon dioxide. In burning to carbon monoxide, the carbon is not completely oxidized and, as shown by the data, the heat produced is considerably less than if it were completely oxidized. This fact is of greatest importance in considering the efficiency of combustion.

The calorific value of a fuel is determined by direct measurement of the heat evolved during combustion in a calorimeter. Although the ash and moisture content of coal from a given mine or locality may vary widely, the heating value of the coal, on a *moisture and ash free* basis, remains relatively constant. It is therefore possible to approximate the heating value of a shipment of coal *as received* if its moisture and ash content are determined, and if the heating value of similar coal on a moisture and ash free basis is known. This may be calculated by Equation 1.

$$\text{Heating value, } \textit{as received} = \frac{\text{Heating value, } \textit{moisture and ash free} \times [100 - (\text{Moisture} + \text{Ash})]}{100} \qquad (1)$$

where, moisture and ash are expressed in per cent.

Typical analyses of the coals of the United States are given in *U. S. Bureau of Mines Bulletin* 446.

As practically all fuels, including coal, oil, and gas, contain hydrogen, water vapor is one of the products of combustion. The amount of water vapor produced is proportional to the hydrogen content of 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, the products of combustion are not cooled to the dew-point and the higher heating value cannot be obtained.

FLAME

The appearance of the flame or products of combustion may serve as an approximate measure of the temperatures developed in the combustion process. The luminosity of a flame is caused by the heating to incandescence of unconsumed particles of combustible matter in the gases, and the higher the temperature of these particles the whiter the flame. Table 2 gives some approximate flame temperature data.

TABLE 2. APPROXIMATE FLAME TEMPERATURE DATA

Appearance of Flame	Temperature F Deg
Red, visible in daylight	1000
Light red	1800
Orange-red	2000
Orange-yellow	2200
Yellow-white	2400
Bright white	2600

AIR REQUIRED FOR COMBUSTION

The weight of air required for the perfect combustion of a pound of fuel may be determined by use of the ultimate analysis of the fuel as applied to Equations 2 to 4. The various elements are expressed in percentages by weight.

Solid and Liquid Fuels:

$$\text{Pounds air required per pound fuel} = 34.56\left[\frac{C}{3} + \left(H - \frac{O}{8}\right) + \frac{S}{8}\right] \quad (2)$$

Gaseous Fuels:

$$\text{Pounds air required per pound fuel} = 2.46\,CO + 34.56\,H_2 + 17.28\,CH_4 + 13.29\,C_2H_2 + 14.81\,C_2H_4 + 16.13\,C_2H_6 + 6.10\,H_2S - 4.32\,O_2 \quad (3)$$

When the analysis is given on a volumetric basis the equation is expressed as follows:

$$\text{Cubic feet air required per cubic foot gas} = 2.39\,(CO + H_2) + 9.56\,CH_4 + 11.98\,C_2H_2 + 14.35\,C_2H_4 + 16.74\,C_2H_6 - 4.78\,O_2 \quad (4)$$

Equations 5 and 6 may be used as approximate methods of determining the theoretical air requirement for any fuel.

$$\text{Pounds air required per pound fuel} = 0.755 \times \frac{\text{Heating value (Btu per pound)}}{1000} \quad (5)$$

$$\text{Cubic feet air required per unit fuel} = \frac{\text{Heating value (Btu per unit)}}{100} \quad (6)$$

Approximate values for the theoretical air required for different fuels are given in Table 3.

It is customary to make use of the analysis of the products of combustion to determine the amount of flue gas produced and the actual

TABLE 3. APPROXIMATE THEORETICAL AIR REQUIREMENTS

Solid Fuel	Pounds Air Per Pound Fuel
Anthracite	9.6
Semi-bituminous coal	11.2
Bituminous coal	10.3
Lignite	6.2
Coke	11.2

Fuel Oil	Pounds Air Per Gallon Fuel
Commercial Standard No. 1	102.6
Commercial Standard No. 2	104.5
Commercial Standard No. 3	106.5
Commercial Standard No. 5	112.0
Commercial Standard No. 6	114.2

Gaseous Fuels	Cubic Feet Air Per Cubic Foot Gas
Natural gas	10.0
Mixed, natural and water gas	4.4
Carbureted water gas	4.4
Water gas, coke	2.1
Coke oven gas	5.2

amount of air supplied for combustion. The analysis of flue gases has been well described in various publications of the *Bureau of Mines* and in the literature and the details of Orsat manipulation need not be considered in this discussion. (See Chapter 11.)

The weight of dry flue gas per pound of fuel burned is used in combustion loss calculations and may be determined by Equation 7.

$$\text{Pounds dry flue gas per pound fuel} = \frac{11\,CO_2 + 8\,O_2 + 7\,(CO + N_2)}{3\,(CO_2 + CO)} \times C \quad (7)$$

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.

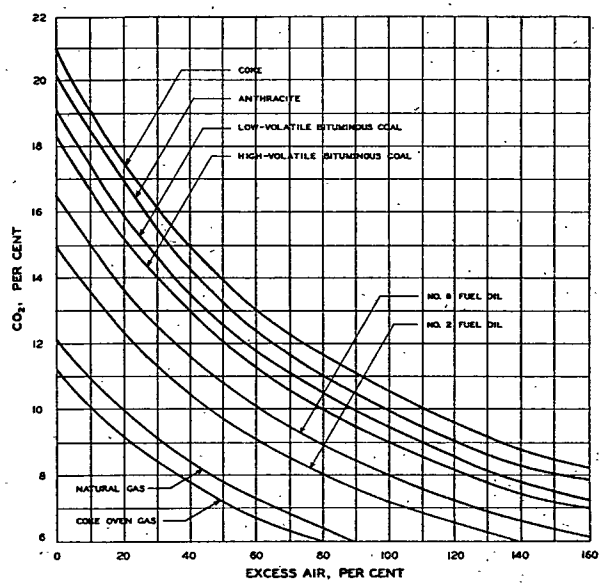


FIG. 1. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

EXCESS AIR

Since one measure of the efficiency of combustion is the relation existing between the amount of air theoretically required for *perfect* combustion and the amount of air actually supplied, a method of determining the latter factor is of value. Equation 8 will give reasonably accurate results, for most solid and liquid fuels, for determining the amount of air supplied per pound of fuel.

$$\text{Pounds dry air supplied per pound of fuel} = \frac{3.04\,N_2}{(CO_2 + CO)} \times C \quad (8)$$

Values for CO_2, CO and N 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.

The difference between the air actually supplied for combustion and the theoretical air required is known as excess air.

$$\text{Per cent excess air} = \frac{\text{Air supplied} - \text{Theoretical air}}{\text{Theoretical air}} \quad (9)$$

Since the calculation is usually made from Orsat readings, Equation 10 will be found to be a convenient statement of this relationship.

$$\text{Per cent excess air} = \frac{100\left(O_2 - \frac{CO}{2}\right)}{N_2 \times 0.264 - \left(O_2 - \frac{CO}{2}\right)} \quad (10)$$

In this formula the symbols represent volumetric percentages of the flue gas constituents as determined by analysis.

The amount of excess air in its relation to the percentage of CO_2 is shown by the curves in Fig. 1 for several fuels. These are approximate values. It should be noted that in hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

TABLE 4. REPRESENTATIVE MAXIMUM CO_2 VALUE

FUEL	THEORETICAL CO_2	CO_2 USUALLY ATTAINED IN PRACTICE
Coke	21.00	12-14
Anthracite	20.20	12-14
Bituminous Coal	18.20	13
No. 2 Fuel Oil	15.00	10.5
No. 6 Fuel Oil	16.50	13.5
Natural Gas	12.00	9.7
Coke Oven Gas	11.00	8.5

Due to the different carbon-hydrogen ratios of the different fuels the maximum CO_2 attainable varies. Representative values for perfect combustion of several fuels are given in Table 4.

In considering the factor of excess air it should be noted that a deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack. An excess of air is always required, however, to eliminate combustible losses occasioned by poor mixing of the fuel and air. It is considered good practice, under usual operating conditions, to supply from 25 to 50 per cent excess air, dependent upon the fuel utilized.

SECONDARY AIR

When bituminous coal is hand-fired in a furnace the volatile matter in the fuel distills off 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 per cent 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. 2 [1]. 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 admitted through the firing door in an attempt to burn carbon monoxide and the volatile combustible distilled from the freshly fired fuel.

The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a

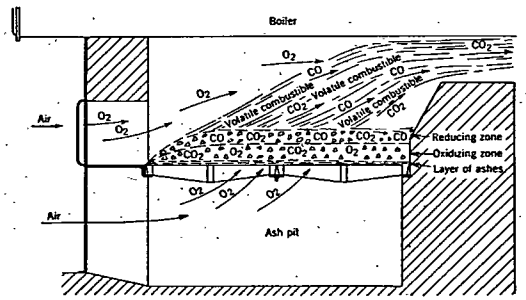


FIG 2. COMBUSTION OF FUEL IN A HAND-FIRED FURNACE

number of factors which include size and type of fuel, depth of fuel bed, and size of fire-pot.

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, which draws in more secondary air 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.

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the effect of slot opening is dependent on whether the ashpit damper is open or closed.

Bituminous coals require a large amount of secondary air during the period after firing in order to burn the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial.

Secondary air that enters the combustion chamber too far removed from the zone of combustion will also be harmful, for the oxygen in the

secondary air will not react with any unburned gases unless the mixture is subjected to high temperatures.

The air requirements of oil and gas burners are discussed in Chapter 17, Automatic Fuel Burning Equipment.

DRAFT REQUIREMENTS

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.
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.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning. If the excess is caused by holes in the fuel bed, or an extremely thin fuel bed, it is often possible to produce a higher rate of burning by increasing the thickness of the bed. The thickness of the fuel bed should not, however, be increased too much because the increased draft resistance will reduce the rate of primary air supply and the rate of burning.

For amount of draft required see Chapter 19, Chimneys and Draft Calculations.

DRAFT REGULATION

Because of the varying heating load demands present 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. Correct and incorrect methods of draft regulation are shown in Fig. 3. The air enters through the ashpit draft door, firing door and by leaks in the setting, whereas the gases leave only through the uptake. By throttling the gases with the damper in the

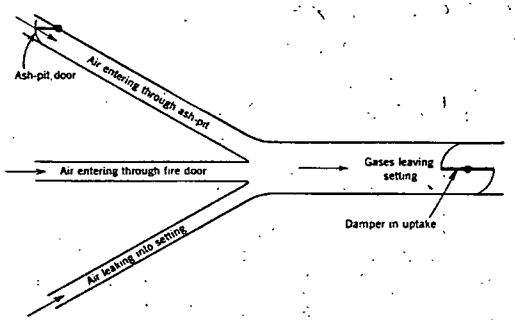


FIG. 3. CORRECT AND INCORRECT METHODS OF DRAFT REGULATION IN A HAND-FIRED FURNACE

uptake all the air entering by each of the three intakes is reduced in the same proportion. If the ashpit draft door is closed, the air admitted through the ashpit is reduced, while it is increased through the other two intake openings.

Methods of control of draft conditions when burning oil or gas are noted in Chapter 17, Automatic Fuel Burning Equipment.

CLASSIFICATION OF COALS

The complex composition of coal makes it difficult to classify it into clear-cut types. Its chemical composition is some indication but coals having the same chemical analysis may have distinctly different burning characteristics. Users are mainly interested in the available heat per

TABLE 5. 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 per cent or more (Dry V.M., 2 per cent or less)	Non-agglomerating[b]
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (Dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semi-anthracite	Dry F.C., 86 per cent or more and less than 92 per cent (Dry V.M., 14 per cent or less and more than 8 per cent)	
II. Bituminous[d]	1. Low volatile bituminous coal	Dry F.C., 78 per cent or more and less than 86 per cent (Dry V.M., 22 per cent or less and more than 14 per cent)	Either agglomerating[b] or non-weathering[f]
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 78 per cent (Dry V.M., 31 per cent or less and more than 22 per cent)	
	3. High volatile *A* bituminous coal	Dry F.C., less than 69 per cent (Dry V.M., more than 31 per cent); 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

[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 per cent 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 per cent 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.

Adapted from *A.S.T.M.* Standards, 1937, Supplement, p. 145, *American Society for Testing Materials*, Philadelphia.

pound of coal, in the handling and storing properties, and in the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space: a treatment applicable to heating boilers is given in a Bureau of Mines Bulletin [2].

Coal composition may be expressed by either an *ultimate* or *proximate* analysis. In the ultimate analysis the proportions of carbon, hydrogen, oxygen, nitrogen, sulphur, and ash are determined. This form of analysis is difficult to make and is used only for extremely close studies. The proximate analysis is more easily made and is satisfactory for most purposes. In this analysis, the proportions of moisture, volatile matter, fixed carbon, and ash are determined. Moisture is obtained by noting the loss of weight of a sample of coal when dried at about 220 F. To determine volatile matter, the dried sample is heated to about 1750 F in a closed crucible, and the loss of weight is noted. The 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 sulphur content is frequently reported with a proximate analysis.

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 per cent 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, so it is 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 its constituents.

A classification of coals is given in Table 5, 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. It is capable of giving a high efficiency in the common types of hand-fired furnaces. A tabulation of the quality of the various anthracite sizes will be found in a Bureau of Mines Report [3]. Standard anthracite sizing specifications are shown in Table 6.

Semi-anthracite has a higher volatile content than anthracite. It is not so hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

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 more abundant bituminous coals it can be burned with less production of smoke, and it is sometimes called *smokeless coal*.

The term *bituminous coal* covers 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 completely melt, 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 of 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 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, the temperatures and time of distillation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil.

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke*, which covers the greater part of the coke sold, and *gas-house coke*. The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at low coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than those of the various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others briquetted.

Petroleum cokes, which are obtained by coking the residue left from the distillation of petroleum, vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

TABLE 6. STANDARD ANTHRACITE SIZING SPECIFICATIONS[a]

Size	Test Mesh, In.		Round Mesh			Maximum Impurities, Per Cent	
			Oversize	Undersize			
	Through	Over	Max. Per Cent	Max. Per Cent	Min. Per Cent	Slate[b]	Bone
Broken	4 3/8	3 1/4	.....	15	7 1/2	1 1/2	2
Egg	3 1/4 to 3	2 1/16	5	15	7 1/2	1 1/2	2
Stove	2 1/16	1 5/8	7 1/2	12 1/2	7 1/2	2	3
Nut	1 5/8	13/16	7 1/2	10	5	3	4
Pea	13/16	9/16	10	15	7 1/2	4	5
Buckwheat	9/16	5/16	10	15	7 1/2	12 Ash	
Rice	5/16	3/16	10	15	7 1/2	13 Ash	
Barley	3/16	3/32	10	20	10	.....	

[a] Approved and adopted by *Anthracite Committee*, State Street Building, Harrisburg, Pa.

[b] When slate content on Broken to Pea inclusive is less than above standards, bone content may be correspondingly increased, but slate content specified above shall not be exceeded in any event and the total maximum impurities shall not exceed those above specified.

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 size coal is the proper size of anthracite for many boilers and furnaces used for heating buildings. It burns well on grates at least 16 in. in diameter and 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, with 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 fire-box.

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 on account of the danger of the fuel 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 within the fire-pot, 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 exercised in manipulation of the grates this fuel can be burned satisfactorily without the aid of any special equipment.

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 the clinker in an easily broken up condition so that it readily can be shaken through the grate.

Forced draft and small mesh grates are frequently used for burning buckwheat anthracite. For greater convenience, domestic stokers are 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[1]. 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 which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the fire-box. 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 impractical. The adherence to these methods insofar as practical, however, will result in better combustion.

The output obtained from any heater with bituminous coal will usually exceed that obtained with anthracite, since bituminous coal burns more rapidly than anthracite and with less draft. Bituminous coal, however, will usually require frequent attention to the fuel bed.

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. Especial care must be taken in hand-firing bituminous coals.

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner*

fuel beds on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher A.P.I. gravity oil always produces less smoke than a high volatile coal or low A.P.I. gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature often produces satisfactory results.

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 a considerable investment in special apparatus is often necessary.

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 ground area 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 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 to the center of the bed forming 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, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring, and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

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

TABLE 7. DETAILED REQUIREMENTS FOR FUEL OILS[a]

GRADE[b]	FLASH POINT F DEG		POUR POINT F DEG	WATER AND SEDIMENT PER CENT	CARBON RESIDUE PER CENT	ASH PER CENT	DISTILLATION TEMPERATURES F DEG			VISCOSITY SECONDS			
							10 PER CENT POINT[c]	90 PER CENT POINT[c]	END POINT	SAYBOLT UNIVERSAL AT 100 F		SAYBOLT FUROL AT 122 F	
	MIN.	MAX.	MAX.	MAX.	MAX.	MAX.	MAX.	MAX. MIN.	MAX.	MAX.	MIN.	MAX.	MIN.
No. 1 Fuel oil—a distillate oil for use in burners requiring a volatile fuel.	100 or Legal	165	0[d]	Trace	0.05 on 10% Residuum[e]		410		560[f]				
No. 2 Fuel oil—a distillate oil for use in burners requiring a moderately volatile fuel.	110 or Legal	190	10[d]	0.05	0.25 on 10% Residuum[e]		440	600					
No. 3 Fuel oil—a distillate oil for use in burners requiring a low viscosity fuel.	110 or Legal	230	20[d]	0.10	0.15 Straight			675	600[g]	45			
No. 5 Fuel oil—an oil for use in burners requiring a medium viscosity fuel.	130 or Legal			1.00		0.10					50	40	
No. 6 Fuel oil—an oil for use in burners equipped with preheaters permitting a high viscosity fuel.	150			2.00[h]								300	45

[a] Recognizing the necessity for low sulphur fuel oils used in connection with heat-treatment, non-ferrous metal, glass and ceramic furnaces and other special uses, a sulphur requirement may be specified in accordance with the following table:

GRADE OF FUEL OIL	SULPHUR, MAX. PER CENT
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No Limit
No. 6	No Limit

Other sulphur 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 conditions.

[d] For use in other than sleeve type blue flame burners carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

[e] The maximum end point may be increased to 690 F when used in burners other than sleeve type blue flame burners.

[f] To meet certain burner requirements the carbon residue limit may be reduced to 0.15 per cent on 10 per cent residuum.

[g] The minimum distillation temperature of 600 F for 90 per cent may be waived if A.P.I. gravity is 26 or lower.

[h] Water by distillation, plus sediment by extraction. Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

TABLE 8. APPROXIMATE GRAVITY AND CALORIFIC VALUE OF STANDARD GRADES OF FUEL OIL

COMMERCIAL STANDARD NO.	APPROXIMATE GRAVITY RANGE A. P. I.	CALORIFIC VALUE BTU PER GALLON
1	38-40	136,000
2	34-36	138,500
3	28-32	141,000
5	18-22	148,500
6	14-16	152,000

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 be either mixed with fine sizes or broken up before using.

FURNACE VOLUME

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and correctly proportioned combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of 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 OILS

The Commercial Standard Specifications for Fuel Oils (CS 12-40) of the *U. S. Department of Commerce* are given in Table 7. These specifications conform with *American Society for Testing Materials* Tentative Specifications for Fuel Oils D396-38T.

The A.P.I. gravity of oil is of interest in its relationship to the calorific value and these data are given in Table 8.

COMBUSTION OF OIL

With oil, as with any kind of fuel, efficient heat production requires that all combustible matter in the fuel shall be completely consumed and that it shall be done with a minimum of excess air. The combustion of oil is a rather rapid chemical reaction. Excess air provides an over supply of oxygen so that all of the oil will be completely oxidized and thus produce all the heat possible. The use of unreasonable quantities of air in excess of theoretical combustion requirements results in lowered efficiencies due to increased stack losses. Such losses, if not accompanied by unburned products of combustion, may be offset somewhat by increasing the secondary heating surfaces of the heat absorbing medium, boiler or furnace. Oil is a highly concentrated fuel composed mainly of

TABLE 9. REPRESENTATIVE PROPERTIES OF GASEOUS FUELS, BASED ON GAS AT 60 F AND 30 IN. HG.

GAS	BTU PER CU FT		SPECIFIC GRAVITY, AIR = 1.00	AIR REQUIRED FOR COMBUSTION, (CU FT)	PRODUCTS OF COMBUSTION				THEORETICAL FLAME TEMPERATURE, (F DEG)
					Cubic Feet			ULTIMATE CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—California	1200	1085	0.67	11.26	1.24	2.24	12.4	12.2	3610
Natural gas—Mid-Continental	970	870	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1.17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1130	1025	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	570	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	590	520	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carbureted water gas	540	495	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	300	280	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	135	125	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

hydrogen and carbon. In its liquid form oil cannot burn. It must be converted into a gas or vapor by some means. If the excess air is to be kept within efficient limits, air must be supplied in carefully regulated quantities. The air and oil vapor must be thoroughly mixed to get a rapid and complete chemical reaction. The better the mixing, the less excess air will be needed. The combustion must take place in a space that maintains the temperatures high so the reaction will be completed.

CLASSIFICATION OF GAS

Gas is broadly classified as being either *natural* or *manufactured*. Natural gas is a mechanical mixture of several combustible and inert gases rather than a chemical compound. Manufactured gas as distributed is usually a combination of certain proportions of gases produced by two or more processes. Representative properties of gaseous fuels commonly used in domestic heating are presented in Table 9.

Natural gas is the richest of the gases and contains from 80 to 95 per cent methane, with small percentages of the other combustible hydrocarbons. In addition, it contains from 0.5 to 5.0 per cent of CO_2, and from 1 to 12 or 14 per cent of nitrogen. The heat value varies from 1000 to 1200 Btu per cubic foot, the majority of natural gases averaging about 1000 Btu per cubic foot. Table 9 shows typical values for the four main oil fields, although values from any one field vary materially.

Table 9 also gives the calorific values of the more common types of manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. However, in any community the variations in gas composition are held within suitable limits so that the performance of approved gas appliances will not be adversely affected.

COMBUSTION OF GAS

The majority of gas burners utilized in central domestic heating plants are of the Bunsen type and operate with a non-luminous flame. In this type of burner part of the air required for combustion is mixed with the gas as primary air, the air and gas mixture being fed to the burner ports. Additional secondary air is introduced around the flame by draft inspiration. In the luminous flame burner, which is sometimes used, all of the air for combustion is brought in contact with the flame as secondary air. This secondary air should be brought into intimate contact with the gas.

Some makes of burners use radiants or refractories to convert some of the energy in the gas to radiant heat. The radiants also serve as baffles in directing the flow of the products of combustion.

The quantity of air given in Table 9 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. In order to insure freedom from carbon monoxide under conditions which may obtain in installations, it is customary to design gas burning appliances for a supply of 30 to 35 per cent of excess air. In individual installations in which flue gas analyses are made, the excess air is sometimes reduced to approximately 20 per cent. The division of the air into primary and secondary is a matter of burner design, the pressure of gas available, and the type of flame desired.

The air gas ratio has a decided effect upon flame propagation. It is necessary that the gas will flow out of the burner ports fast enough so that the flame cannot travel back into the burner head, i.e. *flash back*, but the velocity must not be so high that it blows the flame away from the port.

The maximum and minimum flow speeds from burner ports which may be permitted are known to be very close together when air-gas mixtures in theoretical proportions are being supplied to the burner. As the air-gas ratio is lowered, and the mixture becomes more *gas rich*, the limiting speeds become farther apart, until with 100 per cent gas, in an all-yellow flame, flash back cannot occur and a much higher velocity is needed to blow off the flames.

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[4] shows that the loss of seasonal efficiency is not so great as has been believed and usually is not over 6 per cent because the greater part of the heat is transmitted through the combustion chamber surfaces. The *Bureau of Standards Report* BMS 54[5] 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.

CONDENSATION AND CORROSION

Sulphur dioxide or trioxide formed by the combustion of fuels is the corroding element in flue gases and becomes active whenever moisture is available for the formation of sulphurous or sulphuric acid. It is therefore necessary to maintain a flue gas temperature above 175 F in all parts of appliances, and consequently it is not practicable to recover the latent heat in flue gases. Since some unpreventable condensation occurs during the warming-up period, it is important to design appliances so that all surfaces reach quickly a temperature above the dew-point of the flue gases or have corrosion resistant properties enabling them to withstand the corrosive effect.

DUSTLESS TREATMENT OF COAL

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with various petroleum products, a solution of calcium chloride or a mixture of calcium and magnesium chlorides.

The coal is usually treated at the mine, but sometimes by the local distributor just before delivery. The salt solutions are sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size. Oil for the dustless treatment of coal is also applied under high pressure, in concentrations of 1 to 8 qt per ton of coal, depending upon the characteristics of the coal and oil.

Dustless treatments which are of such a corrosive nature that they may damage coal handling or burning equipment should not be used.

REFERENCES

1—Hand Firing Soft Coal Under Power Plant Boilers (*U. S. Bureau of Mines Technical Paper* No. 80).

2—Five Hundred Tests of Various Coals in Housheating Boilers (*U. S. Bureau of Mines Bulletin* No. 276).

3—Quality of Anthracite as Prepared at Breakers, 1935 (*U. S. Bureau of Mines Report of Investigations*, R. I. 3283).

4—Effect of Soot on Heat Transmission in Boilers (*U. S. Bureau of Mines Report of Investigation* No. 3272).

5—Effect of Soot on the Rating of an Oil-Fired Heating Boiler (*National Bureau of Standards Report* BMS 54).

BIBLIOGRAPHY

Fuels and Their Combustion, by Haslam and Russell (McGraw-Hill Co., 1926).

Principles of Combustion in the Steam Boiler Furnace, by Arthur D. Pratt (Babcock and Wilcox Co.).

Smoke-Producing Tendencies in Coals of Various Ranks, by H. J. Rose and F. P. Lasseter (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 329).

Fundamentals of Combustion in Small Stokers, by C. A. Barnes (*Bituminous Coal Research, Inc., Technical Report* No. IV).

Combustion Efficiencies as Related to Performance of Domestic Heating Plants, by A. P. Kratz, S. Konzo, and D. W. Thomson (*Illinois Engineering Experiment Station Circular* No. 44).

Hand-Firing of Bituminous Coal in the Home, by A. P. Kratz, J. R. Fellows, and J. C. Miles (*Illinois Engineering Experiment Station Circular* No. 46).

Classification and Selection of Illinois Coals, by G. H. Cady (*Illinois State Geological Survey Bulletin* No. 62).

Bituminous Coal Research, Inc.:

Technical Report VI, The Treatment of Coal with Oil and Other Petroleum Products, by J. M. Pilcher and R. A. Sherman.

Information Bulletin No. 4, Dustless Treatment of Coals with Materials Other than Oil, by R. A. Sherman and G. W. Land.

Information Bulletin No. 6, Questions and Answers on the Use of Fuel Oil for Dustless Treatment.

Technical Report VII—Application of Overfire Air Jets, by R. B. Engdahl.

Bureau of Mines Publications:

Bulletin No. 97, Sampling and Analyzing Flue Gases, by Henry Kreisinger and F. K. Ovitz.

Report of Investigations (R. I. 2980), Coke as a Domestic Heating Fuel, by P. Nicholls and B. A. Landry.

Technical Paper No. 303, Value of Coke, Anthracite, and Bituminous Coal for Generating Steam in a Low-pressure Cast-Iron Boiler, by John Blizard, James Neil, and F. C. Houghten.

Bulletin 378, Effect of Preheat, and Distribution of Ash in Fuel Beds, by P. Nicholls.

Bulletin 360, Removal of Soot from Furnaces and Flues by the Use of Salts and Compounds, by P. Nicholls.

Handbook, Questions and Answers for the Home Fireman (Revised), by J. F. Barkley.

Anthracite Industries Laboratories Publications:

Report 2015, Comparison of Sizes, Egg, Stove and Chestnut Anthracite.

Report 2018, Domestic Survey.

Report 2062, Utilization of Anthracite for Domestic Heating.

Report 2204, The Crater Method of Firing.

Report 2403, Anthracite Industries Manual.

Oil and Gas Publications:

Oil Fuels and Burners, by James A. Moyer (McGraw-Hill Co., 1937).

Handbook of Oil Burning, by Harry F. Tapp (*American Oil Burner Association*).

Industrial Gas Series, Combustion (*American Gas Association*).

Comfort Heating (*American Gas Association*).

CHAPTER 17

Automatic Fuel Burning Equipment

Classification of Stokers, Combustion Process and Adjustments, Furnace Design, Classification of Oil Burners, Combustion Chamber Design, Classification of Gas-Fired Appliances, Combustion Process, Ratings, Fuel Burning Rates

AUTOMATIC mechanical equipment for the combustion of solid, liquid, and gaseous fuels is considered in this chapter.

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.

CLASSIFICATION OF STOKERS ACCORDING TO CAPACITY

Stokers may be classified according to their coal feeding rates. The following classification has been made by the *U. S. Department of Commerce*, in cooperation with the *Stoker Manufacturers Association*.

Class 1. Capacity under 61 lb of coal per hour.
Class 2. Capacity 61 to 100 lb of coal per hour.
Class 3. Capacity 101 to 300 lb of coal per hour.
Class 4. Capacity 300 to 1200 lb of coal per hour.
Class 5. Capacity 1200 lb of coal per hour and over.

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 which 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 Figs. 2 and 3. Some stokers, particularly those designed for use with anthracite, automatically remove ash from the ash pit and deposit it in an ash receptacle as shown in Fig. 3. Most of the bituminous models, however, require 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.

Stoker-Fired Boiler and Furnace Units

Boilers, air conditioners, and space heaters especially designed for stokers are available having design features closely coordinating the heat absorber and the stoker. Although efficient and satisfactory performance can be obtained from the application of stokers to existing boilers and furnaces, some of the combination stoker-fired units (Fig. 4) are more compact and attractive in appearance.

Class 2 and 3 Stokers

Stokers in this class are usually of the screw feed type without auxiliary plungers or other means of distributing the coal. They are used extensively for heating plants in apartments and hotels, also, for industrial

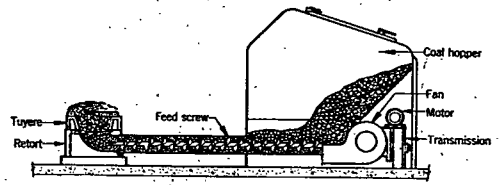


FIG. 1. UNDERFEED STOKER, HOPPER TYPE, CLASS 1

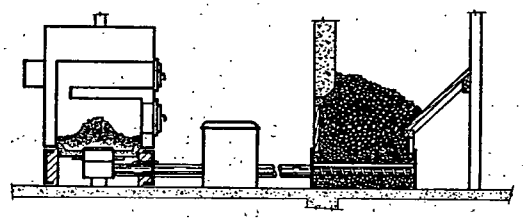


FIG. 2. UNDERFEED STOKER, BIN FEED TYPE, CLASS 1

plants. They are of the underfeed type and are available in both the hopper type, as illustrated in Fig. 5, and the bin feed type, shown in Fig. 6. These units also are built in plunger feed type with an electric motor or a steam or hydraulic cylinder coal feed drive.

Stokers in this class are available for burning all types of anthracite, bituminous and lignite coals. The tuyere and retort design varies according to the fuel and load conditions. Stationary type grates are used on bituminous models and the clinkers formed from the ash accumulate on the grates surrounding the retort.

Anthracite stokers in this class are equipped with moving grates which discharge the ash into a pit below the grate. This ash pit may be located on one or both sides of the grate and on some installations is of sufficient capacity to hold the ash for several weeks' operation.

Class 4 Stokers

Stokers in this group vary widely in details of design and several methods of feeding coal are employed. The underfeed stoker is widely used, although a number of the overfeed types are used in the larger sizes. Bin-feed, as well as hopper models, are available in both underfeed and overfeed types.

Class 5 Stokers

The prevalent stokers in this field are: (1) underfeed side cleaning, (2) underfeed rear cleaning, (3) overfeed flat grate, and (4) overfeed inclined grate.

Underfeed side cleaning stokers are made in sizes up to approximately 500 boiler horsepower. 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. 7.

The rear cleaning underfeed stoker is usually of the multiple retort design and 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. A typical traveling-grate stoker is illustrated in Fig. 8.

Another distinct type of overfeed flat-grate stoker is the spreader (Figs. 9 and 10) type in which coal is distributed either by rotating paddles or by air over the entire grate surface. 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 ash pit or receptacle from which it may be removed either manually or automatically.

Larger underfeed anthracite stokers operate on the same principle, except that the retort is rectangular and the refuse spills over only one or two sides of the grate. Anthracite for stoker firing is usually the No. 1 buckwheat or No. 2 buckwheat size.

Because the majority of the smaller 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 non-combustible. 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

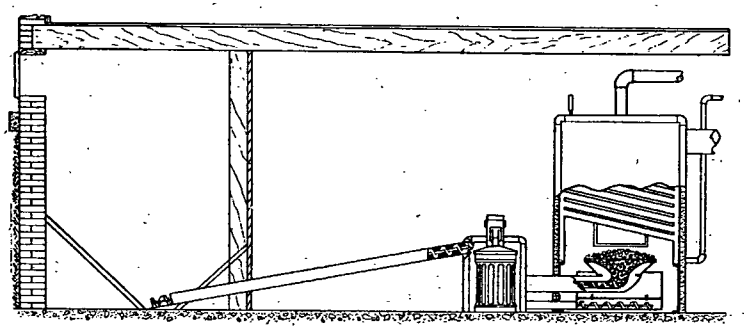


FIG. 3. UNDERFEED ANTHRACITE STOKER WITH AUTOMATIC ASH REMOVAL, BIN TYPE

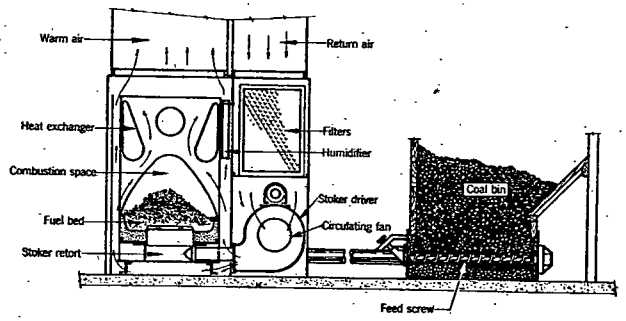


FIG. 4. STOKER-FIRED WINTER AIR CONDITIONING UNIT

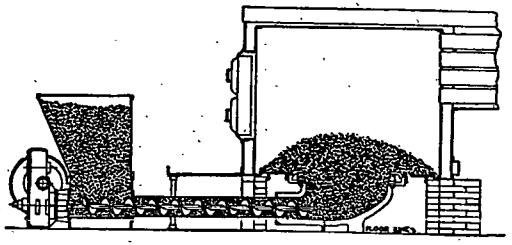


FIG. 5. UNDERFEED SCREW STOKER, HOPPER TYPE, CLASS 2, 3 OR 4

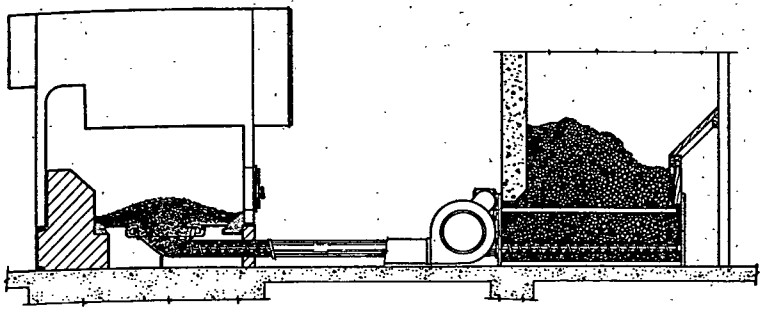


FIG. 6. UNDERFEED SCREW STOKER, BIN TYPE, CLASS 2, 3 OR 4

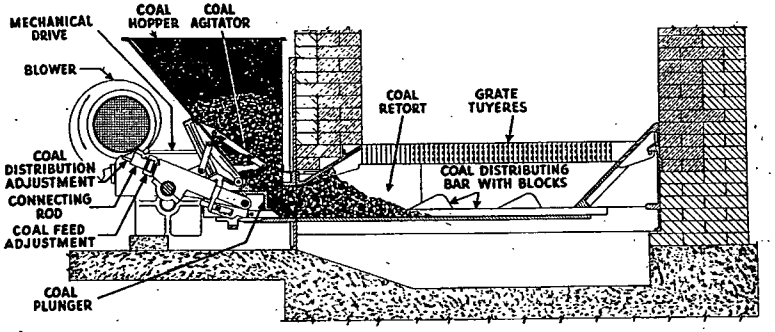


FIG. 7. UNDERFEED SIDE CLEANING STOKER

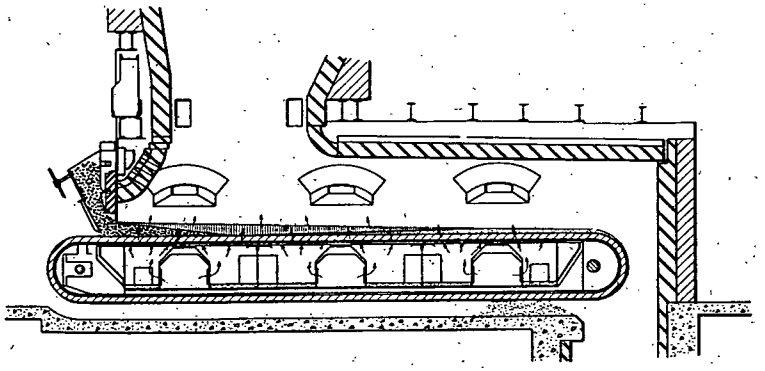


FIG. 8. OVERFEED TRAVELING-GRATE STOKER

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, 3 and 4 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, 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.

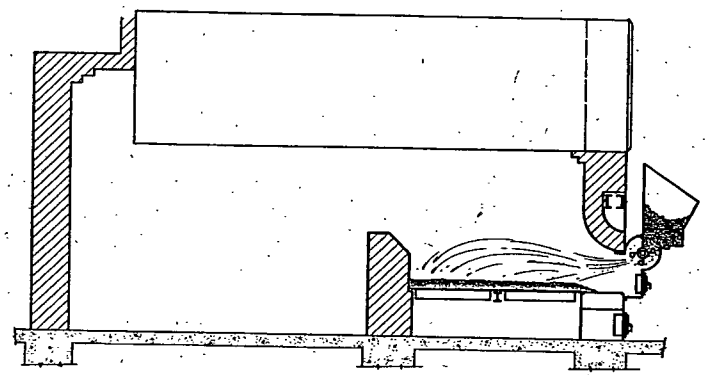


FIG. 9. OVERFEED SPREADER STOKER (ROTOR TYPE)

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 ash pit 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. Frequently, a rear combustion arch is also required.

In addition to the use of rocking grates, the overfeed inclined-grate stoker is provided with an ash plate 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 for 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 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

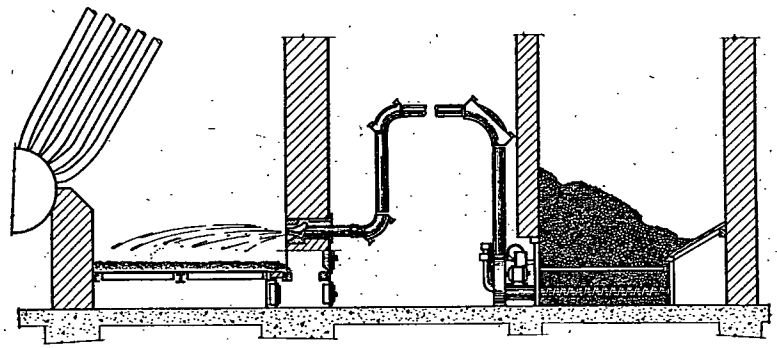


FIG. 10. OVERFEED SPREADER STOKER (PNEUMATIC TYPE)

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 fuel ratio.

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. The efficiency of combustion may be determined by analyzing the flue gases as explained in Chapter 16.

Furnace Design

Although there is considerable variation in stoker, boiler, and furnace design, the stoker industry, from long-time experience, has established certain rules for the proportioning of furnaces for domestic, and commercial stokers. The stoker installer and designer of stoker-fired equipment should give careful consideration to these factors.

The *Stoker Manufacturers Association* has published standard recommendations on setting heights for stokers having capacities up to 1200 lb of coal per hour[2].

The empirical formulas for determining these setting heights are:

For burning rates up to 100 lb coal per hour

$$H = 0.1125\,B + 15.75 \quad (1)$$

For burning rates from 100 to 1200 lb coal per hour

$$H = 0.03\,B + 24 \quad (2)$$

where

H = minimum setting height, inches, for steel boilers. For cast-iron boilers height may be $\frac{7}{8}\,H$.

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. 11[2].

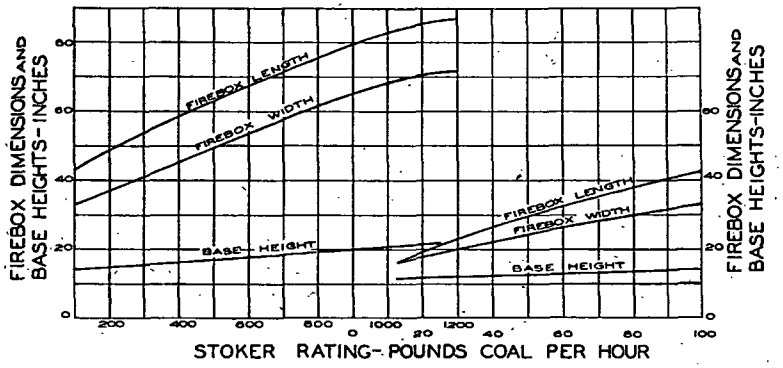


FIG. 11. SUGGESTED MINIMUM FIREBOX DIMENSIONS AND BASE HEIGHTS[a]

[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 12,000 Btu per pound. Under certain conditions smaller fireboxes will permit satisfactory performance but these dimensions are preferred normal minimums.

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 rests with the experience of the installer, or designer, with a particular stoker, the type of fuel used, and the construction of the boiler or furnace.

Rating and Sizing Stokers

The capacity or rating of small underfeed stokers is usually stated as the burning rate in pounds of coal per hour. Codes for establishing uniform methods of rating anthracite, and bituminous coal stokers have been adopted by the *Stoker Manufacturers Association*[3].

The Association also has adopted a uniform method of selecting stokers that is published in convenient tables and charts[2]. The required capacity of the stoker is calculated as follows:

$$\frac{\text{Load (Btu per hour)}}{\text{Heating value of coal (Btu per pound)} \times \text{over-all efficiency of stoker and boiler or furnace}} = \text{Stoker burning rate required (pounds of coal per hour)}$$

In determining the total load placed on a stoker-fired boiler by a steam or hot water heating system, a piping and pick-up 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 heat delivery from the stoker of the smallest household type to the largest industrial unit can be regulated accurately with fully automatic controls. The smaller heating applications are controlled normally by a

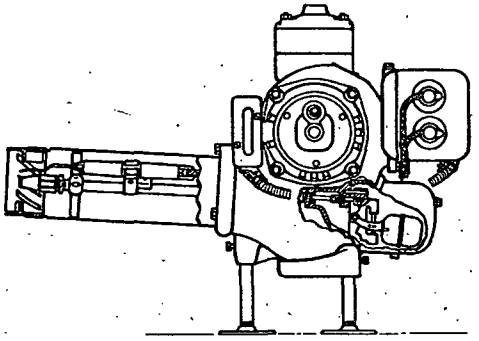


FIG. 12. LOW PRESSURE ATOMIZING OIL BURNER

thermostat placed in the building to be heated. Limit controls are supplied to prevent excessive temperature or pressure being developed in the furnace or boiler and refueling controls are used to maintain ignition during periods of low heat demand. Automatic low water cut-outs are recommended for use with all automatically-fired steam boilers. (See Chapter 34.)

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically from liquid fuels. Two methods are employed for the preparation of the oil for the combustion process; atomization, and vaporization. The simpler types of burners depend upon the natural chimney draft for supplying the air for combustion. Other burners provide mechanical air supply or a combination of atmospheric, and mechanical. Ignition is accomplished by an electrical spark or hot wire, or by an oil or gas pilot. Some burners utilize a combination of these methods. Continuously operating burners may use manual ignition. Burners of different types

operate with luminous or non-luminous flame. Operation may be intermittent, continuous with high-low flame, or continuous with graduated flame.

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)

Gun type burners may be divided into two classes, low-pressure, and high-pressure atomization. In the first group, a mixture of oil and primary air is pumped as a spray through the nozzle at a pressure of 2 to 7 lb per square inch. Secondary air is supplied by a fan. Ignition is obtained by means of a high-voltage electric spark used alone, or as primary ignition for a gas pilot. Various features of a low pressure atomizing burner are shown in Fig. 12.

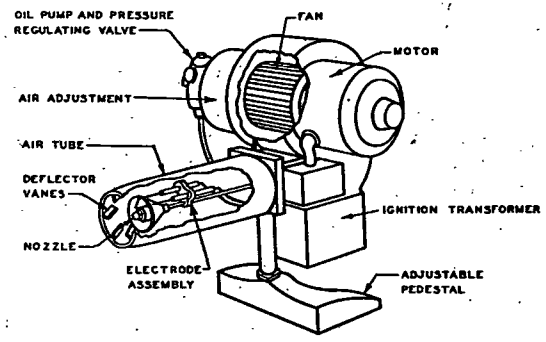


FIG. 13. HIGH-PRESSURE ATOMIZING OIL BURNER

The high-pressure atomizing type, illustrated in Fig. 13, is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and arranged so that a spray of atomized oil is introduced, at about 100 lb per square inch, and mixed in the combustion chamber with the air stream emerging from the air tube. A variety of patented shapes is employed at the end of the air tube to influence the direction and speed of the air and thus the effectiveness of the mixing process.

This type of burner utilizes a fan to supply the air for combustion, and ignition is established by a high-voltage electric spark that may be operative continuously while the burner is running, or just at the beginning of the running period. Gun type burners operate 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. Pressure-atomizing burners generally use the distillate oils, No. 1, 2 or 3 grade. (See Chapter 16.)

Rotary Type

This class of burners may be divided into two groups: vertical, and horizontal. Most of the smaller rotary burners are of the vertical type, and use the lighter distillate oils, No. 1 or 2 grade.

The most distinguishing feature of vertical rotary burners is the principle of flame application. These burners are of two general types: the center flame and wall flame. In the former type (Fig. 14), the oil is atomized by being thrown from the rim of a revolving disc or cup and the flame burns in suspension with a characteristic yellow color. Combustion is supported by means of a bowl-shaped chamber or hearth. The wall flame burner (Fig. 15) differs in that combustion takes place in a ring of stainless steel or refractory material, which is placed around the hearth. Dependent upon combustion adjustment, these burners may operate with either a semi-luminous or non-luminous flame.

Both types of vertical rotary burners are further characterized by their installation within the ash pit of the boiler or furnace. Various types of

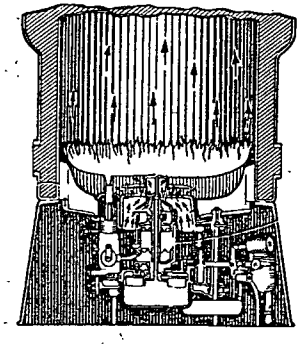


FIG. 14. CENTER FLAME VERTICAL ROTARY BURNER

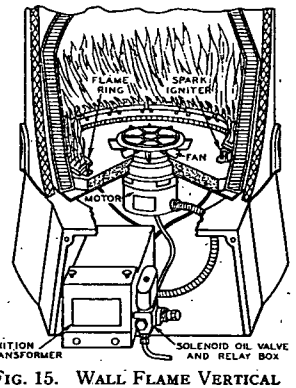


FIG. 15. WALL FLAME VERTICAL ROTARY BURNER

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.

Horizontal rotary burners are used principally to burn the heavier oils, Nos. 5 and 6 grades, principally in larger commercial and industrial installations, although domestic sizes are available. Such burners are of the mechanical atomizing type, using rotating cups which throw the oil from the edge of the cup at high velocity into the surrounding stream of air delivered by the blower (Fig. 16).

Horizontal rotary burners commonly use a combination electric-gas ignition system, or are lighted manually. Primary air for combustion is supplied by a blower, and secondary air, often introduced through a checkerwork in the combustion chamber, is controlled by chimney draft. These burners operate with a luminous flame, usually on high-low or continuous setting.

In larger installations, burners may be installed in multiple in a common combustion chamber. Because of the high viscosity oils used in these

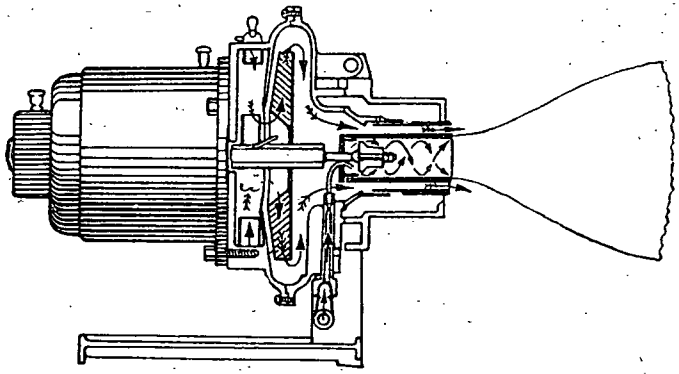


FIG. 16. HORIZONTAL ROTATING CUP OIL BURNER

burners, it is customary to preheat the oil between the tank and the burner. Preheating when delivering from tank car, or truck is often required in cold weather. As with pressure atomizing burners, a refractory combustion chamber, built within the boiler furnace, is a part of the installation.

Vaporizing Burners

This type burner transforms the oil into a combustible vapor by the application of heat from a plate, or cracking chamber. The class may be subdivided into burners supplying combustion air mechanically, and those utilizing the natural chimney draft. Vaporizing pot-type burners are restricted to the use of the lighter distillates.

These burners are designed for continuous or intermittent operation, with manual ignition, and the rate of burning is controlled by a metering valve. Flame may be luminous or non-luminous, dependent upon adjustment. A burner of this type is illustrated in Fig. 17.

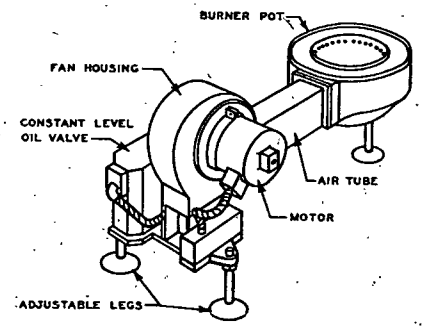


FIG. 17. VAPORIZING POT-TYPE BURNER

Vaporizing burners of the natural draft type are used as the firing device in integral space heaters, water heaters, or furnace units. Only those with full mechanical draft are suitable for conversion installation.

Oil-Fired Boiler and Furnace Units

A number of types of specially designed oil-fired boiler-burner and furnace-burner units are available. A typical boiler-burner unit is shown in Fig. 18. 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, 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.

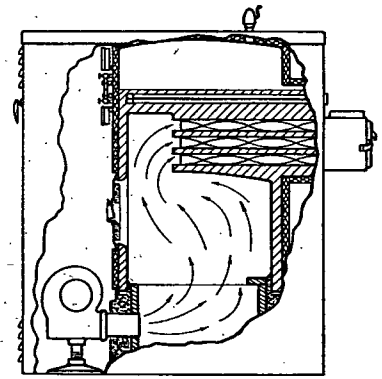


FIG. 18. TYPICAL BOILER-BURNER UNIT

Operating Requirements for Mechanical Draft Oil Burners

The *U. S. Department of Commerce* in conjunction with the oil burner industry has established commercial standards for automatic mechanical draft oil burners for domestic installations which cover installation requirements and performance tests[4].

Combustion Process

Efficient combustion must produce a clean flame and use a relatively small excess of air, *i.e.*, between 25 and 50 per cent. 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. This is accomplished before the oil vapor mixes with air to any extent and if the air and oil vapor temperatures are high and the fire pot hot, a clear blue flame is produced.

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 proceeds quickly. 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.

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 an appreciable period following the initial adjustments of oil and air. Eventually certain changes may occur, however, that will cause the per cent of excess air to 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 by-pass 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.

Air leakage into the boiler or furnace setting should be reduced to a minimum. The amount of air leakage will be determined by the draft in the combustion chamber. It is important that this draft should be reduced as low as is consistent with the proper disposal of the gases 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 very helpful in maintaining such values.

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 that the draft will be uniform within the limits given above 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 usually sufficient to analyze only for carbon dioxide (CO_2). A showing of 10 to 12 per cent indicates

the best adjustment if the flame is clean. Most of the good installations show from 8 to 10 per cent CO_2. Taking into account the potential hazard of low excess air (high CO_2), a setting to give 10 per cent CO_2 constitutes a reasonable standard for most 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 shall be as nearly air tight 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 refractory or firebrick 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. In this way as much firebrick as possible is close to the flame so it may be kept hot. 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, as in this class successful performance depends upon this factor.

Boiler Settings

As 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 ash pit 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 Btu per cubic foot per hour. 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. The essential in combustion chamber design is to provide against 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

Controls for oil burner operation, including devices for the safety and protection of a boiler or furnace, are fully described in Chapter 34.

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, yellow flame, or power burner types.

The use of gas has resulted in the production of a number of types of domestic gas heating appliances, and systems. These may be classified

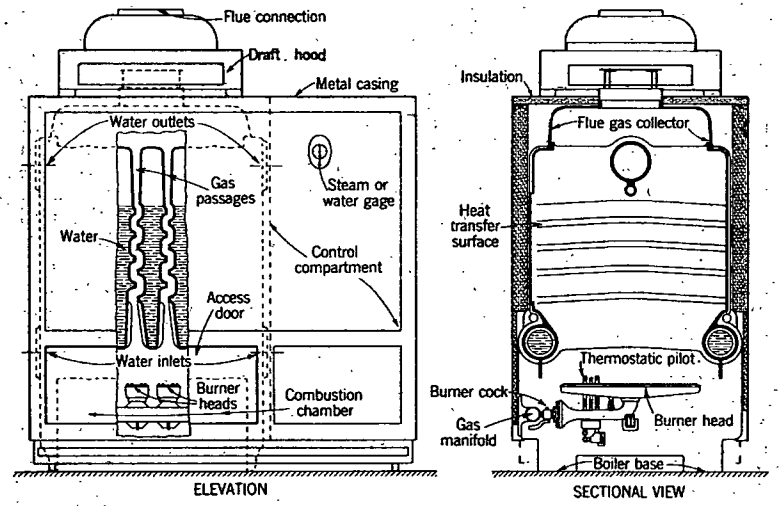


FIG. 19. GAS-FIRED BOILER

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. Unit heaters, space heaters and circulators may be had for installation 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, thermostatic 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. 19.

Conversion burners are usually complete burner and control units designed for installation in existing boilers and furnaces. Burner heads

are of circular or rectangular shape in order to fit in space available. The control equipment is generally the same as for gas boilers and furnaces. Various baffles made of clay radiants or metal are used for the purpose of guiding the products of combustion along the heating surface in the firebox or flues. Automatic air dampers are supplied on many models to prevent flow of air into the firebox when the burner is not operating. A typical gas conversion burner is shown in Fig. 20.

Burners of this type are available in sizes ranging from 80,000 to 500,000 Btu per hour capacity. Burners of even larger capacity, for use with natural gas in large steel boilers are usually engineered by the local utility or contractor. They are available in an infinite number of sizes because the burner may be an assembly of multiple burner heads filling the entire firebox.

Domestic sizes of conversion burners should be installed with due attention to the method of venting. Draft hoods, conforming to Ameri-

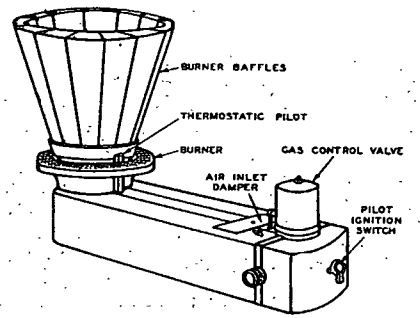


FIG. 20. TYPICAL GAS CONVERSION BURNER

can Standard Requirements, should be installed in place of the dampers used with a solid fuel.

One form of central heating system is the warm air floor furnace[5]. The use of these furnaces is adaptable to mild climates. They are 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 all rooms are on the ground floor, thus giving the heating system flexibility. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below.

Unit Type Heaters

Space heaters may be used for auxiliary heating, but in many cases are installed for furnishing heat to entire buildings. With the exception of wall heaters, they are semi-portable.

Parlor heaters or *circulators* are usually of the cabinet type. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base, passes up inside the jacket around a heating section, and out of the heater at, or near, the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation.

The burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass around baffles within the heating element, and out the flue at the back near the top. They are well adapted for residence room heating and also for stores and offices.

Radiant heaters give off a considerable portion of their heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. An atmospheric burner is supported near the center of the base. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are portable; however, there are also types which are encased in a jacket that fits into the wall with a grilled front.

Gas-fired steam and hot water radiators are other types of room heating appliances. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to steam or hot water radiators. They are usually constructed of sheet metal hollow sections. The products of combustion circulate through the sections and are discharged from a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

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, thermostatic pilot and controls for burners and fan. They are usually mounted in an elevated position from which the heated air is directed downward by louvres. Some unit heaters are suspended from the ceiling, and others are free-standing floor units of the heat tower type.

Unit heaters are available in two types, classified according to their use, with, or without ducts. Only those types of unit heaters tested and approved as warm air furnaces can be connected safely to ducts, as they have sufficient blower capacity to deliver an adequate air supply against duct resistance and are equipped with limit controls.

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. A ratio of about 3 parts primary air to 1 part gas for manufactured gas, and a 5½ to 1 ratio for natural gas, are generally used as theoretical values. 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.

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 per cent if the appliance is to meet *A.S.A.* approval. 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 obtainable.

Little difficulty should be had in maintaining efficient combustion when burning gas. The fuel supply is normally held to close limits of variation in pressure and calorific value and the rate of heat supply is nominally constant. Because the force necessary to introduce the fuel into the combustion chamber is an inherent factor of the fuel, no draft by the chimney is required for this purpose. The use of a draft hood insures the maintenance of constant low draft condition in the combustion chamber with a resultant stability of air supply. A draft hood is also helpful in controlling the amount of excess air and preventing back drafts that might extinguish the flame. (See Chapter 16.)

Due to the use of draft hoods and gas pressure regulators both the input and combustion conditions of gas appliances are maintained quite uniform until deposits of dirt, corrosion, or scale accumulate in the air inlet openings, burner ports or on the heating surface. Periodic cleaning is necessary to keep any gas appliance in proper operating condition.

Measurement of the Efficiency of Combustion

The efficiency of combustion may be judged from the percentage of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. The CO_2 and O_2 may be obtained by means of an Orsat apparatus but the CO must be determined by more accurate equipment. It is customary to use simple indicators to determine whether CO is present and to make adjustments of the appliances to reduce the CO below 4/100 of one per cent before continuing tests in which the CO_2 and O_2 can then be found by use of the Orsat apparatus. Since the ultimate CO_2 for any gas depends on the total hydrogen content the quality of the combustion should not be judged from the value of the CO_2 in the flue gas without reference to the ultimate CO_2 obtainable. Practical values of CO_2 will usually be from 8 to 14 per cent depending on the gas used.

Ratings for Gas Appliances

Input rating for a gas appliance is established by demonstrating that the appliance can meet the Approval Requirements of the *A.S.A.* The tests are conducted at the *A.G.A. Testing Laboratories*. Output rating is determined from the approved input and an average efficiency stated in the Approval Requirements and is the heat available at the outlet.

Sizing Gas-Fired Heating Plants

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 per cent 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 per cent is usually sufficient for pick-up allowance.

In those installations, in mild climates where 100 per cent outside air is used, furnaces should be of larger size in order to provide adequate capacity and quick pick-up 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 18.

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 3,000 ft.

Installations should be made in accordance with recommendations shown in the publications of the *American Gas Association.*

Controls

Temperature controls for gas burners are described in Chapter 34. Some central heating plants are equipped with push-button or other manual control. The main gas valve may be of either the snap action or throttling type. Automatic electric ignition is available.

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 pick-up. General values for these allowances previously have been noted. Detailed information for piping and pick-up allowances for steam, and hot water systems is given in Chapter 18 and for warm air systems in Chapters 21 and 22.

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. 21, 22 and 23 for the several fuels.

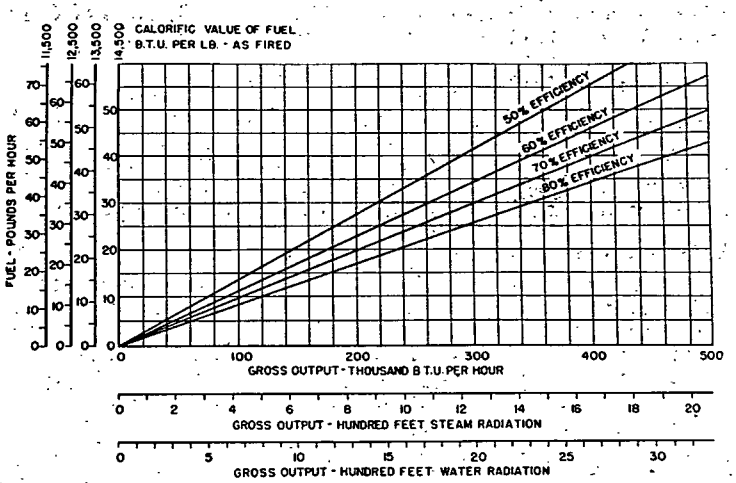


FIG. 21. COAL FUEL BURNING RATE CHART

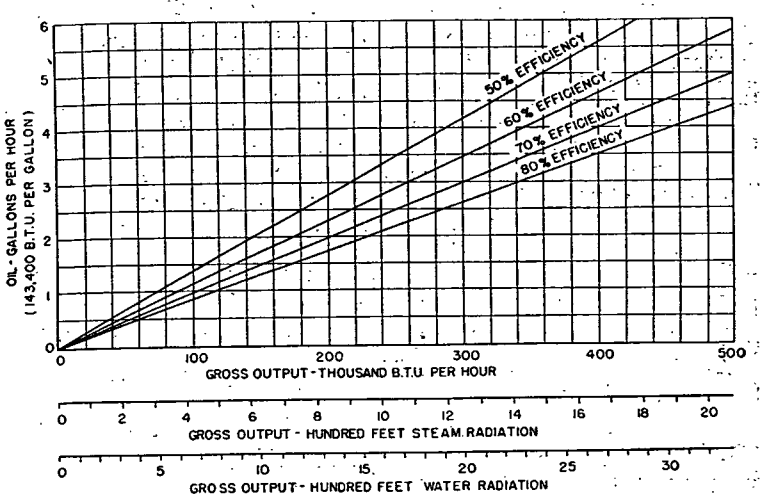


FIG. 22. OIL FUEL BURNING RATE CHART[a]

[a]This chart is based upon No. 3 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. 2 oil (141,000 Btu per gallon) 1.017; No. 4 oil (144,500 Btu per gallon) 0.992; No. 5 oil (146,000 Btu per gallon) 0.982; and No. 6 oil (150,000 Btu per gallon) 0.956.

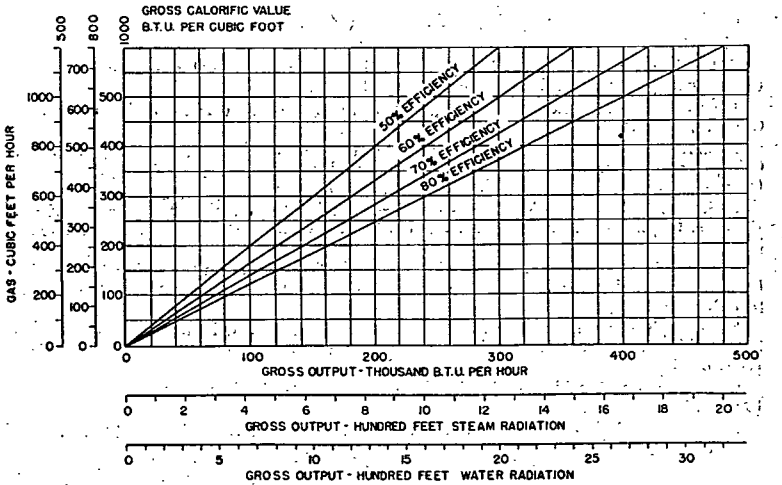


FIG. 23. GAS FUEL BURNING RATE CHART

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.

REFERENCES

[1]—Domestic Burners for Pennsylvania Anthracite (Underfeed Type), (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS48-40).

[2]—*Stoker Manufacturers Association Manual:* Industry Standards, Recommended Practices, Technical Information. Published by *Stoker Manufacturers Association*, 307 N. Michigan Ave., Chicago 1, Ill.

[3]—Code for Determination of Rated Capacities of Anthracite Underfeed Stokers, adopted June 1, 1944, and a Code for Determination of Rated Capacities of Bituminous Underfeed Stokers, adopted May 3, 1944. See *Stoker Manufacturers Association Manual*.

[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. CS(E)104-43).

[5]—Gas Floor Furnaces, Gravity Circulating Type (*U. S. Department of Commerce, National Bureau of Standards*, Commercial Standard No. CS99-42).

BIBLIOGRAPHY

Performance Expectancy of Domestic Underfeed Stokers for Anthracite, by Allen J. Johnson (*Transactions, A.I.M.E.*, Coal Division, Vol. 119, 1936).

The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers—Burning Tests on Four Typical Coals, by R. A. Sherman, E. R. Kaiser and H. R. Limbacher, Technical Report No. 1, Bituminous Coal Research, Inc. (July, 1937) Part II.

Oil Fuels and Burners, by James A. Moyer (McGraw-Hill).

Handbook of Oil Burning, by Harry F. Tapp.

A Study of the Oil Burner as Applied to Domestic Heating, by Arthur H. Senner (Technical Bulletin 109, *U. S. Department of Agriculture*).

Progress in Domestic Oil Heating, by Rene J. Bender (*Mechanical Engineering*, October, 1942).

A.S.H.V.E. Research Report No. 907—Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. Transactions, Vol. 37, 1931, p. 517).

A.S.H.V.E. Research Report No. 925—A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 317).

Air Supply and Its Effect on Performance of Oil Burners and Heating Boilers, by L. E. Seeley, J. H. Powers and E. J. Tavanlar (A.S.H.V.E. Transactions, Vol. 39, 1933, p. 75).

Study of Fuel Burning Rates and Power Requirements of Oil Burners in Relation to Excess Air, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 319).

Oil Burning in Residences, by D. W. Nelson (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 355).

A Study of Oil-Fired Heating Boilers, by R. C. Cross and W. R. Lyman (*Heating and Ventilating*, October, 1931).

Combustion, *American Gas Association*.

Comfort Heating, *American Gas Association*.

Approval Requirements of Central House Heating Gas Appliances, *American Standards Association*.

A Method for Determining Fuel Burning Rates in Heating Boilers Fired by Automatic Devices, by R. C. Cross (*Heating and Ventilating*, January, 1932).

Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 185).

CHAPTER 18

Heating Boilers and Furnaces

BOILERS: Construction, Types, Design Considerations, Testing and Rating 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

IN presenting the subject of Boilers and Furnaces this chapter is divided into two parts; the first dealing with boilers and the second treating warm air furnaces.

HEATING BOILERS

Steam and hot water boilers for low pressure heating are built in a wide variety of types and sizes, many of which are illustrated in the *Catalog Data Section*. They are made of steel or cast-iron.

CONSTRUCTION

The only code governing the construction of low-pressure heating steel and cast-iron boilers that has gained recognition on a national basis 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 to 15 psi for steam and 30 psi for hot water heating boilers. Hot water boilers may be used for higher working pressures for heating purposes or for hot water supply when designed and tested for the higher pressure.

TYPES OF HEATING BOILERS

Heating boilers are classified in a number of different ways, such as:

(*a*) According to materials of construction. These are steel and cast-iron. Very few non-ferrous boilers are made.

(*b*) 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.

(*c*) According to the specific purpose or application for which the boiler is used, such as space heating or domestic hot water supply.

(*d*) 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:

(*a*) Square or rectangular boilers with vertical sections and rectangular grates commonly known as sectional boilers.

(*b*) Round boilers with horizontal pancake sections and circular grates.

Cast-iron boilers are usually shipped in sections and assembled at the place of installation. In the majority of boilers the sections are assembled with push nipples and tie rods. 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.

Capacities of cast-iron boilers range generally from capacities required for small residences up to about 12,000 sq ft of steam radiation. There are a few boilers made with capacities up to 18,000 sq ft of steam radiation. For larger loads, boilers must be installed in multiple.

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. 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 35,000 sq ft of steam radiation.

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 when installing 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 lb per square inch. 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 non-ferrous 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 possible 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 16 and 17.)

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 heat transfer rates in heating boilers 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.

TESTING AND RATING CODES

The Society has adopted four solid fuel testing codes, a solid fuel rating code and an oil fuel testing code.

ASHVE Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—

(Revision of June, 1929) [1], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics.

ASHVE Performance Test Code for Steam Heating Solid Fuel Boilers —Code No. 3—(Edition of 1929)[1] is intended for use with ASHVE Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers [2]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler.

The ASHVE Standard Code for Testing Steam Heating Boilers Burning Oil Fuel [3], (Adopted June, 1932), is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers.

In 1938 the Society adopted a Standard Code for Testing Stoker-Fired Steam Heating Boilers [4], (Adopted June, 1938), which is intended to provide a test method for determining the efficiency and performance characteristics of any stoker and boiler combination burning any type of solid fuel such as anthracite or bituminous coal.

The *Steel Boiler Institute, Inc.* has adopted (June 12, 1945) a Rating Code for Commercial Steel Boilers and Residential Steel Boilers and for Testing Oil-Fired Residential Steel Boilers. The commercial boilers (defined as those having 129 to 2500 sq ft of heating surface) are rated in square feet (steam) on the basis of heating surface with limitations set for grate area, furnace volume and furnace height. The residential boilers (defined as those having not more than 177 sq ft of heating surface) are rated on tests for oil-fired boilers, with limitations in relation to heating surface and testing conditions. Stoker-fired and gas-fired *residential* boilers are rated (*SBI* Net Rating) not in excess of the oil-fired rating. Hand-fired residential boilers are rated (*SBI* Net Rating) not greater than 14 times the heating surface.

Tables 1 and 2 show the *SBI* ratings of residential and commercial steel boilers respectively.

TABLE 1. *SBI* NET RATING DATA FOR RESIDENTIAL STEEL BOILERS—OIL FIRED[a]

SBI Net Rating			Minimum Furnace Volume Cu Ft	Heating Surface Sq Ft
Sq Ft Steam	Sq Ft Water	Btu		
275	440	66000	2.5	16
320	510	77000	2.9	19
400	640	96000	3.6	24
550	880	132000	5.0	32
700	1120	168000	6.4	41
900	1440	216000	8.2	53
1100	1760	264000	10.0	65
1300	2080	312000	11.8	77
1500	2400	360000	13.6	88
1800	2880	432000	16.4	106
2200	3520	528000	20.0	129
2600	4160	624000	23.6	153
3000	4800	720000	27.3	177

[a] Stoker-fired and Gas-fired *SBI* Net Rating not greater than Oil-fired. Hand-fired, *SBI* Net Rating (Steam) not greater than 14 times the square feet of heating surface.

TABLE 2. *SBI* RATINGS FOR COMMERCIAL STEEL BOILERS

Mechanically Fired						Hand Fired					Flow Tappings		
SBI Rating			*SBI* Net Rating			*SBI* Rating			*SBI* Net Rating				
Sq Ft Steam	Sq Ft Water	Btu	Sq Ft Steam	Sq Ft Water	Btu	Minimum Furnace Volume Cu Ft	Minimum Furnace Height[a] In.	Heating Surface Sq Ft	Sq Ft Steam	Sq Ft Water	Btu	Minimum Grate Area Sq Ft	Steam Outlet I.P.S. In.	Steam Return I.P.S. In.
2,190	3,500	526,000	1,800	2,880	432,000	15.7	26	129	1,800	2,880	432,000	1,500	2,400	360,000	7.9	6	3
2,680	4,280	643,000	2,200	3,520	528,000	19.2	28	158	2,200	3,520	528,000	1,830	2,930	439,000	8.9	6	3
3,160	5,050	758,000	2,600	4,160	624,000	22.6	29¼	186	2,600	4,160	624,000	2,170	3,470	521,000	9.7	6	3
3,650	5,840	876,000	3,000	4,800	720,000	26.1	29½	215	3,000	4,800	720,000	2,500	4,000	600,000	10.5	6	3
4,250	6,800	1,020,000	3,500	5,600	840,000	30.4	30	250	3,500	5,600	840,000	2,920	4,670	700,000	11.4	6	3
4,860	7,770	1,166,000	4,000	6,400	960,000	34.8	30½	286	4,000	6,400	960,000	3,330	5,330	800,000	12.2	6	3
5,470	8,750	1,313,000	4,500	7,200	1,080,000	39.1	31¼	322	4,500	7,200	1,080,000	3,750	6,000	900,000	13.4	6	3
6,080	9,720	1,459,000	5,000	8,000	1,200,000	43.5	31¾	358	5,000	8,000	1,200,000	4,170	6,670	1,000,000	14.5	6	3
7,290	11,660	1,750,000	6,000	9,600	1,440,000	52.1	32¾	429	6,000	9,600	1,440,000	5,000	8,000	1,200,000	16.4	8	4
8,500	13,600	2,040,000	7,000	11,200	1,680,000	60.8	34	500	7,000	11,200	1,680,000	5,830	9,330	1,400,000	18.1	8	4
10,330	16,520	2,479,000	8,500	13,600	2,040,000	73.8	35½	608	8,500	13,600	2,040,000	7,080	11,330	1,700,000	20.5	8	4
12,150	19,440	2,916,000	10,000	16,000	2,400,000	86.8	37½	715	10,000	16,000	2,400,000	8,330	13,330	2,000,000	22.5	8	4
15,180	24,280	3,643,000	12,500	20,000	3,000,000	108.5	40¼	893	12,500	20,000	3,000,000	10,420	16,700	2,500,000	25.6	8	4
18,220	29,150	4,373,000	15,000	24,000	3,600,000	130.2	43	1,072	15,000	24,000	3,600,000	12,500	20,000	3,000,000	28.4	8	4
21,250	34,000	5,100,000	17,500	28,000	4,200,000	151.8	46	1,250	17,500	28,000	4,200,000	14,580	23,330	3,500,000	30.9	8	4
24,290	38,860	5,830,000	20,000	32,000	4,800,000	173.5	48¾	1,429	20,000	32,000	4,800,000	16,670	26,670	4,000,000	33.2	8	4
30,360	48,570	7,286,000	25,000	40,000	6,000,000	216.9	54½	1,786	25,000	40,000	6,000,000	20,830	33,330	5,000,000	37.4	10	6
36,430	58,280	8,743,000	30,000	48,000	7,200,000	260.3	60½	2,143	30,000	48,000	7,200,000	25,000	40,000	6,000,000	41.2	10	6
42,500	68,000	10,200,000	35,000	56,000	8,400,000	303.6	66	2,500	35,000	56,000	8,400,000	29,170	46,670	7,000,000	44.7	10	6

[a] Bituminous Stoker Fired.

The *Institute of Boiler and Radiator Manufacturers* has adopted a method of rating cast-iron heating boilers based upon performance obtained under tests. This code[5] now applies to sectional boilers having a grate width of 41 in. or less. Eventually the Institute intends to include all sizes of cast-iron boilers in this program of testing and rating.

The *American Gas Association* has adopted a method of rating gas designed boilers based upon performance under tests. This is described in Approval Requirements for Central Heating Gas Appliances.

The *Heating, Piping and Air Conditioning Contractors National Association* has adopted a method of rating boilers based on their physical characteristics for those 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.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees 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 per cent
Bituminous Coal, hand fired	50 to 65 per cent
Stoker fired	60 to 75 per cent
Oil and Gas fired	70 to 80 per cent

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

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 above 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 *Heating, Piping and Air Conditioning Contractors National Association* 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 2 (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 *pick-up 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

Factors Involved in Boiler Selection

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 6, 8 and 14. 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 equivalent square feet of equivalent radiation.

2. *Hot Water Supply Load.* The estimated maximum heat in Btu per hour required to heat water for domestic use.

When the hot water supply is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. A common practice is to add 240 Btu per hour to the radiation load for each gallon of storage tank capacity. For more specific information see Chapter 50.

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 28. In average house heating systems, it is common practice to consider the piping tax to be equal to 25 per cent of the Net Load. In determining Net *I=B=R* Ratings from Gross *I=B=R* Output, the piping factor allowed varies from 30 per cent for small boilers to 12 per cent for larger boilers.

4. *Warming-Up or Pick-Up 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 radiation and piping. The factors to be used for determining the allowance to be made should be selected from Table 3.

Other items to be considered in boiler selection are:

(*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 break-down, 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.

Selection of 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[6], or from recommendations of the *Heating, Piping and Air Conditioning Contractors National Association*[7] 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 pick-up do not apply. In such a case, the selection must be based on the gross output.

Selection of Steel Heating Boilers

SBI catalog ratings in accordance with the previously mentioned *Steel Boiler Institute, Inc.* 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 177 sq ft of heating surface and having *SBI* net ratings (steam) of not more than 3,000 sq ft if mechanically fired and

TABLE 3. WARMING-UP ALLOWANCES FOR HAND-FIRED LOW-PRESSURE STEAM AND HOT WATER HEATING BOILERS[a,b,c]

DESIGN LOAD (REPRESENTING SUMMATION OF ITEMS 1, 2, AND 3)		PERCENTAGE CAPACITY TO ADD FOR WARMING-UP[c]
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggart (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 292); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 35); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to hand-fired, solid fuel boilers. A factor of 20 per cent over design load is adequate when automatically-fired fuels are used.

[d]240 Btu per square foot.

2,480 sq ft if hand fired, are classified as residence size. An insulated residence boiler for oil, gas or stoker firing may carry a *net load* expressed in square feet of steam radiation of not more than 17 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 per cent of the *SBI* Net Rating. The *SBI* Net Rating (square feet steam) for hand-fired residence boilers is not greater than 14 times the heating surface. If the heat loss from the piping system exceeds 20 per cent of the installed radiation, the excess is to be considered as a part of the *net load*.

Selection Based on Heating Surface and Grate Area

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 square foot) 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 per cent rating on a boiler-horsepower basis. *SBI* ratings for hand firing are based on 10 sq ft of heating surface per boiler horsepower.

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 per hour (see Table 4).
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 per cent.

$$G = \frac{500{,}000}{6 \times 13{,}000 \times 0.60} = 10.7 \text{ sq ft}$$

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 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of 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 per cent for boilers of 200 sq ft and less to 35 per cent for boilers of 4000 sq ft and larger. 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[8].

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.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

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

TABLE 4. PRACTICAL COMBUSTION RATES FOR COAL-FIRED HEATING BOILERS OPERATING AT MAXIMUM LOAD ON NATURAL DRAFT OF FROM ⅛ IN. TO ½ IN. WATER[a]

KIND OF COAL	SQ FT GRATE	LB OF COAL PER SQ FT GRATE PER HOUR
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

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

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.

Particular attention should be given to *fitting connections* to secure conformity with the *ASME* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns 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 facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent cleanout plug should be provided in the case of a single return connection.

Blow-off 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 blow-off 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.

For further data on pipe connections for steam and hot water heating systems, see Chapter 23 and 24 and 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 air-tight 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; and (*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; and (*c*) boiler operating at excessive output.

3. *Water disappears from 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 condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; and (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation 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; and (*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; and (*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; and (*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; and (*e*) improper reduction in breeching size.

8. *Low carbon dioxide.* This may be due on 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 clean-out 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 line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When pressure recedes, 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 lb, close blow-off, 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. The use of soda, or any alkali, vinegar or any acid is not recommended for cleaning heating boilers because of the difficulty of complete removal and the possibility of subsequent injury, after the cleaning process has been completed.

Insoluble compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel 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 some one 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.

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.*

TYPES OF FURNACES

Warm air furnaces may be classified in several different ways:

a. According to method of heat distribution—these are either gravity or mechanical (blower) furnaces.

b. According to fuels for which the furnaces are designed—these are coal hand-fired or stoker-fired, oil, gas, or wood.

c. According to materials of construction—they are cast-iron, low carbon steel, and occasionally high temperature steel alloys.

d. 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 the 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 per cent 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 design 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.

Fans and Motors

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 may limit the maximum operating fan speed. Two-speed motors have given successful operating results and are recommended. 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 33. 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 per cent 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 even 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 3,000,000 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 in the number of square feet of heating surface. Conservative practice indicates that at no time in the heating-up period should the furnace surface be required to emit more than an average of 3500 Btu per square foot. A higher rate of heat emission tends to increase the heat loss up the chimney, and raise fuel consumption, to shorten the life of the furnace, and to overheat the air. The ratio of heating surface to grate area of furnaces for this type of work should never be less than 30 to 1 and as indicated previously may run as high as 50 to 1.

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 43. Ducts are designed by the method outlined in Chapter 41.

MATERIALS AND CONSTRUCTION

Cast-Iron Furnaces

Cast-iron furnaces are made in a multiplicity of designs or shapes. For solid fuels they are generally of round sectional construction, the sections being cemented or bolted together. Various types of radiators for secondary convection heat transfer are employed. Such radiators are of the circular, doughnut type, or tubular type.

Cast-iron is frequently used in the construction of gas or oil-fired furnaces, designs varying considerably with two general types in common use: multi-sectional type and those with single combustion chambers having auxiliary secondary surface.

Cast-iron 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.

Steel Furnaces

Formed sheet steel construction is frequently used in furnace design. 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. Types of design employed vary greatly, although perhaps the most common type consists of a drum and circumferential or rear radiator. Steel gas furnaces may also be sectional in design or may be combinations of common combustion chambers and sectional or tubular radiation surfaces connected to a flue gas collector.

Steel furnaces are made in capacities ranging from those for small insulated residence application with inputs of 40,000 Btu per hour to capacities as large as 600,000 Btu.

Steel furnaces have low heat capacities as a result of their relatively low mass and, therefore, deliver heat rapidly on demand.

RATING OF FURNACES—TESTING AND RATING CODES

Warm air furnaces are generally rated in Btu per hour output at the bonnet (point of heat generation) or at the register (point of heat delivery).

Rating Equations for Gravity Warm Air Furnaces[9]

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:

a. *Hand-fired furnaces Converted to Stoker, Gas, or Oil Firing.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times S \times 1.333 \qquad (2)$$

b. *Hand-fired furnaces, with ratios of heating surface to grate area greater than 15 to 1 and less than 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times S \times 1.333 \qquad (3)$$

c. *Hand-fired furnaces with ratios of heating surface to grate area in excess of 25 to 1.*

$$\text{Bonnet Capacity in Btu per hour} = 1785 \times 25 \times G \times 1.333 \qquad (4)$$

where

S = heating surface, in square feet.
G = actual grate area, in square feet.

The *Register Delivery Rating* is equal to 0.75 x (*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 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:

a. 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.

b. 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 inch	2nd inch	3rd inch	Over 3 in.
Ratio of Effective Area to Total Area	0.40	0.30	0.20	None

c. 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 inch	2nd inch	3rd inch	Over 3 in.
Ratio of Effective Area to Total Area	0.30	0.20	0.15	None

d. 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.

e. 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 and treated for purpose of rating as follows:

a. 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.

b. 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 empirical equations similar to Equations 2, 3 and 4 for gravity furnaces, except that a constant of 2265 is used in place of the 1785.

The following testing and rating codes have been generally accepted in the industry:

Commercial Standards 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 based upon performance under tests. This is described in the Approval Requirements for Central Heating Gas Appliances.

Commercial Standards 118-44 is a method of rating oil-burning floor furnaces by test.

Commercial Standard CS 104-43 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 [10] 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 formulae of the *National Warm Air Heating and Air Conditioning Association* are based on 55 per cent efficiency for gravity coal furnaces and 65 per cent efficiency for forced air coal furnaces. In the tentative Oil Testing Code the contemplated minimum efficiency is 70 per cent for oil fired forced air furnaces. Gravity gas furnaces approved by the *American Gas Association* are assigned a rating based on 75 per cent efficiency. All *forced air gas-fired furnaces* approved by *American Gas Association* are assigned a rating based on 80 per cent 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 2,000 to 4,500 Btu per (hour) (square foot) of heating surface may be obtained without unduly high metal temperatures.

   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 having 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 1,000 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 at 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 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 to burn those who touch them, or to create 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 in any part of the unit which might cause unsafe operation.

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 7/8 in. flange is normally used for this purpose.

d. Suitable provision shall 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 FOR FURNACES

Water evaporating pans are usually located in air which has been heated by contact with the heating surfaces. To change water into vapor capable of being carried in an air stream as part of the mixture, about 1000 Btu per pound are required. There is a trend in present practice toward heating the water in addition to heating the air. 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.

Sprays are usually controlled by solenoid valves wired in parallel with the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Sprays used in connection with commercial or heavy duty plants should be a regulation type of commercial spray. In all cases provision must be made to flush out accumulation of lime and dirt.

REFERENCES

1—See A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, pp. 322 and 332.

2—See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 42.

3—See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 23.

4—See A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 366.

5—*I = B = R* Testing and Rating Code for Low Pressure Heating Boilers (*Institute of Boiler and Radiator Manufacturers*).

6—*I = B = R* Ratings for Cast-Iron Boilers (*Institute of Boiler and Radiator Manufacturers*).

7—Engineering Standards, Part II, Net Square Feet Radiation Loads in 70 Deg Fahr., Recommended for Low Pressure Heating Boilers, 1943 (*Heating, Piping and Air Conditioning Contractors National Association*).

8—Comfort Heating, 1938, pp. 35 to 39 (*American Gas Association*).

9—For definition of Heating Surfaces and Ratings of Wood-Burning Furnaces see Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems, (*National Warm Air Heating and Air Conditioning Association*).

10—Recommended forms for municipal installation and fire codes are included in Volume 7—Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (*National Warm Air Heating and Air Conditioning Association*).

CHAPTER 19

Chimneys and Draft Calculations

Theoretical and Available Draft, Determining Chimney Sizes, Factors Affecting Required Draft, Domestic Chimneys, Observed Test Performance, Draft Requirements of Domestic Appliances, Construction Details, General Considerations

A DRAFT, in the older sense, is a current of air and the draft of a furnace or boiler is that air current which flows through the fire-box and furnishes the oxygen for combustion. In engineering, however, the word *draft* has come to mean that pressure difference which causes this air current to flow and the word will be used in this sense in this chapter.

Draft is usually measured in inches of water and it is proper to speak of the draft in the fire-box or in the smoke breeching, etc., meaning the difference in pressure between the gases within and the air without those parts of a system. Draft is called positive when the pressure within such a part is less than that outside.

Draft is classified as natural and mechanical, depending on whether it is produced by a chimney or by a blower, and mechanical draft is further classified as induced or forced, depending on whether the air is drawn through or forced through the combustion chamber.

Chimneys can serve both to create a draft and to dispose of combustion products at a desirable height. For the latter purpose, chimneys, stacks, or, in the case of ships, funnels, are used in conjunction with mechanical-draft systems.

THEORETICAL DRAFT

If the air in one of two equal chimneys is heated while that in the other is not, the air in the heated chimney will be less heavy than that in the other chimney and a manometer or other pressure gage connecting the two at the bottom will indicate a pressure difference, called natural draft. The pressure of the air at the tops of the two chimneys will be equal, so that the pressure difference between them at the bottom will depend only on their height and the difference in density of the air they contain. The density of the air in either chimney is inversely proportional to its absolute temperature, so that the difference in pressure between them at the bottom will be proportional to their height and to the difference between the reciprocals of the absolute temperatures within them.

The pressure at the bottom of an unheated (and uncooled) chimney will be the same as that of the air outside, so that the unheated chimney can be dropped from the foregoing illustration. The manometer reading will be the same if its free connection is left open to the atmosphere.

These considerations in conjunction with those of barometric pressure and the difference in density of flue gases from that of air lead to the following formula:

$$D_t = 2.96\, H B_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \qquad (1)$$

where

H = height of chimney, feet.
B_o = existing barometric pressure, inches of mercury.
W_o = density of air at 0 F and 1 atmosphere pressure, pounds per cubic foot.
W_c = density of flue gas at 0 F and 1 atmosphere pressure, pounds per cubic foot.

T_o = temperature of air surrounding the chimney, Fahrenheit degrees absolute.
T_c = average or effective temperature of the gases in the chimney, Fahrenheit degrees absolute.

The quantity D_t, yielded by the formula, is the pressure difference between the gas inside and air outside of the chimney, in inches of water, when no flow occurs in the chimney. The quantity is variously known as the theoretical draft, the static draft or the computed draft. It is very useful in predicting and analyzing chimney performance, but it is seldom if ever attained in an actual chimney on account of the friction incident to gas flow, wind effects, etc.

AVAILABLE DRAFT

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 formulae available for ducts, such as the Fanning equation. This procedure results in formulas for the available draft as follows:

For a *cylindrical* stack:

$$D_a = 2.96\,HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126W^2T_cfL}{D^5B_oW_c} \qquad (2)$$

and for a *rectangular* stack:

$$D_a = 2.96\,HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.000388\,W^2\,T_cfL\,(x+y)}{x^2y^2B_o\,W_c} \qquad (3)$$

where

D_a = available draft, inches, water gage.
H = height of chimney above grate, feet.
B_o = existing barometric pressure, inches of mercury.
W_o = density of air at 0 F, 1 atmosphere pressure.
W_c = density of flue gas at 0 F, 1 atmosphere pressure.
T_o = temperature of atmosphere, degrees Fahrenheit absolute.
T_c = temperature of flue gas, degrees Fahrenheit absolute.
W = flue gas flow rate, pounds per second.
f = coefficient of friction.
L = length of friction duct (= H approximately), 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 2 and 3.

1. The *barometric pressure*, represented by B_o, 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:

$$W_c = 0.131CO_2 + 0.095O_2 + 0.083N_2 \qquad (4)$$

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 W_c may be assumed at 0.09.

The density effect on the chimney gases due to superheated water vapor resulting from moisture and hydrogen in the fuel, or due to any air infiltrations in the chimney proper is disregarded. Though 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 outside 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 5 was deduced [1]:

$$T_c = \frac{3.13\,T_i\left[\left(\frac{H_b}{3}\right)^{0.96} - 1\right]}{H_b - 3} \qquad (5)$$

where

T_i = absolute temperature at the center of the connection from the breeching, Fahrenheit degrees.
H_b = the 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 in general 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 chimney design, especially when the construction or the materials are unusual, it is recommended that use be made of the Reynolds number [2] in determining the friction factor, f.

The following problem illustrates the use of Equation 2:

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_o = 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.

Substituting these values in Equation 2 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. 1 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 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 2 and then plotting the results in the manner shown in Fig. 1.

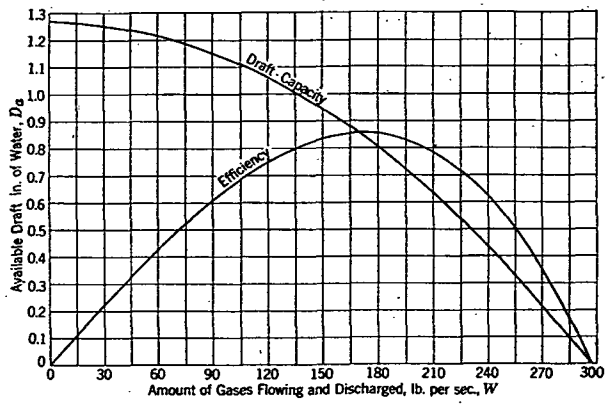


FIG. 1. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

Fig. 2 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 weight of 0.09 lb per cubic foot, 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 2 or 3.

DETERMINING CHIMNEY SIZES

If the required performance for a proposed chimney is known and *if a chimney-gas velocity is assumed*, Equation 2 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:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \qquad (6)$$

The weight of gas per second, $W = 12.075\,\frac{D^2VB_o\,W_c}{T_c}$ from which

$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \qquad (7)$$

where

H = required height of chimney above grate, 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 6 and 7 can be differentiated and

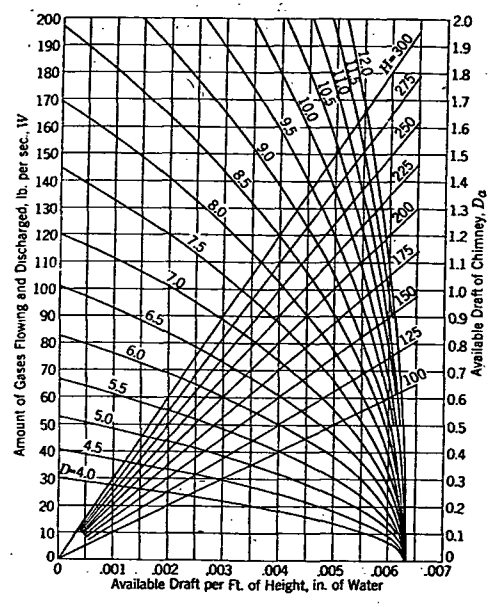


FIG. 2. CHIMNEY PERFORMANCE CHART

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.

the differential set equal to zero to find the minimum. Solution for velocity then yields the following equation:

$$V_e = \left(\frac{0.772T_c\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right)\sqrt{\frac{WT_c}{B_oW_c}}}{fW_c}\right)^{2/5} \qquad (8)$$

where V_e = economical chimney gas velocity, feet per second.

Equations 6, 7 and 8 can of course be simplified if values are assumed for some of the factors in it. Some typical figures for boiler plants are:

Average chimney gas temperature 500 F	T_c = 960 F absolute
Average atmospheric temperature 62 F	T_o = 522 F absolute
Average coefficient of friction 0.016	f = 0.016
Average chimney, gas density, 0 F, 1 Atmosphere	W_c = 0.09 lb per cubic foot
Barometer reading, sea level	B_o = 29.92 in. Hg

When these values are substituted in Equations 8, 7 and 6 respectively, the results are:

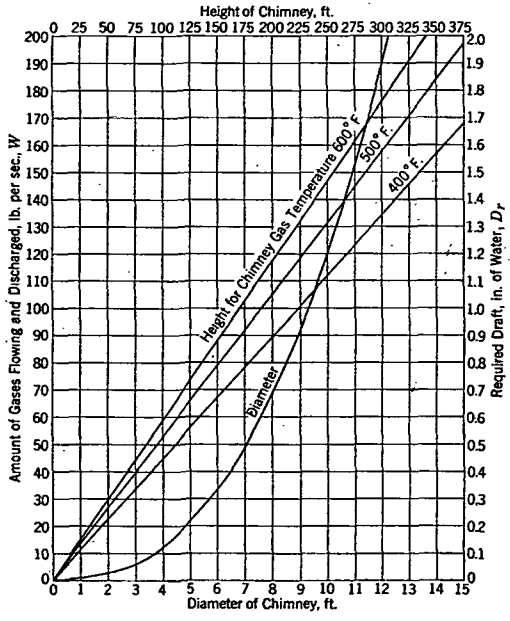
$V_e = 13.7W^{1/5}$ (9) $\qquad D = 1.5W^{2/5}$ (10) $\qquad H = 190D_r$ (11)

Fig. 3 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities computed from Equations 9, 10 and 11. They are based on the operating factors used in reducing Equations 6, 7 and 8 to their simpler form. The sizes shown by the curves in the chart 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 secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 6, 7 and 8.

FACTORS AFFECTING REQUIRED DRAFT

The foregoing considerations deal with chimney size selection when the required draft and flue gas volume and temperature are known. The



Diameter values also for gas temperatures of 400, 500 and 600 F.

FIG. 3. ECONOMICAL CHIMNEY SIZES

required draft is, of course, equal to the sum of all the resistances to gas flow from the ash pit door to and including the chimney connection.

Fig. 4 presents information on the fuel-bed draft loss for various kinds of coal burned at different rates and 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 41.) The friction in straight ducts can be estimated by means of the last terms of Equations 2 and 3.

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

hour per square foot for each Fahrenheit degree temperature difference between the gases and surrounding air.

DOMESTIC CHIMNEYS

The height of a chimney for a residence or apartment is generally limited by the height of the building since it is desirable to have the chimney architecturally congruous with the building. The height desirable from an architectural viewpoint and the location of the chimney may be disadvantageous to the operation of the boiler or furnace and it is therefore important that the manufacturer of the fuel burning appliance to be installed be consulted in regard to the adequacy of the chimney.

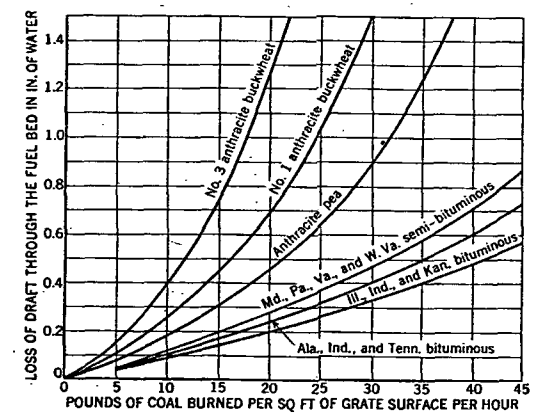


FIG. 4. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

A chimney in order to provide a satisfactory performance must have adequate *height* and *area*, be of permanently tight construction and should be as smooth internally as practicable.

It should be remembered that mechanically fired devices, oil burners and stokers, are equipped with blowers so that, with these devices, the chimney is not required to overcome the resistance of a fuel bed. Nevertheless, a draft in the fire box, of about 0.03 in. of water is considered desirable so that any small openings in the fire box or flue passages will result in leakage of air in, and not leakage of combustion products out, of such parts. This is not to be taken to condone leaks in fire boxes. Such leaks adversely affect plant efficiency.

OBSERVED TEST PERFORMANCE

The observed performances of some brick chimneys[3] are given in Tables 1 and 2.

The tests on which these data are based were made at various outside temperatures as shown and, to make them comparable among themselves, the observed drafts were corrected to 32 F, 1 atmosphere pressure, by the formula:

TABLE 1. TEMPERATURE AND DRAFT IN 9 IN. BY 9 IN. MASONRY CHIMNEY[a]

CHIMNEY HEIGHT, FT		FUEL OIL EQUIVALENT[c] OF CHIMNEY GASES		FLUE GAS TEMP. AT INLET, F	AVERAGE CHIMNEY GAS TEMP. F	OUTSIDE TEMP. F	OBSERVED DRAFT INCH WATER	COMPUTED STATIC DRAFT INCH WATER	CORRECTED DRAFT[b]		INDICATED FRICTION LOSS INCH WATER
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch Water	Computed Static Inch Water	
35	31.5	0.5	10	236	142	68	0.040	0.053	0.076	0.089	+0.013
35	31.5	1.5	8	200	158	81	0.043	0.055	0.086	0.098	+0.012
35	31.5	0.5	10	400	224	77	0.081	0.095	0.128	0.142	+0.014
35	31.5	1.5	8	400	278	80	0.092	0.117	0.139	0.164	+0.025
35	31.5	0.5	10	1000	488	77	0.163	0.191	0.206	0.234	+0.028
35	31.5	1.5	8	1000	601	78	0.169	0.214	0.216	0.261	+0.045
30	27.5	0.5	10	250	160	83	0.043	0.049	0.070	0.076	+0.006
30	27.5	1.5	8	250	195	84	0.055	0.064	0.097	0.106	+0.009
30	27.5	0.5	10	400	213	68	0.075	0.084	0.107	0.116	+0.009
30	27.5	1.5	8	400	269	72	0.094	0.108	0.125	0.139	+0.014
30	27.5	0.5	10	1000	460	64	0.146	0.169	0.174	0.197	+0.023
30	27.5	1.5	8	1000	608	72	0.168	0.193	0.204	0.228	+0.025
25	22.3	0.5	10	250	171	92	0.034	0.037	0.072	0.076	+0.003
25	22.3	1.5	8	200	171	94	0.034	0.036	0.074	0.076	+0.002
25	22.3	0.5	10	400	243	78	0.066	0.073	0.097	0.105	+0.006
25	22.3	1.5	8	400	301	79	0.085	0.091	0.116	0.123	+0.006
25	22.3	0.5	10	1000	504	79	0.124	0.137	0.154	0.167	+0.013
25	22.3	1.5	8	1000	642	80	0.146	0.159	0.177	0.190	+0.013
20	17.2	0.5	10	250	179	94	0.029	0.032	0.059	0.062	+0.003
20	17.2	1.5	8	200	188	94	0.031	0.032	0.059	0.060	+0.001
20	17.2	0.5	10	400	270	81	0.058	0.062	0.082	0.086	+0.004
20	17.2	1.5	8	400	297	55	0.075	0.081	0.087	0.093	+0.006
20	17.2	0.5	10	600	330	50	0.086	0.091	0.095	0.100	+0.005
20	17.2	1.5	8	600	425	55	0.104	0.106	0.116	0.118	+0.002
20	17.2	0.5	10	1000	526	37	0.120	0.130	0.123	0.134	+0.010
20	17.2	1.5	8	915	588	44	0.129	0.134	0.135	0.141	+0.005
15	11.3	0.5	10	250	182	87	0.023	0.023	0.041	0.041	+0.000
15	11.3	1.5	8	200	180	84	0.025	0.024	0.042	0.041	−0.001
15	11.3	0.5	10	400	258	32	0.053	0.055	0.054	0.056	+0.002
15	11.3	1.5	8	400	293	43	0.056	0.057	0.060	0.061	+0.001
15	11.3	0.5	10	600	369	34	0.068	0.071	0.069	0.071	+0.003
15	11.3	1.5	8	600	428	40	0.072	0.076	0.075	0.078	+0.004
15	11.3	0.5	10	1000	577	32	0.094	0.092	0.094	0.092	−0.002
15	11.3	1.5	8	1000	654	55	0.092	0.090	0.100	0.098	−0.002

[a]Actual inside dimensions of flue lining 7¼ × 7¼ in.
[b]Corrected for outside temperature 32 F. Barometer 29.92 in. Hg.
[c]½ gal per hour of fuel oil burned with 10 per cent CO_2 produces 18.4 cfm or 1.38 lb per minute of flue gases (corrected to 70 F, 1 atmosphere pressure); 1½ gal per hour of fuel oil burned with 8 per cent CO_2 produces 67.5 cfm or 5.06 lb per minute of flue gas (corrected to 70 F, 1 atmosphere pressure).

TABLE 2. TEMPERATURE AND DRAFT IN 9 IN. BY 13 IN. MASONRY CHIMNEY[a]

CHIMNEY HEIGHT, FT		FUEL OIL EQUIVALENT[c] OF CHIMNEY GASES		FLUE GAS TEMP. AT INLET, F	AVERAGE CHIMNEY GAS TEMP. F	OUTSIDE TEMP. F	OBSERVED DRAFT INCH WATER	COMPUTED STATIC DRAFT INCH WATER	CORRECTED DRAFT[b]		INDICATED FRICTION LOSS INCH WATER
Nominal	Height Above Thimble	Fuel Oil Gal/Hr	CO_2 Per Cent						Observed Inch Water	Computed Static Inch Water	
35	31.5	0.5	10	204	125	78	0.035	0.034	0.078	0.077	−0.001
35	31.5	1.5	8	206	155	82	0.047	0.052	0.093	0.098	+0.005
35	31.5	1.5	8	600	355	84	0.140	0.141	0.193	0.194	+0.001
35	31.5	0.5	10	600	272	78	0.104	0.117	0.152	0.165	+0.013
35	31.5	0.5	10	1000	438	77	0.163	0.178	0.205	0.221	+0.015
35	31.5	1.5	8	1000	542	84	0.189	0.201	0.236	0.247	+0.012
30	27.5	0.5	10	260	155	83	0.040	0.045	0.081	0.086	+0.005
30	27.5	1.5	8	202	159	83	0.048	0.047	0.089	0.088	−0.001
30	27.5	0.5	10	605	278	80	0.089	0.103	0.129	0.144	+0.014
30	27.5	1.5	8	595	378	72	0.130	0.144	0.161	0.175	+0.014
30	27.5	0.5	10	1007	438	91	0.135	0.148	0.179	0.191	+0.013
30	27.5	1.5	8	995	609	89	0.166	0.168	0.225	0.227	+0.002
25	22.3	0.5	10	258	155	86	0.034	0.034	0.069	0.069	0.000
25	22.3	1.5	8	200	160	86	0.039	0.037	0.074	0.072	−0.002
25	22.3	0.5	10	600	312	86	0.086	0.090	0.122	0.126	+0.004
25	22.3	1.5	8	592	382	83	0.106	0.110	0.140	0.144	+0.004
25	22.3	0.5	10	1007	462	85	0.120	0.127	0.154	0.161	+0.007
25	22.3	1.5	8	1007	607	84	0.151	0.153	0.184	0.185	+0.002
20	17.2	0.5	10	280	175	79	0.030	0.028	0.062	0.060	−0.002
20	17.2	1.5	8	250	192	81	0.044	0.041	0.069	0.066	−0.003
20	17.2	0.5	10	598	324	74	0.089	0.079	0.089	0.099	+0.010
20	17.2	1.5	8	604	410	74	0.090	0.094	0.111	0.115	+0.004
20	17.2	0.5	10	1015	496	79	0.094	0.107	0.115	0.128	+0.013
20	17.2	1.5	8	1000	645	74	0.122	0.127	0.140	0.147	+0.005
15	11.3	0.5	10	270	172	88	0.022	0.021	0.041	0.040	−0.001
15	11.3	1.5	8	200	166	85	0.022	0.020	0.040	0.038	−0.002
15	11.3	0.5	10	604	356	88	0.055	0.052	0.073	0.070	−0.003
15	11.3	1.5	8	599	431	84	0.066	0.062	0.082	0.078	−0.004
15	11.3	0.5	10	1000	560	82	0.066	0.074	0.082	0.091	+0.008
15	11.3	1.5	8	1000	682	85	0.083	0.082	0.101	0.100	−0.001

[a]Actual inside dimensions of flue lining 6½ × 11 in.
[b]Corrected for outside temperature 32 F. Barometer 29.92 in. Hg.
[c]½ gal per hour of fuel oil burned with 10 per cent CO_2 produces 18.4 cfm or 1.38 lb per minute of flue gases (corrected to 70 F, 1 atmosphere pressure); 1½ gal per hour of fuel oil burned with 8 per cent CO_2 produces 67.5 cfm or 5.06 lb per minute of flue gas (corrected to 70 F, 1 atmosphere pressure).

$$O_2 = O_1 + S_2 - S_1 \qquad (12)$$

where

S_1 = computed static (theoretical) draft, experimental conditions.
S_2 = computed static (theoretical) draft, standard conditions.
O_1 = observed draft, experimental conditions.
O_2 = observed draft corrected to standard conditions.

It will be noted that the observed draft exceeded the computed static draft during some observations on the shorter chimneys. This is mainly attributed to the draft producing effect of the hot gases immediately above the chimney. By means of a manometer it was found that a measurable draft existed in this gas column for some distance above the chimney top. However, the temperatures in the chimney were measured with unshielded thermocouples and the actual gas temperatures may have been higher for this reason than the observed temperatures on which the computations of draft were based. The tests were made in calm weather.

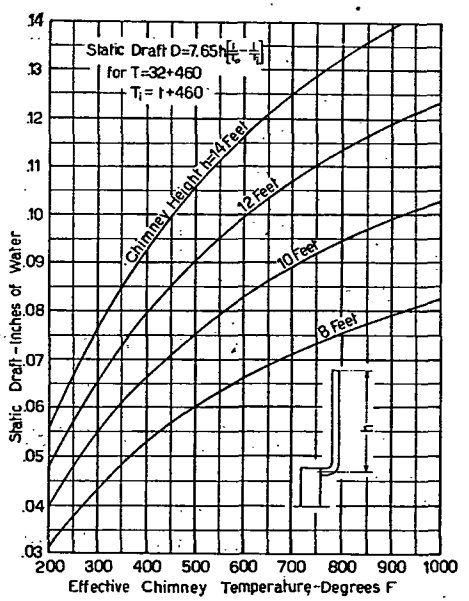


FIG. 5. COMPUTED STATIC DRAFT FOR SHORT CHIMNEYS

Tests were made at the *National Bureau of Standards* to find the draft produced by round, metal smoke pipe set in a vertical position to act as chimneys. Curves are presented in Fig. 5 showing the computed static or theoretical draft for short chimneys for various heights and temperatures. For this purpose, the density of chimney gases was assumed to be the same as that of air at the same condition, since the error thus introduced was not considered important in this case. The results of the tests showed that the following procedures would yield the available draft for 6-in. flue pipe used as a chimney within 10 per cent for the range shown and for fuel burning rates from about one-quarter to three-quarters of a gallon of oil per hour.

Using the temperature at the smoke collar of the heater, find the static draft corresponding to the available chimney height. Then:

1. If the chimney is bare, multiply the static draft by 0.76 to find the available draft.

2. If the chimney is insulated with 1 in. of air-cell material with a ¼-in. air space, multiply the static draft by 0.85 to find the available draft.

3. If the chimney is insulated with 1 in. of air-cell material with a 1-in. air space, open at top and bottom for ventilation, multiply the static draft by 0.81 to find the available draft.

4. If the chimney is insulated with 1 in. of air-cell material and has a 1-in. air space closed at top and bottom to prevent ventilation, multiply the static draft by 0.85 to find the available draft.

The use of the 1-in. air-cell asbestos insulation in the tests discussed is not to be construed as an approval of such insulation in all cases in regard to fire resistance. Several laboratories are working on the fire resistance aspects of the problem but definite rules are not yet available. For coal- or oil-burning devices, a bare smoke pipe is probably safe if kept 2 ft or more from any woodwork and the better the pipe is insulated, or the lower its temperature, the nearer it can be placed to combustible materials.

DRAFT REQUIREMENTS OF DOMESTIC APPLIANCES

Typical flue-gas temperatures and drafts required at rated output for several kinds of domestic heating appliances[4] are contained in Table 3.

TABLE 3. DRAFTS REQUIRED BY TYPICAL DOMESTIC HEATING DEVICES OR APPLIANCES

DEVICE	DRAFT, INCHES WATER	STACK TEMPERATURE F DEG
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 fire-box. [b]For chestnut sized anthracite.

CHIMNEYS FOR GAS HEATING

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 down draft in the chimney from impairing combustion, to provide a relief opening for the products of combustion during down draft 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. While the air drawn in lowers the gas temperature in the chimney, 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 (H_2O) and carbon dioxide (CO_2). In the case of manufactured gas, the presence of organic sulphur compounds, generally between 3 and 15 grains per hundred cubic feet, gives rise to minute percentages of sulphur dioxide and sulphur tri-oxide.

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 of moisture, but also of dilute sulphur trioxide.

Vitreous tile linings with joints which prevent retention of moisture and linings made of non-corrosive materials are advantageous. The protection of unlined chimneys has been investigated and the results indicate that after the loose material has been removed, the spraying with a water emulsion of asphalt chromate will provide excellent protection.

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.

Since a gas designed appliance must be able to operate at rated input (plus 10 or 15 per cent) without chimney connection, and without producing carbon monoxide, the only function of the chimney is to remove the products of combustion from the room. The chimney provides draft to overcome the friction in the flue pipe and chimney, but does not draw air into the appliance.

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 Btu input to the appliance. One chart[5] designed to facilitate selection is shown in Fig. 6. The assumptions made in preparing the chart as well as its limitations should be noted carefully.

Since Fig. 6 has been prepared for circular flues, relative capacities for rectangular and semi-elliptical flues[6] are shown in Fig. 7.

When a flue is connected to several appliances, the number of horizontal runs of various sizes which may be substituted for the single run having a diameter equal to that of the flue may be obtained from Table 4.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Building Code recommended by the *National Board of Fire Underwriters*, Article XI, Sections 1101 to 1105, in which the following are some of the important provisions listed in the 1943 edition:

(*a*) Chimneys erected within or attached to a structure shall be constructed of brick, of solid block masonry, or of reinforced concrete.

(*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*) Every such chimney shall be properly capped with brick, terra cotta, stone, cast-iron, concrete or other approved non-combustible, weatherproof material.

(*d*) Chimneys shall be wholly supported on approved masonry or self-supporting fireproof construction.

(*e*) No such chimney shall be corbeled from a wall more than 6 in.; nor shall such 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 two-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.

(*f*) 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.

(*g*) Smoke flues for warm air, hot water and low pressure steam heating furnaces shall have walls not less than 8 in. thick; the walls may be of solid masonry using brick, stone or concrete, or of solid moulded or solid cast chimney units of concrete, or of burned clay, or of suitably reinforced solid concrete cast in place; provided that for stone masonry other than sawed or dressed stone in courses, the thickness shall be not less

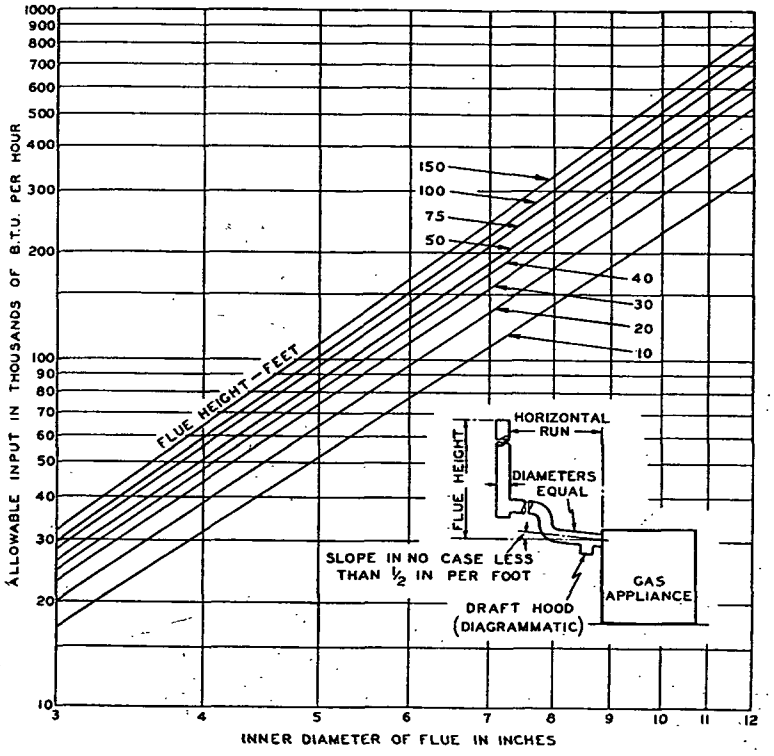


FIG. 6. ALLOWABLE BTU INPUT TO CIRCULAR FLUES FOR DOMESTIC GAS APPLIANCES WITH DRAFT HOODS

Notes:

1. Chart is based on: average flue temperature of 150 F, outside temperature of 60 F, barometric pressure of 30 in. Hg, 100 per cent excess air and 100 per cent dilution at draft hood.
2. Based on terra-cotta lined flues. With rough brick flues, capacities are 15 per cent less.
3. Based on condition that horizontal run is not greater than 20 ft except for a flue height less than 20 ft, in which case the horizontal run is not to have greater length than the height of the flue.
4. Two long radius elbows are included in the horizontal run, the diameter of which is equal to that of the flue.
5. Each additional elbow reduces the allowable horizontal run by a length in feet equal to the diameter in inches.
6. When the horizontal run has an effective length in excess of that given (or additional elbows) the next larger size of flue should be chosen. It is desirable that long horizontal runs be insulated to reduce heat loss of flue products and to conserve draft.
7. Capacities should be reduced 3.5 per cent for each 1000 ft above sea level.

than 12 in. The walls shall be properly bonded, or tied with non-corrosive metal anchors. In dwellings and buildings of like heating requirements the thickness of the chimney walls may be reduced to not less than 3¾ in. when lined with a flue lining conforming to the Code requirements.

(*h*) Required flue linings shall be made of fire clay or other refractory clay to withstand the action of flue gases and to resist, without softening or cracking, the temperatures to which they will be subjected, but not less than 2,000 F, or of cast-iron of approved quality, form and construction. Approved corrosion resistant linings may be used in flues for gas appliances.

(*i*) Required clay flue linings shall be not less than ⅝ in. thick for the smaller flues and increasing in thickness for the larger flues.

(*j*) Flue linings shall be built ahead of the construction of the chimney as it is carried up, carefully bedded one on the other in mortar as hereinafter specified with close fitting joints left smooth on the inside.

(*k*) Flue linings shall start from a point not less than 8 in. below the intake. They

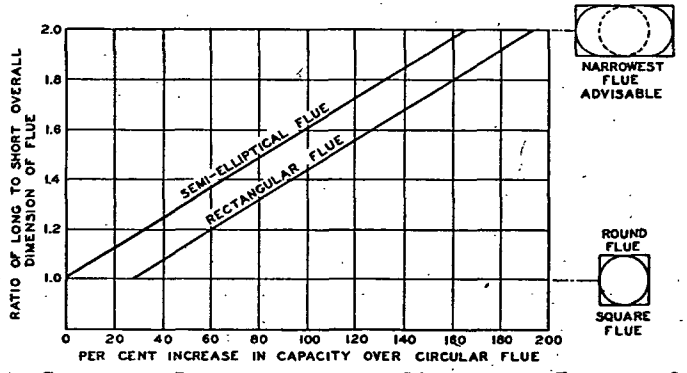


FIG. 7. CAPACITY OF A RECTANGULAR FLUE OR A SEMI-ELLIPTICAL FLUE, WITH SEMI-CIRCULAR ENDS HAVING ITS MINIMUM WIDTH EQUAL TO THE DIAMETER OF A CIRCULAR FLUE, COMPARED WITH THE CAPACITY OF THE CIRCULAR FLUE

shall extend, as nearly vertically as possible, for the entire height of the chimney. It is recommended that flue linings be extended 4 in. above the top or cap of the chimney.

(*l*) Only Portland cement mortar, cement lime mortar or fire clay mortar shall be used in setting flue linings.

For *gas appliances* the Building Code specifies lined chimneys and metal smoke stacks for all 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 non-combustible, corrosion resistant piping of adequate strength and heat insulating value, and having bell and spigot or other acceptable joints. Fig. 6 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 of gas appliances only.*

2. *Type B* vent material should not be used for external chimney flues and external runs of it should not exceed 3 ft outside the building roof. When this requirement makes it necessary to cross over through attic space, the piping should be pitched not less than 45 deg.

3. Because of the small size and low temperature, *Type B* vents should be provided with a vent cap with wire screening to prevent building of birds' nests.

4. Each appliance should have the equivalent of a 4 in. diameter *Type B* vent, even though the appliance may have a 3 in. flue collar. A typical minimum vertical flue size for a frame dwelling is 6 in. in diameter or equivalent.

5. When several floor furnaces are to be vented, it is acceptable practice to connect each of these by means of 4 in. vents to a common 6 in. vertical vent. Lateral piping must have adequate pitch, ¼ in. per foot, and should not exceed 20 ft in horizontal length.

6. An alternate method of connecting several appliances is to run separate *Type B* vents to the attic and then to connect them by means of cross-over piping and Y fittings to a common vertical vent passing through the roof. This reduces the number of holes in the roof.

All flue mortar for flues or vent pipes from gas burning appliances shall be acid resisting.

GENERAL CONSIDERATIONS FOR CHIMNEYS

The draft of domestic chimneys may be subject to a variety of influences not usually encountered in power chimneys[7]. 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

TABLE 4. EQUIVALENT FLUE PIPE SIZES[a]

DIAMETER OF HORIZONTAL RUNS	SIZE OF FLUE						
	3	4	5	6	8	10	12
3	1	2	3	5	9	12	22
4		1	2	3	5	7	11
5			1	2	3	4	7
6				1	2	3	5
8					1	2	3

[a]Comfort Heating (*American Gas Association*).

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 be negative. Although the chimney should extend well above the highest part of the roof, it is impracticable to carry it much beyond this point.

It is also important to consider the source of the air supply for proper combustion. Usually the boiler or furnace is located in the basement. When the furnace room has windows or doors opening to the outside on two or more sides of the house, the leakage of air will be sufficient for combustion, even though the windows and doors may be shut. If, however, the leakage is not sufficient to prevent an appreciable drop of pressure in the furnace room below that of the air outside, the chimney draft will be reduced by the difference between the atmospheric pressure outside and that inside the boiler room. In case the boiler room is fairly tight and is open to the outside on only one side of the house, then 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.

It is not to be assumed that increasing the cross-section area of a chimney will always effect a cure for poor draft. The opposite result may be experienced because of the cooling effect of the larger area. This reduces the theoretical draft and the velocity of the gases, and affords a greater opportunity for counter currents in the chimney. Sometimes the only practical remedy for a chimney with bad draft, when the chimney is of the proper size and is affected by conditions beyond control, is to resort to mechanical draft. This can often be done at small expense and the arrangement can be such that the fan or blower need be operated only when conditions are bad.

Two or more chimneys, either large or small, should never be connected together especially near the bottom. Hot gases in an inverted U-tube thus formed will be in unstable equilibrium. Cold air will descend through one such chimney, from the top, and drive the hot gases out of the other and thus annul 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 un-fired 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 chimneys should not be located opposite each other. The connection from the larger device should be reasonably low down and that from the smaller, up near the ceiling, so that each device can be serviced as well as possible, regardless of the treatment of the other.

Excessive height in a chimney does no harm but means for controlling the draft are more than ordinarily essential if the chimney is too large in capacity. Coal-burning devices often have air leaks around the fire-box and the draft doors sometimes fit poorly so that the fire cannot be controlled at a low rate. Perhaps the simplest remedy for such cases is the barometric damper which admits air into the flue pipe and thus reduces the draft.

Directions for building chimneys for fireplaces are contained in Department of Agriculture Farmers' Bulletin No. 1230.

It is considered bad practice to connect any heating device to a fireplace flue unless the fireplace is effectively sealed.

REFERENCES

1—Notes on Power Plant Design, by E. F. Miller and James Holt (*Massachusetts Institute of Technology*, 1930).

2—A.S.H.V.E. RESEARCH REPORT NO. 1105—Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten, J. B. Schmieler, J. A. Zalovcik and N. Ivanovic (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 35) and for more complete discussion see Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering*, August, 1933).

3—Observed Performance of Some Experimental Chimneys, by R. S. Dill, P. R. Achenbach and J. T. Duck (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 351).

4—*National Bureau of Standards Commercial Standards:* 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; and *Trade Standards:* TS3536a Solid-Fuel Burning Forced Air Furnaces, TS3518 Oil-Burning Floor Furnaces Equipped With Vaporizing Pot-Type Burners.

5—Comfort Heating, 1938, p. 71 (*American Gas Association*).

6—Comfort Heating, 1938, p. 74 (*American Gas Association*).

7—Chimneys and Draft (Chapter 32 in Winter Air Conditioning, by S. Konzo, published by *National Warm Air Heating and Air Conditioning Association*, 1939).

CHAPTER 20

Estimating Fuel Consumption For Space Heating

Fuel Consumption Records, Calculated Heat Loss Estimation Method, Degree-Day Method, Unit Fuel Consumption per Degree-Day, Degree-Day as an Operating Unit, Maximum Demands and Load Factors

MANY methods are in use for estimating in advance of actual operation the anticipated heat or fuel consumption of heating plants over long or short periods. With suitable modification in procedure these same general methods are frequently useful in checking the degree of effectiveness with which heat or fuel is utilized during plant operation.

In applying any of these estimating methods to the consumption of a particular building plant it should be noted that (*a*) reliable records of past heat or fuel consumptions of the building under consideration will usually produce more trustworthy estimates of future consumptions than will any data obtained by averages or from other similar buildings; (*b*) where no past records exist useful data can sometimes be obtained from records of similar buildings with similar plants *in the same locality;* (*c*) records of consumption, which are averages from many types of plants in many types of buildings in various localities, can produce no better than an average estimate which may be far from accurate; (*d*) estimates based on computed heat losses without the benefit of operating data are wholly dependent on how well the computation represents the actual facts.

Estimates based on computed heat losses alone are frequently the only ones possible to obtain, especially where new equipment is put into unusual buildings and there is a scarcity of records and an absence of experience data. Such estimates also have to be made where direct information is not obtainable as, for example, if a survey is being made without the assistance or knowledge of the building operator and thus without information as to the actual consumption. Estimates of this kind are also useful in some cases where a *relative* standard of performance is desired to serve as a base of comparisons in a campaign of fuel utilization.

In interpreting and evaluating heat or fuel consumption estimates as well as in their preparation, 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 allowed for in the estimating method will become controlling and thus give discrepant and even ridiculous results.

Of the various estimating methods in use attention is directed in this discussion to but two as they are illustrative of all, viz: (1) calculated heat loss method, and (2) degree-day method.

CALCULATED HEAT LOSS METHOD

This method is theoretical and assumes constant temperatures for very definite hours each day throughout the entire heating season. It does not

take into account factors which are difficult to evaluate such as opening of windows, abnormal heating of the building, poor heating systems, winter heat gains, such as sun effect, and many others.

In order to apply this method the hourly heat loss from the building under maximum load, or design condition, is computed following the principles discussed in Chapters 6 and 8 and the method described and illustrated in Chapter 14.

In some cases, however, depending on the presence of interior partitions, the computed heat loss is modified when used for estimating the heat or fuel consumption. If the building has no interior walls or partitions then, by the method of Chapters 8 and 14, the infiltration losses are calculated by using only half the total window crack. In such a building the calculated loss need not be modified in order to prepare heat or fuel estimates by this method. Where the building does contain interior walls or partitions instead of using as the calculated heat loss (H) which is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i), it is more desirable to let $H = H_t + \frac{H_i}{2}$.

In predicting fuel consumption for heating a building by the Calculated Heat Loss Method, the general equation is:

$$F = \frac{H\,(t - t_a)\,N}{E\,(t_d - t_o)\,C} \qquad (1)$$

where

F = quantity of fuel or energy required (in the units in which C is expressed).
H = calculated heat loss, Btu per hour, during the design hour, based on t_o and t_d $\left(\text{generally } H = H_t + H_i \text{ but may on occasion equal } H_t + \frac{H_i}{2}\right)$.
t = average inside temperature maintained during heating period, Fahrenheit degrees.
t_a = average outside temperature through estimate period, Fahrenheit degrees (for cities with an Oct. 1-May 1 heating season, see Table 1, Chapter 14).
t_d = inside design temperature, Fahrenheit degrees (usually 70 F).
t_o = outside design temperature, Fahrenheit degrees (see Table 1 in Chapter 14).
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.

Although the assumption of an Oct. 1-May 1 heating season is reasonably accurate in the well-populated New York-Chicago zone, it is not valid as far north as Minneapolis nor farther south than Washington, D. C. and St. Louis. Consequently, it is suggested that allowance be made for this variation, especially in the far north or southern cities.

Example 1. A residence in Chicago is to be heated to 70 F from 6 A.M. to 10 P.M. and 55 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 outside. 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 pound, and since it is purchased steam, the efficiency can be assumed as 100 per cent. From Table 1, Chapter 14, t_a = 36.4 F. The average inside temperature is:

$$\frac{(16 \times 70) + (8 \times 55)}{24} = 65 \text{ F.}$$

Substituting in Equation 1:

$$F = \frac{120{,}000\,(65 - 36.4)\,5088}{1.00\,[70 - (-10)]\,1000} = 218{,}275 \text{ lb.}$$

Example 2. How much would the fuel cost to heat the building in Example 1 during an average heating season with coal at $8 per ton and with a calorific value of 11,000 Btu per pound, assuming that the seasonal efficiency of the plant was 55 per cent?

Solution. Substituting in Equation 1: $F = \frac{120{,}000\,(65 - 36.4)\,5088}{0.55\,[70 - (-10)]\,11{,}000} = 36{,}079$ lb = 18 tons, which, at $8 per ton, costs $144.

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 outside temperature during the heating season is 36.4 F. The seasonal efficiency will be 75 per cent. The heating plant will be thermostatically controlled, and a temperature of 55 F will be maintained from 11 P.M. to 7 A.M. Assume that the price of gas is 7 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:

$$t_a = \frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The maximum hourly heat loss will be:

$$H = 92{,}000 - \frac{26{,}000}{2} = 79{,}000 \text{ Btu.}$$

$$M = \frac{79{,}000\,(66.3 - 36.4) \times 24 \times 210}{100{,}000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

2204.6 × $0.07 = $154.32 = estimated fuel cost per year of heating building.

Several time-saving procedures have been devised for quickly estimating the heat consumption 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[1] 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. Equation 2 is for a one-story residence and Equation 3 is intended for two-story structures.

$$H_1 = A\,(0.45 + U_w + U_c + U_f)\,(t_d - t_o) \qquad (2)$$

$$H_2 = A\,(0.45 + 1.2\,U_w + 0.5\,U_c + 0.5\,U_f)\,(t_d - t_o) \qquad (3)$$

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.
U_w = coefficient of transmission for outside wall.
U_c = coefficient transmission for ceiling from air in rooms to air in attic space (for ventilated attics).
U_f = coefficient of transmission for floor from air in rooms to air in basement.
t_d = inside design temperature, Fahrenheit degrees.
t_o = outside design temperature, Fahrenheit degrees.

Both the graphical method and short-cut formula have been found to give accurate and consistent results for the average residence, but if precise estimates are required, the procedure outlined in Chapter 14 should be used.

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 floor respectively.

Example 4. What would be the total gas consumption over a full heating season of a gas-fired gravity warm air furnace designed according to the Code[2], and with four 12 in. and two 8 in. round leaders to the first floor and six 10 in. leaders to the second floor, if the gas has a heating value of 500 Btu per cubic foot, the plant operates at a 70 per cent seasonal efficiency and is designed to maintain an average inside temperature of 65 F when it is 10 F outside in a city where the average outside temperature is 45 F and the heating season is 5088 hr long?

Solution. The area of the round leaders is: 12 in., 113 sq in.; 10 in., 79 sq in.; and 8 in., 50 sq in. The total Btu transmitted is:

First Floor:	$[(4 \times 113) + (2 \times 50)] \times 111$	= 61,272 Btu per hour.
Second Floor:	$(6 \times 79) \times 167$	= 79,158 Btu per hour.
	Total	140,430 Btu per hour.

Substituting this total heat loss value as H in Equation 1 gives:

$$F = \frac{140{,}430\ (65 - 45)\ 5088}{0.70\ (70 - 10)\ 500} = 680{,}483 \text{ cu ft gas.}$$

DEGREE-DAY METHOD

This method is based on consumption data which have been taken from buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the Calculated Heat Loss Method, it is considered by many to be of more value for practical use.

The amount of heat required by 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*[3] determined from experiment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the mean outside temperature. In other words, on a day when the 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 but are seldom used and are of little value except where the inside temperature to be maintained as, for example, in warehouses, differs greatly from the usual inside temperature range of 68 F to 72 F.

The normal or average number of degree-days, on a 65 F basis, which have occurred over a long period of years, by months, on a 65 F basis are given for various United States and Canadian and Newfoundland cities in Table 1. The United States values were computed from daily mean temperatures recorded by the Weather Bureau over a 43-year period [4] from 1899 to 1941. The number of degree-days for a calendar day of a given year was obtained by taking the difference between 65 F and the mean temperature determined from a reading of the maximum and minimum thermometers for a particular locality. The daily normal was established by taking an average of the 43 daily degree-day figures. These daily values were then added to obtain a monthly normal, and the yearly or seasonal degree-day figure was established by taking a summation of the monthly values. In general, attempts to apply the degree-day

method to fuel consumption over a period of less than a month are of questionable value.

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 will 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 outside temperature variable eliminated.

Studies made by the *National District Heating Association*[5] of the metered steam consumption of 163 buildings located in 22 different cities and served with steam from a district heating company substantiate the fact that the 65 F base originally chosen by the gas industry is approximately correct.

Formula for Degree-Day Method

The general equation for calculating the probable fuel consumption by the degree-day method is:

$$F = U \times N \times D \qquad (4)$$

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 heat loss instead of actual amount of radiation installed).
D = number of degree-days for the estimate period.

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 and are obtained as a result of the collection of operating information. Certain of this information is presented later but before referring to these data attention is directed to the nature of the unit.

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 square foot of radiation, per cubic foot of heated building space, or 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 which are usually unheated. In the absence of data on net heated volume a figure of 80 per cent of the gross volume may be used to obtain the estimated net heated volume. Regardless of which volume figure is used, this unit basis is not considered accurate for estimating fuel consumption due to differences in types of construction, exposure and ratio of exposed area to cubic contents.

TABLE 1. NORMAL DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND[a]

State	City	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Total
Ala.	Birmingham	586	502	318	133	23	1	0	1	10	111	351	582	2618
	Mobile	392	317	178	54	3	0	0	0	1	45	203	374	1567
Ariz.	Phoenix	404	261	152	47	7	0	0	0	0	18	168	389	1446
Ark.	Fort Smith	761	623	393	159	34	1	0	0	12	130	409	708	3230
	Little Rock	700	584	370	148	30	1	0	0	10	121	382	659	3005
Calif.	Los Angeles	276	232	209	158	104	27	1	0	5	43	110	225	1390
	San Francisco	464	340	314	269	252	196	198	180	122	140	242	426	3143
Colo.	Denver	1026	900	792	523	271	62	9	8	124	415	721	1012	5863
	Grand Junction	1230	887	667	381	152	22	1	1	60	352	745	1149	5647
Conn.	New Haven	1103	1024	848	521	222	46	3	12	89	341	660	1010	5879
D.C.	Washington	929	842	637	345	104	14	0	2	43	254	558	870	4598
Fla.	Jacksonville	295	241	131	39	3	0	0	0	0	24	142	286	1161
Ga.	Atlanta	662	568	389	173	32	2	0	1	12	130	393	640	3002
	Savannah	395	335	200	68	6	1	0	0	1	46	209	386	1647
Idaho	Boise	1065	840	680	441	248	87	10	17	135	394	717	1025	5659
Ill.	Chicago	1219	1084	871	536	260	67	7	8	84	334	709	1108	6287
	Springfield	1145	988	732	382	130	15	1	3	64	290	666	1047	5463
Ind.	Evansville	942	826	588	289	82	5	0	1	32	207	540	875	4387
	Indianapolis	1104	982	751	414	155	22	2	4	65	299	663	1026	5487
Iowa	Des Moines	1326	1143	849	449	165	27	2	6	99	357	768	1200	6391
	Sioux City	1410	1216	913	488	194	37	3	10	124	405	841	1268	6909
Kan.	Dodge City	1050	878	669	354	134	16	1	3	58	276	644	994	5077
	Topeka	1104	932	666	331	112	11	0	2	54	258	623	1008	5101
Ky.	Lexington	964	862	650	353	123	14	1	3	47	258	600	916	4791
	Louisville	931	824	595	302	92	7	0	1	34	218	549	875	4428
La.	New Orleans	319	251	130	32	1	0	0	0	0	23	147	305	1208
	Shreveport	522	421	243	83	10	0	0	0	4	71	274	499	2127
Me.	Eastport	1353	1242	1090	778	532	297	159	148	274	528	831	1219	8451
	Portland	1296	1184	1003	668	375	135	29	46	179	459	792	1172	7338
Md.	Baltimore	922	844	650	347	98	12	0	2	34	230	531	852	4522
Mass.	Boston	1101	1027	852	538	248	66	8	16	98	338	651	1000	5943
Mich.	Detroit[b]	1228	1141	943	569	252	56	7	15	109	387	753	1120	6580
	Marquette	1471	1358	1208	801	493	220	87	102	257	556	927	1306	8786
Minn.	Duluth	1726	1504	1267	809	514	232	82	100	289	636	1066	1541	9766
	Minneapolis	1597	1379	1082	580	255	59	8	24	163	484	943	1415	7989
Miss.	Vicksburg	498	413	239	85	9	0	0	0	5	77	268	479	2073
Mo.	Kansas City	1083	921	657	327	106	11	0	2	50	242	599	986	4984
	St. Louis	999	865	616	305	88	7	0	1	36	218	558	917	4610
	Springfield	971	835	599	306	105	11	1	2	45	232	560	900	4567
Mont.	Havre	1548	1374	1109	623	338	129	28	56	275	604	1010	1380	8474
Neb.	Lincoln	1254	1070	798	411	161	23	1	6	84	328	732	1142	6010
	Omaha	1285	1098	813	414	152	21	1	4	84	326	741	1163	6102
Nev.	Winnemucca	1126	887	763	539	320	109	11	23	190	502	802	1099	6371
N. H.	Concord	1339	1212	1005	633	310	98	19	46	189	492	827	1221	7391
N. J.	Atlantic City	944	893	762	489	215	37	1	2	40	250	552	864	5049
N. M.	Santa Fe	1094	892	786	544	297	62	10	15	129	451	772	1072	6124
N. Y.	Albany	1276	1174	955	549	221	44	4	14	116	410	756	1139	6658
	Buffalo[b]	1220	1168	1010	672	352	90	15	24	125	410	747	1102	6935
	New York	1025	958	785	467	176	29	1	4	51	276	600	934	5306
N. C.	Raleigh	699	618	436	213	47	5	0	1	17	155	417	673	3281
	Wilmington	528	481	328	147	25	2	0	0	5	92	310	514	2432
N. D.	Bismarck	1720	1478	1187	650	335	104	20	44	240	605	1059	1527	8969
Ohio	Cincinnati	1009	905	681	374	132	16	1	3	52	274	613	953	5013
	Cleveland	1141	1074	891	558	255	56	8	14	92	354	688	1040	6171
	Columbus	1082	982	762	436	167	26	2	6	68	318	674	1013	5536
Okla.	Oklahoma City	850	696	459	204	56	3	0	0	22	156	461	791	3698
Ore.	Baker	1221	994	839	600	411	210	56	73	261	540	849	1165	7219
	Portland	776	622	530	370	241	106	29	29	109	298	540	729	4379
Pa.	Philadelphia	957	885	696	379	117	16	0	2	36	236	548	877	4749
	Pittsburgh[b]	1043	971	767	448	170	29	3	7	69	324	656	979	5466
S. C.	Charleston	450	386	245	84	7	1	0	0	1	48	227	421	1870
	Columbia	568	488	312	129	18	1	0	0	6	98	330	554	2504

TABLE 1. NORMAL DEGREE-DAYS FOR CITIES IN THE UNITED STATES, CANADA AND NEWFOUNDLAND[a] (CONCLUDED)

State Province	City	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Total
S. D.	Huron	1586	1366	1044	579	264	66	9	21	174	507	966	1415	7997
	Rapid City	1288	1152	980	607	333	104	15	29	188	503	843	1183	7225
Tenn.	Knoxville	774	669	478	229	55	3	0	0	20	189	497	751	3665
	Memphis	709	605	389	159	32	1	0	0	13	127	383	660	3078
	Nashville	785	681	476	222	54	2	0	0	19	170	470	741	3620
Tex.	El Paso	611	433	288	107	15	1	0	0	6	90	369	618	2538
	Fort Worth	582	470	269	100	16	1	0	0	5	76	288	549	2356
	Houston	366	279	144	31	3	0	0	0	1	29	164	343	1360
	San Antonio	382	290	148	38	4	0	0	0	0	32	169	361	1424
Utah	Modena	1191	944	811	569	336	93	7	11	156	503	833	1151	6605
	Salt Lake City	1084	868	708	450	232	62	4	5	95	377	712	1040	5637
Vt.	Burlington	1444	1327	1117	679	330	98	22	47	196	510	866	1294	7930
Va.	Lynchburg	829	735	545	290	82	12	1	2	37	232	524	793	4082
	Norfolk	709	654	492	257	65	6	0	0	9	132	397	664	3385
	Richmond	812	727	548	281	74	9	0	1	27	199	491	775	3944
Wash.	Seattle	765	644	595	440	304	162	71	71	174	372	558	708	4864
	Spokane	1136	935	748	488	282	113	21	37	185	483	816	1061	6305
W. Va.	Elkins	1030	950	774	492	233	59	15	23	114	403	724	997	5814
	Parkersburg	975	888	671	371	129	170	1	3	55	287	619	922	5091
Wis.	Green Bay	1498	1337	1099	663	325	89	17	39	176	495	891	1327	7956
	LaCrosse	1500	1291	1001	533	222	50	7	22	154	459	865	1338	7442
	Milwaukee	1329	1181	969	621	340	100	13	18	120	409	788	1198	7086
Wyo.	Cheyenne	1188	1070	994	726	455	160	42	48	247	592	880	1147	7549
	Lander	1423	1198	1000	674	406	150	28	44	263	630	1020	1401	8237
Alta.	Calgary	1674	1428	1240	750	496	270	124	186	450	744	1170	1395	9,927
	Edmonton	1829	1512	1302	720	434	270	124	186	450	713	1230	1519	10,289
B. C.	Vancouver	899	756	713	510	341	180	62	31	270	496	660	837	5,755
Man.	Winnipeg	2139	1820	1581	810	465	90	.....	62	270	744	1320	1829	11,130
N.B.	Moncton	1519	1428	1178	810	465	210	.....	93	300	620	930	1333	8,886
N. S.	Halifax	1302	1176	1085	780	496	210	.....	.....	210	496	780	1147	7,682
Ont.	Ottawa	1674	1484	1271	690	279	30	.....	.....	210	589	990	1457	8,674
	Port Arthur	1829	1624	1426	900	558	240	62	86	360	713	1140	1550	10,488
	Toronto	1333	1204	1209	720	372	60	.....	.....	180	558	870	1209	7,715
P.E.I.	Charlottetown	1178	1120	1209	870	529	210	.....	.....	600	558	870	1240	8,384
Que.	Montreal	1581	1428	1209	720	310	.....	.....	.....	180	558	960	1395	8,341
	Quebec	1705	1484	1333	870	434	120	.....	31	270	651	1050	1519	9,467
Sask.	Saskatoon	2108	1820	1581	810	465	210	62	155	450	806	1290	1736	11,493
Newf.	St. John's	1286	1215	1153	900	680	412	202	175	339	613	834	1110	8,919

[a]Computed from daily mean temperatures recorded by U. S. Weather Bureau over a 43-year period from 1899 to 1941. Data for Canadian cities abstracted from *Heating & Ventilating*, October, 1939. The National Joint Committee on Weather Statistics, in cooperation with the U. S. Weather Bureau, is preparing revised Degree-Day Normals which will be available in 1946.

[b]Data for these cities and possibly other localities are based on readings taken at more than one official Weather Station during the 43-year period of analysis and are subject to local verification, as these figures are being examined for possible revision by the U. S. Weather Bureau.

The calculated heat loss or its equivalent square feet of calculated radiator surface may be used as the unit. The use of the unit equivalent direct 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 in using the unit equivalent direct radiation 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, although the one most widely used seems to be *units of fuel per degree-day per square foot of*

TABLE 2. UNIT FUEL CONSUMPTION CONSTANTS (U) FOR GAS[a]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 60 F.

Heating Value of Gas Btu per Cu Ft	Hot Water			Steam			Warm Air	
	Cu Ft Gas per Degree-Day per Sq Ft Radiator			Cu Ft Gas per Degree-Day per Sq Ft Radiator			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 500 Sq Ft	300 to 700 Sq Ft	Over 700 Sq Ft	Gravity	Fan Systems
500	0.142	0.135	0.128	0.242	0.231	0.220	0.855	0.820
535	0.132	0.126	0.120	0.226	0.215	0.206	0.800	0.766
800	0.089	0.085	0.081	0.151	0.144	0.137	0.534	0.513
1000	0.071	0.068	0.065	0.121	0.115	0.110	0.428	0.410
1 Therm = 100,000 Btu	Gas Consumption in Therms per Degree-Day							
	0.000708	0.000675	0.000642	0.00121	0.00115	0.00110	0.00428	0.00409

[a]Abstracted from Comfort Heating, *American Gas Association*, 1938.

TABLE 3. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR OIL[b]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 55 F.

Unit	Efficiency in Per Cent				
	40	50	60	70	80
Gal Oil per Sq Ft Steam Radiator	0.00172	0.00137	0.00114	0.00098	0.00086
Gal Oil per Sq Ft Hot Water Radiator	0.00108	0.00086	0.00072	0.00062	0.00054
Gal Oil per 1000 Btu per Hour Heat Loss	0.00715	0.00571	0.00476	0.00409	0.00358

[a]Based on a heating value of 140,000 Btu per gallon.

[b]Abstracted by permission from *Degree-Day Handbook* (Second Edition, 1937), by C. Strock and C. H. B. Hotchkiss.

TABLE 4. UNIT FUEL CONSUMPTION[a] CONSTANTS (U) FOR COAL[b]

Based on 0 F Outside Temperature, 70 F Inside Temperature, and 8-Hour Reduction to 55 F.

Unit	Efficiency in Per Cent				
	40	50	60	70	80
Lb Coal per Sq Ft Steam Radiator	0.0200	0.0160	0.0133	0.0114	0.0100
Lb Coal per Sq Ft Hot Water Radiator	0.0125	0.0100	0.0084	0.0072	0.0063
Lb Coal per 1000 Btu per Hour Heat Loss	0.0825	0.0666	0.0550	0.0471	0.0412

[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.

equivalent direct radiator surface. The equivalent heating load for the hot water supply is not included in the latter unit, but it generally includes the piping load.

Since this unit is the one most widely used at present the unit fuel consumptions given in succeeding paragraphs of this chapter make use of this unit to a considerable extent, although it should be understood that most of these units of consumption can be transposed as desired.

Estimating Gas Consumption

Values of the Unit Fuel Consumption Constant (U) for gas are given in Table 2 for various gas heating values, and different types and sizes of heating plants. They are based on an inside design temperature of 70 F and an outside design temperature of 0 F and apply only to these conditions. For other outside design conditions corrections must be made as given in Table 5.

The factors in Table 2, 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 per cent addition or deduction in these cases is recommended by *A.G.A.* publications. Estimates for industrial buildings where low inside temperatures are maintained cannot be made from this table.

TABLE 5. CORRECTION FACTORS FOR OUTSIDE DESIGN TEMPERATURES[a]

Outside Design Temp. F Deg.	−20	−10	0	+10	20
Correction Factor	0.778	0.875	1.000	1.167	1.400

[a]The multipliers in Table 5, which are high for mild climates and low for cold regions are not in error as might appear. The unit figures in Tables 2, 3, and 4 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 per 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.

For gas heating values other than those given in Table 2, simply interpolate or extrapolate. It will also be noted that Table 2 applies only to small installations. In general the larger the installation the smaller the unit gas consumption becomes and the values in the table should be used with care, if at all, in large gas-burning installations.

Example 5. Estimate the gas required to heat a building located in Chicago, Ill., which has 6287 degree-days and a gas heating value of 800 Btu per cubic foot. The calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperature of −10 F and 70 F.

Solution. From Table 2, the fuel consumption for a design temperature of 0 F with 800 Btu gas is found to be 0.085 cu ft of gas per degree-day per square foot of hot water radiation. From Table 5, the correction factor is 0.875 for −10 F outside design temperature, hence $0.875 \times 0.085 = 0.07438$. By Equation 4,

$$F = 0.07438 \times 1000 \times 6287 = 467{,}000 \text{ cu ft.}$$

Estimating Oil Consumption

Unit fuel consumption factors for oil, similar to those for gas in Table 2, are given in Table 3. The factors in Table 3 apply only to an inside design temperature of 70 F and an outside design temperature of 0 F. For other outside design temperatures, the constants in Table 3 must be multiplied by the values in Table 5 as explained under Estimating Gas Consumption.

Values given in Table 3 assume the use of oil with a heating value of 140,000 Btu per gallon. For other heating values, multiply the values in Table 3 by the ratio of 140,000 divided by the heating value per gallon of fuel being used.

Example 6. Estimate the seasonal oil consumption of an oil-fired boiler in a building located in Minneapolis having a calculated heat loss of 192,000 Btu per hour, burning 144,000 Btu per gallon oil and operating at a seasonal efficiency of 60 per cent. The outside design temperature for Minneapolis is −20 F, and the inside design temperature is 70 F.

Solution. From Table 3, under 60 per cent efficiency and opposite the bottom column, the value of U is found to be 0.00476 gal per 1000 Btu hourly heat loss for 0 F outside temperature.

The correction factor for −20 F outside design temperature from Table 5 is 0.778. Solving, $0.778 \times 0.00476 = 0.00370$. Making a further correction for the heating value:

$$0.0037 \times \frac{140{,}000}{144{,}000} = 0.0036 \text{ gal per 1000 Btu per hour calculated heat loss per degree-day.}$$

From Table 1, the normal degree-days for Minneapolis is 7989. Since U is expressed in 1000 Btu, N is equal to 192. Substituting in Equation 4:

$$F = 0.0036 \times 7989 \times 192 = 5525 \text{ gal.}$$

Estimating Coal or Coke Consumption

Coal or coke consumption estimates are made in exactly the same procedure as for oil. Values of U are given in Table 4 which only apply to inside design temperatures of 70 F and an outside design temperature of 0 F. A correction must be made for other conditions by use of the multiplying factors in Table 5. Data in Table 4 are based on 12,000 Btu per pound 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 7. A building in Marquette, Mich., has an hourly heat loss at design conditions of 240,000 Btu per hour. Based on an inside design temperature of 70 F and an outside design temperature of −20 F, what will be the estimated normal seasonal coal consumption for heating if 12,000 Btu per pound fuel is burned at a 50 per cent seasonal efficiency, and what part of the total will be used during November, December, and January?

Solution. From Table 4, U is 0.0666 lb of coal per 1000 Btu per hour heat loss. Correcting for the outside design temperature of −20 F from Table 5, the value of U is $0.778 \times 0.0666 = 0.0518$. From Table 1, D is 8786 and from the problem, N is 240.

Substituting in Equation 4:

$$F = 0.0518 \times 240 \times 8786 = 109{,}200 \text{ lb.}$$

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 1, the average numbers of degree-days for November, December, and January in Marquette are 927, 1306, and 1471, a total of 3704. The yearly total is 8786, so that during these three months the estimated consumption is:

$$\frac{3704}{8786} \times 109{,}200 = 46{,}200 \text{ lb.}$$

Estimating Steam Consumption

In estimating steam consumption the efficiency is generally assumed at 100 per cent. 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 5) when the unit figures are in terms of square feet of radiation 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 5 may be used:

$$F = \frac{H \times 24 \times D}{1000} \tag{5}$$

where

F = pounds of steam required for estimate period.
H = calculated heat loss, Btu per (hour) (degree difference).
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 inside and outside temperature difference. When degree days are taken from Table 1, an average inside temperature of approximately 65 F is assumed throughout the period. If an average inside temperature other than approximately 65 F is to be used, the number of degree-days should be obtained for the new base.

Example 8. An eight story building in Pittsburgh maintains daytime temperatures of 70 F but allows night temperature to drop to not lower than 60 F. Its calculated heat loss is 10,500 Btu per (hour) (*degree temperature difference*). What is the estimated average yearly steam consumption for building heating?

Solution. Since the average inside temperature is approximately 65 F, the degree-days from Table 1, based on 65 F may be used. Therefore, from Table 1, Pittsburgh has 5,466 degree-days per normal season. Inserting in Equation 5

$$F = \frac{10{,}500 \times 24 \times 5466}{1{,}000} = 1{,}377{,}432 \text{ lb of steam.}$$

Consideration has been given to the difference in steam utilization of different types of buildings and Table 6 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 hot water is not included in the values given in Table 6.

Example 9. A store in Philadelphia with a heating system designed to maintain 70 F inside 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 6, a store would use 0.624 lb of steam per degree-day per 1000 cu ft heated space. From Table 1, Philadelphia has 4749 degree-days per normal year. Inserting in Equation 4:

$$F = 0.624 \times 250 \times 4749 = 740{,}000 \text{ lb of steam.}$$

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 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 figure, by eliminating the outside temperature

TABLE 6. 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
Total or Average	896	1890	0.651	13.4

[a]Principles of Economical Heating, *National Association of Building Owners and Managers.*

variable, indicates whether the operating efficiency of the plant is above or below the previous month or year.

The figures in Table 7 illustrate a typical example of a method of using the degree-day for making heating comparisons for one building for two consecutive heating seasons. The heat quantity figures inserted are pounds of steam, but a similar comparison could be made using pounds of coal, gallons of oil, or cubic feet of gas.

For such a comparison, a two-year record is often used, as shown in Table 7. The year under consideration may then be compared, month by month, with the previous year. Column 3, Consumption for Heating, would be used if the same fuel is used for heating and process steam. Some reasonable figure must be assumed for the process requirement and should be deducted from the amount shown in column 2. This would leave in column 3 only the fuel chargeable to heating. The degree-day values in column 5 are obtainable from the local Weather Bureau. Figures in column 6 are obtained by dividing corresponding values in column 3 by the degree days in column 5. The heating index in column 6 is, then, a figure of heat consumption, corrected for outdoor temperature, and should be relatively constant month by month. Column 7 in Table 7 may be used if the heat consumption is to be compared on a building volume basis with average values shown in Table 6.

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

TABLE 7. HEAT CONSUMPTION RECORD FOR COMPARISON

HEATING SEASON		COL. 1	COL. 2: TOTAL CONSUMPTION	COL. 3: CONSUMPTION FOR HEATING	COL. 4: AVG. MEAN TEMP.	COL. 5: DEG DAYS 65 F BASE	COL. 6: LB/DEG DAY	COL. 7: LB/DEG DAY/M CU FT
1942-43		Sept.	337,500	170,500	65	146	1,170	0.575
		Oct.	834,200	667,200	53	339	1,966	0.970
		Nov.	1,446,600	1,279,600	44	641	1,990	0.982
		Dec.	2,176,400	2,009,400	25	1,233	1,630	0.804
		Jan.	2,332,200	2,165,200	22	1,297	1,670	0.822
		Feb.	2,131,100	1,964,100	28	1,106	1,775	0.888
		Mar.	2,021,900	1,854,900	31	1,032	1,799	0.885
		Apr.	1,241,500	1,074,500	43	647	1,660	0.818
		May	672,500	505,500	55	303	1,670	0.822
		June	258,600	91,600	.....	50	1,830	0.905
		July	188,400	.....	.....	.....	.....	.....
		Aug.	180,100	.....	.....	.....	.....	.....
		Total	13,821,000	.....	.....	.....	.....	.....
1943-44		Sept.	330,200	146,200	61	167	875	0.431
		Oct.	887,100	703,100	52	410	1,718	0.845
		Nov.	1,525,200	1,341,200	39	812	1,653	0.815
		Dec.	2,045,500	1,861,500	28	1,120	1,660	0.817
		Jan.	1,933,400	1,749,400	30	1,044	1,670	0.825
		Feb.	1,990,200	1,806,200	30	1,111	1,624	0.800
		Mar.	1,984,100	1,800,100	31	1,021	1,760	0.868

If, for example, the heat consumption in March, 1943, is compared with that in March, 1944, it will be found that in the latter the steam consumption is 1799 − 1760 = 39 lb less which is a decrease of 2.2 per cent.

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 are available for a number of buildings in Detroit, as shown in Table 8[6].

These maximum demands were measured by an attachment on the condensation meter and therefore represent the amounts of condensation 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 8, 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.

SEASONAL EFFICIENCY

The task of predicting fuel consumption within reasonably accurate limits is a simple one where sufficient experience data are available for the fuel in question. Such data can be analyzed to the point where average unit factors can be determined and expressed in such terms as, for example, average gallons of oil actually burned per square foot of calculated steam radiator surface per degree-day. The unit U can be inserted directly in Equation 2 without reference to efficiency. Such experience factors are available for gas (see Table 2) and for district steam (Table 6), but not for coal or oil.

Since values of U are not available for oil or coal, an assumed seasonal efficiency E must be used. Selection of a value for this E must be made

TABLE 8. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS

BUILDING CLASSIFICATION	LOAD FACTOR	LB OF DEMAND PER HOUR PER SQ FT OF EQUIVALENT INSTALLED RADIATOR SURFACE
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

with caution, for its use implies a meaning not commonly associated with the word efficiency and consequently is frequently misleading.

The input of heat to a building consists not only of the energy in the fuel but that from occupants, the sun, appliances, processes, and all other sources. In many cases these make up, over a period, an important percentage of the total heat required, and if they are not taken into account a calculation of *efficiency* can show a figure over 100 per cent.

For this and other reasons the actual seasonal efficiency is a difficult thing to determine. Published data are widely scattered and insufficient. From the available published material it is found that the seasonal efficiency varies over a wide range, depending on the fuel used, and it varies widely even for a given fuel. For example, in a recent survey of 30 houses in one locality there was found a variation of from 45 to 75 per cent in the utilization efficiency depending on the fuel[7].

REFERENCES

1—Graphical Method of Calculating Heat Losses, by Paul D. Close (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 345).

2—Standard Gravity Code for the Design and Installation of Gravity Warm Air Heating Systems (11th edition), and the Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems (*National Warm Air Heating and Air Conditioning Association*).

3—See Industrial Gas Series, House Heating (*third edition*), published by the *American Gas Association*.

4—Received from U. S. Weather Bureau September, 1943.

5—Report of Commercial Relations Committee, *Proceedings, National District Heating Association*, 1932.

6—The Heat Requirements of Buildings, by J. H. Walker and G. H. Tuttle (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 171).

7—Heat Losses and Efficiencies of Fuels in Residential Heating, by R. A. Sherman and R. C. Cross (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 185).

CHAPTER 21

Gravity Warm Air Systems

Warm Air Leaders, Stacks, and Registers; Return Air Ducts, Grilles, and Shoe Connections; Outline of Design Procedure

WARM air heating systems of the *gravity* type are described in this chapter [1]. In these systems the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing, while in the mechanical type a fan may supply all or part of the motive head.

In general, a 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 top or sides near 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 run in the inside partitions of the building are called stacks. The heated air is finally 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. A sectional view of a typical plant showing good installation practice is given in Fig. 1.

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.

WARM AIR LEADERS, STACKS, AND REGISTERS

In a gravity circulating warm-air furnace system, the size of the leader pipe to a given room depends upon the length of the leader and the temperature of the warm air entering the room at the register. For most successful operation, the furnace should be centrally located with respect to register and stack positions so that the leaders will be of uniform length and as short as possible, in which case the frictional resistance to air flow and the temperature loss from the ducts will be about the same for all runs.

In the Standard Code for Installation of Gravity Warm Air Heating Systems, the design was originally based on the heat carrying capacities per square inch of leader pipe area with register air temperatures of 175 F. In a recent revision of the entire design procedure, as shown in the section entitled *Outline of Design Procedure*, the carrying capacities of leader pipes have been expressed directly in terms of Btu per hour.

In general it is advisable to use two or more leader pipes to rooms requiring more than the capacity of a 12 in. round pipe. The tops of all sizes of leader pipes should be cut into the furnace bonnet at the same elevation, and from this point there should be a uniform upgrade of at least 1 in. per foot of run. Leaders over 12 ft in length, or having a large number of elbow fittings should be avoided if possible. In cases where such leaders are necessary, it is recommended that smooth transition fittings be used, and that duct insulation be applied. Asbestos paper, unless of the corrugated type, should not be considered as insulation. To assist in balancing the air distribution of the system, a damper should be placed in each leader pipe except one, this latter leader preferably being connected to a room heated at all times, such as a living room.

In a gravity circulating system, the ratio of stack to leader area is quite important, although little is gained by providing wall stacks with areas in excess of 75 per cent of their connected leader pipe area. In most cases a 3¼ in. × 12 in. stack is the largest which can be installed in normal wall construction. Hence, any room having a heat loss much in excess of 9000 Btu per hour, will require two or more stacks, or one oversized stack built into a 6 in. studding space, providing the design register temperature is to be retained at the value of 175 F as recommended.

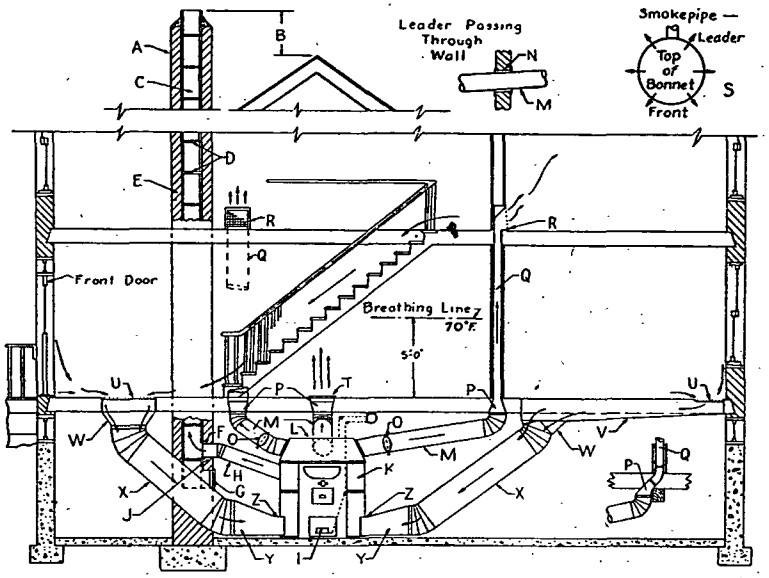


FIG. 1. A SECTIONAL VIEW OF A TYPICAL PLANT SHOWING GOOD INSTALLATION PRACTICE[a]

A. House chimney, no bends nor offsets.
B. Top of chimney at least 2 feet above ridge of roof.
C. Flue lining, fireclay.
D. All joints air tight.
E. At least 8 in. brick.
F. No other connection beside that to furnace.
G. Cleanout frame and door, airtight.
H. Smoke pipe, end flush with inner surface of flue.
I. Draft door.
J. Use flue thimble.
L. Casing hood or bonnet, top of all leader collars on same level.
K. Casing body.
M. Round leader, pitch 1 in. per foot.
N. Sleeve with air space around leader where passing through wall.
O. Dampers in all leaders.
P. Transition fittings.
Q. Rectangular wall stack.
R. Baseboard register.
S. Distribute pipes equally around bonnet.
T. Floor register.
U. Return air face.
V. Panning under joist.
W. Transition collar.
X. Round return pipe.
Y. Transition shoe.
Z. Top of shoe at casing not above grate level.

[a]From *N.W.A.H.&A.C. Assn.* Standard Code Application Manual.

Registers used for discharging warm air into rooms should have a net area not less than the area of the leader pipe to which the register is attached. First story registers should be connected through boot and register box extensions having areas at least equal to leader areas. Upper story registers should be of the same width as the wall stack, and should be placed either in the baseboard or sidewall, preferably without offsets.

First story registers may be of the baseboard or floor type, with the former location preferred. High sidewall registers in gravity systems deliver more warm air into the room than do baseboard registers, but most of the additional air merely results in high temperatures at the ceiling.

RETURN AIR DUCTS, GRILLES, AND SHOE CONNECTIONS

The ducts through which air is returned to the furnace should be designed to minimize resistance to air flow. They should be of ample area, in excess of the total area of warm-air pipes, and should be streamlined. Horizontal ducts should pitch at least ½ in. per foot downward toward the furnace, avoiding fittings which would require lifting of the return air after the duct has passed under some obstacle.

The return air grilles should have free areas at least equal to the ducts to which they connect and should be installed in the floor, or in the base-

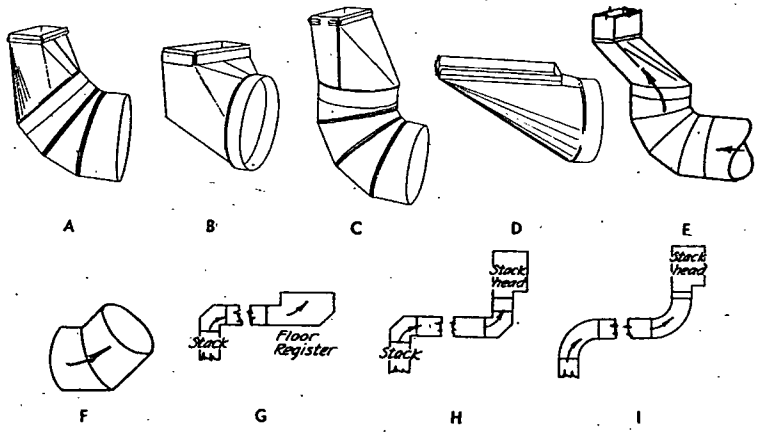


FIG. 2. TYPICAL WARM AIR BOOTS

board with the top edge of the grille not more than about 14 in. above the floor line. Frictional resistance in the return air system is as detrimental as is resistance in the warm-air system, so that care should be exercised in locating return air grilles which require long return ducts.

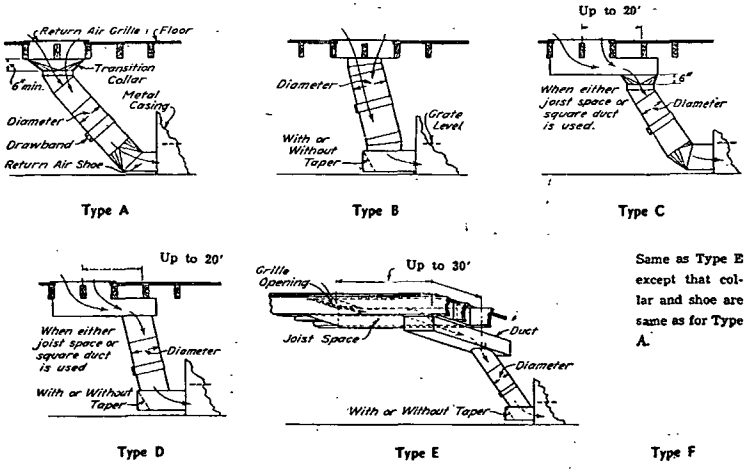
The placement and number of return grilles will depend upon the size, details, and exposure of the house. Small compactly built houses may be adequately served by a single return grille effectively placed in the central hall. It is usually desirable to have two or more returns, provided that in two-story residences one return is placed to effectively receive the return air at the foot of the stairs. A return air connection must be carried to any room whose floor level is below that of adjacent rooms.

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus, in rooms having only small windows the grille can be brought as close to the furnace as possible, but if the room has large window exposure the grille should be located near the exposure. The frictional resistance of the long ducts used in parallel with short return ducts must be reduced to compensate for the length. Return ducts from upstairs rooms may be

necessary in spaces which are closed off from the rest of the house or which have much outdoor exposure. Return grilles on different floor levels should not be connected to the same vertical return duct.

Ducts returning air to the furnace should avoid heat sources which tend to reheat the return air. If the duct must be run over the top of the furnace, or above the vent pipe from the furnace, insulation should be interposed between the heat source and the duct.

Circulation of air is facilitated if the air can slide down a pipe inclined at approximately 45 deg and into a furnace shoe connection having a cross-sectional area equal to that of the pipe. The top of the return shoe should enter the casing below the level of the grate in the case of a coal



Note: For Types C, D, E, and F return-air duct systems, reduce the carrying capacities shown in Table 8 by 1 per cent for each 4 ft additional length in the horizontal run.

FIG. 3. TYPICAL ARRANGEMENTS OF RETURN-AIR DUCT SYSTEMS

furnace, and not more than 14 in. above the floor in the case of oil or gas furnaces. In order to accomplish this the shoe is made wide.

OUTLINE OF DESIGN PROCEDURE

The data underlying the design procedure are given in detail in a circular[2] issued by the University of Illinois. In this procedure the design of the warm-air duct system is considered as an entire unit, so that for a given heat loss the sizes of leaders, stacks, boots, stackheads, and registers are all correlated. Similarly in the case of return ducts, the selection specifies a complete unit consisting of return grille, return duct, and shoe connection.

Recommended Standard Sizes

For the purpose of simplification and standardization, selected combinations of commercial sizes of warm air pipes, return air pipes, ducts, grilles, fittings, and registers are designated as Combination Numbers. The numbers assigned and the combinations selected as standard are listed in the following tables [3].

Table 1—Combination Numbers 1 to 5—First Story Warm Air Ducts and Registers.

Table 2—Combination Numbers 11 to 16—Second Story Warm Air Ducts, Single Wall Stacks, Fittings, and Registers.

Table 3—Combination Numbers 21 to 24—Second Story Warm Air Ducts, Double Wall Stacks, Fittings, and Registers.

Table 4—Combination Numbers 31 to 38—Return Air Ducts, Fittings, and Grilles.

The selected types of boots are shown in Fig. 2 and their resistances expressed in equivalent elbows are shown in Table 5. It is essential that free areas be maintained throughout fittings.

The selected types of return air ducts and fittings are shown in Fig. 3.

TABLE 1. FIRST STORY WARM-AIR DUCTS[a]

Combination No.	Leader Pipe Diameter, In.	Register Size, In.		
		Floor	Baseboard	
			Size	Extension
1	8	8 x 10	10 x 8	2¼
2	9	9 x 12	12 x 8	2¼
3	10	10 x 12	12 x 9	3¼
4	12	12 x 14	13 x 11	5¼
5	14	14 x 16	.....	.....

[a]When the calculations indicate a requirement for a given room greater than Combination No. 4, two or more smaller units totalling the required capacity are recommended.

TABLE 2. SECOND STORY WARM-AIR DUCTS—SINGLE WALL STACKS AND FITTINGS

Combination No.	Leader Pipe Diameter, In.	Stack[b] Size In.	Register Size, In.			
			Floor	Baseboard		Sidewall
				Size	Extension	
11	8	10 x 3¼	8 x 10	10 x 8	2¼	10 x 8
12	9	12 x 3¼	9 x 12	12 x 8	2¼	12 x 8
14	10	14 x 3¼	10 x 12	12 x 8	2¼	12 x 8
15	12	12 x 5¼	.....	12 x 9	3¼	.....
16	12	14 x 5¼	.....	13 x 11	5¼	.....

[b]Recommended stack sizes. Tables may also be applied to 3 in. and 3½ in. stack depths.

TABLE 3. SECOND STORY WARM-AIR DUCTS—DOUBLE WALL STACKS AND FITTINGS

Combination No.	Leader Pipe Diameter, In.	Stack Size, In.		Register Size, In.			
				Floor	Baseboard		Sidewall
		Internal	External		Size	Extension	
21	8	2½[c] x 10	3⅛[c] x 10⅝	8 x 10	10 x 8	2¼	10 x 8
22	8	3 x 10	3⅝ x 10⅝	8 x 10	10 x 8	2¼	10 x 8
23	9	2½[c] x 12	3⅛[c] x 12⅝	9 x 12	12 x 8	2¼	12 x 8
24	9	3 x 12	3⅝ x 12⅝	9 x 12	12 x 8	2¼	12 x 8

[c]Commercial sizes vary ⅛ in. from values shown.

Carrying Capacity

The Btu carrying capacities of the selected warm air and return air combinations are shown in Tables 6, 7 and 8:

Table 6—Combination Numbers 1 to 5—Warm Air Carrying Capacities to first story registers with 1 to 5 elbows and with leaders 4 to 24 ft long.

Table 7—Combination Numbers 11 to 16 and 21 to 24—Warm Air Carrying Capacities to second story registers with 1 to 5 elbows with leaders 4 to 24 ft long.

Table 8—Combination Numbers 31 to 38—Return Air Carrying Capacities for types *A*, *B*, *C*, *D*, *E*, and *F* return combinations.

TABLE 4. RETURN AIR DUCTS

Combination No.	Duct Dia. In.	Area at Shoe Connection, Sq In.	Metal Grille Sizes, Choose One A	B	C	When Joist Lining is Used[d] No. of Joists Lined	Minimum[e] Depth, In.	When Duct is Used, Choose One	
31	10			8 x 14	10 x 12	1	7	14 x 6	12 x 8
32	12		6 x 30	8 x 24	12 x 14	1	9	22 x 6	16 x 8
33	14	170	8 x 30	10 x 24	14 x 16	1	12	28 x 6	22 x 8
34	16	220	10 x 30	12 x 24		2	8	28 x 8	22 x 10
35	18	280	12 x 30	14 x 24		2	10	36 x 8	28 x 10
36	20	340	14 x 30	18 x 24		2	12.5	36 x 10	30 x 12
37	22	420	18 x 30			2	15.0	42 x 10	36 x 12
38	24	500	20 x 30			2	18.0	42 x 12	36 x 14

[d]Based on 14 in. space between joists.

[e]Use full depth of joist except when joist depth is less than minimum depth required, *when pan must be used.*

TABLE 5. RESISTANCES OF WARM AIR BOOT COMBINATIONS EXPRESSED IN ELBOW EQUIVALENTS

Warm Air Boot	Name of Combination	Equivalent No. of 90-Deg Elbows
A	45-Deg Angle Boot and 45-Deg Elbow	1
B	90-Deg Angle Boot	1
C	Universal Boot and 90-Deg Elbow	1
D	End Boot	2
E	Offset Boot	2½
F	45-Deg Angle	½
G	Floor Register—Second Story	3
H	Offset	3
I	Offset	2

Design Procedure

The steps to be taken in designing a gravity warm air duct system are:

1. Calculate the heat loss from each room as explained in Chapters 6, 8 and 14.

2. Prepare a layout showing (*a*) furnace, (*b*) chimney connection, (*c*) warm air registers (whether floor, baseboard or wall), (*d*) return air grilles.

3. Indicate on each warm air run (using symbols shown in Fig. 4): (*a*) whether the room to be heated is on the first or second story, (*b*) the approximate length of leader pipe in the basement, (*c*) the number of right angle elbows required, including the elbow at the boot connection (see Fig. 2), (*d*) whether the register is to be located in the floor, in the baseboard, or in the wall.

4. Show the number and proposed locations of return air grilles and the type of return air system (see Fig. 3).

TABLE 6. WARM AIR CARRYING CAPACITY, BTU DELIVERED, FIRST STORY REGISTERS[a]
Length of Leader Pipe—in Feet

Combination No.	No. of Elbows	4 Ft	6 Ft	8 Ft	10 Ft	12 Ft	14 Ft	16 Ft	18 Ft	20 Ft	22 Ft	24 Ft
1	1	6,020	5,850	5,680	5,510	5,340	5,170	5,000	4,830	4,660	4,490	4,320
2		7,620	7,400	7,180	6,970	6,760	6,540	6,320	6,110	5,890	5,680	5,460
3		9,400	9,140	8,870	8,600	8,340	8,070	7,810	7,540	7,270	7,010	6,740
4		13,350	12,970	12,590	12,210	11,830	11,450	11,080	10,700	10,320	9,950	9,560
5		17,520	17,020	16,530	16,040	15,560	15,050	14,550	14,050	13,560	13,060	12,560
1	2	5,850	5,660	5,490	5,330	5,160	5,000	4,840	4,670	4,510	4,340	4,180
2		7,360	7,150	6,940	6,730	6,520	6,320	6,110	5,910	5,700	5,500	5,290
3		9,090	8,840	8,580	8,320	8,060	7,800	7,550	7,290	7,040	6,780	6,520
4		12,910	12,540	12,170	11,800	11,430	11,060	10,690	10,320	9,950	9,580	9,210
5		16,940	16,450	15,990	15,500	15,040	14,550	14,080	13,600	13,120	12,650	12,150
1	3	5,620	5,460	5,310	5,150	4,990	4,830	4,670	4,510	4,350	4,190	4,030
2		7,120	6,910	6,710	6,510	6,310	6,110	5,900	5,700	5,500	5,300	5,100
3		8,780	8,530	8,280	8,030	7,780	7,530	7,290	7,040	6,800	6,550	6,300
4		12,450	12,100	11,750	11,400	11,050	10,700	10,350	10,000	9,650	9,300	8,950
5		16,360	15,900	15,440	14,970	14,510	14,050	13,600	13,130	12,660	12,200	11,750
1	4	5,420	5,260	5,110	4,960	4,800	4,650	4,500	4,350	4,190	4,040	3,890
2		6,860	6,660	6,460	6,270	6,080	5,890	5,690	5,500	5,300	5,110	4,910
3		8,460	8,200	7,980	7,740	7,500	7,260	7,020	6,780	6,550	6,310	6,070
4		12,010	11,670	11,330	10,990	10,650	10,310	9,970	9,630	9,280	8,950	8,610
5		15,770	15,320	14,880	14,420	13,990	13,540	13,100	12,650	12,200	11,750	11,310
1	5	5,240	5,090	4,940	4,790	4,640	4,500	4,350	4,200	4,050	3,910	3,760
2		6,630	6,440	6,250	6,060	5,880	5,690	5,500	5,320	5,130	4,940	4,750
3		8,180	7,950	7,720	7,490	7,260	7,030	6,800	6,560	6,330	6,100	5,860
4		11,610	11,290	10,950	10,620	10,300	9,970	9,640	9,320	8,990	8,660	8,320
5		15,250	14,800	14,380	13,950	13,520	13,090	12,650	12,230	11,800	11,370	10,940

[a]Additional values for 6 and 7 elbows are given in original Manual.

TABLE 7. WARM AIR CARRYING CAPACITY, BTU DELIVERED, SECOND STORY REGISTERS[a]

Length of Leader Pipe—in Feet

COMBINATION No.[b,c]	No. OF ELBOWS	4 FT	6 FT	8 FT	10 FT	12 FT	14 FT	16 FT	18 FT	20 FT	22 FT	24 FT
11-22	1	8,370	8,140	7,900	7,670	7,430	7,190	6,950	6,710	6,470	6,240	6,000
12-24		10,040	9,760	9,470	9,190	8,900	8,620	8,330	8,050	7,770	7,480	7,200
14		11,710	11,380	11,050	10,720	10,390	10,060	9,720	9,390	9,060	8,730	8,400
15		16,200	15,750	15,300	14,840	14,380	13,920	13,460	13,000	12,550	12,100	11,640
16		18,920	18,390	17,850	17,310	16,780	16,240	15,710	15,180	14,640	14,100	13,570
11-22	2	7,940	7,720	7,500	7,280	7,050	6,830	6,600	6,370	6,150	5,930	5,700
12-24		9,540	9,270	9,000	8,730	8,460	8,190	7,920	7,650	7,380	7,110	6,840
14		11,120	10,810	10,500	10,180	9,870	9,550	9,230	8,920	8,610	8,290	7,980
15		15,400	14,970	14,530	14,100	13,670	13,230	12,800	12,380	11,930	11,500	11,070
16		17,980	17,470	16,960	16,450	15,950	15,430	14,830	14,420	13,910	13,400	12,890
11-22	3	7,530	7,320	7,110	6,900	6,680	6,470	6,250	6,040	5,830	5,620	5,400
12-24		9,030	8,780	8,520	8,270	8,010	7,750	7,500	7,240	6,990	6,730	6,470
14		10,530	10,240	9,940	9,650	9,350	9,050	8,750	8,450	8,160	7,860	7,560
15		14,580	14,180	13,780	13,370	12,960	12,530	12,120	11,710	11,300	10,890	10,480
16		17,040	16,550	16,070	15,580	15,110	14,620	14,140	13,660	13,180	12,700	12,210
11-22	4	7,120	6,920	6,720	6,520	6,310	6,110	5,900	5,700	5,500	5,300	5,100
12-24		8,530	8,290	8,050	7,810	7,570	7,330	7,080	6,840	6,600	6,360	6,120
14		9,950	9,670	9,390	9,110	8,830	8,550	8,260	7,980	7,700	7,420	7,140
15		13,780	13,390	13,000	12,610	12,220	11,830	11,440	11,050	10,670	10,280	9,890
16		16,080	15,620	15,170	14,710	14,260	13,810	13,350	12,900	12,440	11,980	11,530
11-22	5	6,700	6,510	6,320	6,130	5,940	5,750	5,560	5,370	5,180	4,990	4,800
12-24		8,040	7,810	7,580	7,350	7,130	6,900	6,670	6,440	6,220	5,990	5,760
14		9,370	9,110	8,850	8,580	8,310	8,050	7,780	7,510	7,250	6,980	6,720
15		12,970	12,600	12,240	11,870	11,500	11,140	10,770	10,400	10,040	9,680	9,310
16		15,140	14,710	14,280	13,850	13,420	13,000	12,570	12,140	11,710	11,280	10,850

[a]When floor registers are used, see Fig. 2.
[b]No. 21 for Btu values multiply 11-22 values by 0.83.
[c]No. 23 for Btu values multiply 12-24 values by 0.83.

5. From Table 6, for first story, or from Table 7 for second story, select the combination number for the warm air system which will supply the heat required to each room, with the number of elbows and length of leader pipe previously determined. Then, using the combination number as found, read directly in Tables 1, 2, or 3 the leader, stack, and register sizes required.

6. From Table 8 select the combination number for the return air system to correspond with the Btu serviced and the type of return air system. Then from Table 4 select the duct and grille sizes, etc., corresponding to the same combination number.

7. Select a furnace having a register delivery, in Btu per hour, equal to the total heat loss from the structure.

TABLE 8. RETURN AIR—CARRYING CAPACITY—BTU SERVICED

RETURN AIR COMBINATION No.	DUCT DIA IN.	TYPE A BTU PER HR	TYPES B AND C BTU PER HR	TYPE D BTU PER HR	TYPE E BTU PER HR	TYPE F BTU PER HR	RETURN AIR COMBINATION No.
31	10	11,300	9,500	7,800	5,000	7,800	31
32	12	16,300	13,700	11,300	7,200	11,300	32
33	14	22,200	18,700	15,300	9,800	15,300	33
34	16	29,000	24,400	20,000	12,800	20,000	34
35	18	36,700	30,800	25,300	16,200	25,300	35
36	20	45,300	38,000	31,300	20,000	31,300	36
37	22	54,800	46,000	37,800	24,100	37,800	37
38	24	65,200	54,800	45,000	28,700	45,000	38

Examples 1 and 2 will illustrate the use of the tables in selecting warm air and return system sizes.

Example 1. For a room which has a heat loss of 22,500 Btu per hour select the size of first story warm air system. There are three elbows and the leader is approximately 10 ft long.

Solution: Since 22,500 Btu is beyond the capacities shown in Table 6, it is necessary to select two units of 11,250 each. From Table 6 in 10 ft leader column and in section for three elbows find 11,400 as nearest capacity which corresponds to Combination Number 4 in first column. Refer to Combination Number 4 in Table 1 and find that the leader should be 12 in. in diameter and should be used with a 12 x 14 in. floor register or a 13 x 11 in. baseboard register with a 5¼ in. extension.

Example 2. What is the size of a return system of Type D which is to service 35,000 Btu. per hour?

Solution: From Table 8 find Combination Number 37 which will service 37,800 Btu per hour. Refer to Table 4 to find that Combination Number 37 will require a 22-in. diameter duct, a shoe area of 420 sq in., a metal grille 18 x 30 in., a duct 42 x 10 in. or 36 x 12 in. If joist lining is used the minimum depth should be 15 in. for two 2-joist spaces 14 in. wide, or 10 in. for three joist spaces.

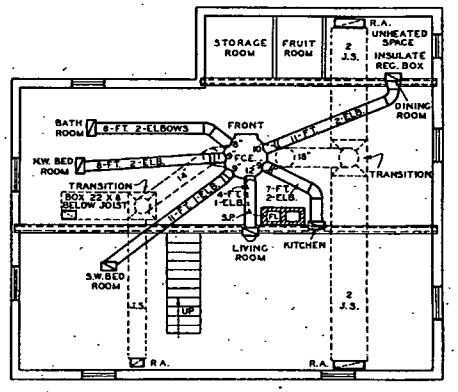


FIG. 4. TYPICAL BASEMENT LINE DRAWING

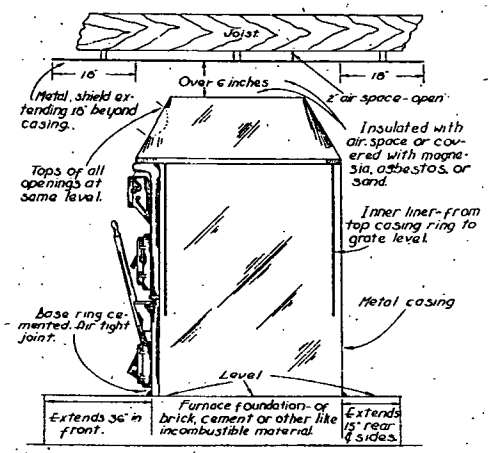


FIG. 5. DETAILS OF FURNACE BONNET, CASING, AND FOUNDATION (FROM GRAVITY CODE AND MANUAL)

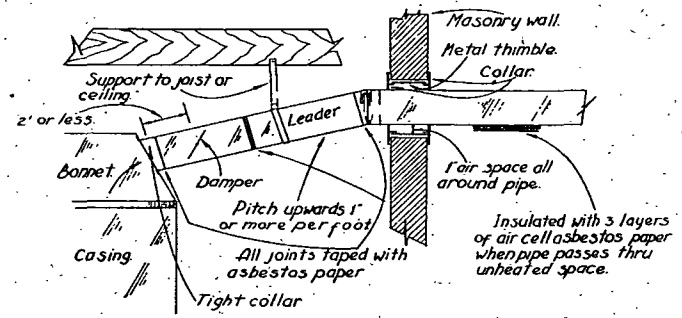


FIG. 6. DETAILS OF BONNET AND LEADER OF GRAVITY WARM-AIR FURNACE (FROM GRAVITY CODE AND MANUAL)

Figs. 5 and 6 show recommended practice as given in the *N.W.A.H. &A.C. Assn.* Gravity Code and Manual. For construction, design features, and ratings of gravity furnaces see Chapter 18.

REFERENCES

1—The engineering data were obtained from University of Illinois, *Engineering Experiment Station Bulletins* Nos. 141, 188, 189 and 246; Warm Air Furnaces and Heating Systems, by A. C. Willard, A. P. Kratz, V. S. Day, and S. Konzo. See also Gravity Code and Manual for Gravity Warm Air Heating Systems, published by the *National Warm Air Heating and Air Conditioning Association.*

2—Simplified Procedure for Selecting Capacities of Duct Systems for Gravity Warm-Air Heating Plants, by A. P. Kratz and S. Konzo (*University of Illinois, Engineering Experiment Station Circular* 45, Dec., 1942).

3—Gravity Code and Manual for Gravity Warm Air Heating Systems, Second Edition, 1945, *National Warm Air Heating and Air Conditioning Association.*

CHAPTER 22

Mechanical Warm Air Systems

Air Distribution, Standard Combinations of Parts, Automatic Controls, Simplified Design of Heating System, Cooling Methods, Cooling System Design

IN mechanical warm air or fan furnace heating systems [1], the air circulation is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom, as in gravity systems described in Chapter 21. The advantages of mechanical systems, as compared with gravity systems, are:

1. The furnace need not be centrally located but may be placed in any part of the basement.

2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view where desired.

3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.

4. Humidity control is more readily attained.

5. The air may be cleaned by sprays or filters, or both.

6. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.

7. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

8. Ventilation air may be positively introduced and heated.

The construction features of mechanical warm air furnace units and discussions of the function and selection of the various parts, such as the furnace, casings, motors, filters and controls are included in Chapter 18.

AIR DISTRIBUTION

The conditions of comfort obtained in a room are greatly influenced by the type of register used and the locations of the supply registers and return grilles. In general it has been found that changes in the type, air velocity, and location of the supply register affect the room conditions much more than the changes in the location of the return grilles. 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 dropping off from the exposed walls and glass. Another method is to locate the supply openings near the floor on the inside wall and the return openings near the greatest outside exposure. In any case the warm air registers should be located so that the air stream never discharges directly into space that will normally be occupied by people at rest. Tests in the Warm Air Research Residence [2] have indicated that continuous fan operation provided better results than intermittent fan operation.

Register and Grille Openings

Supply registers located in the floor require attention to keep them clean and are usually avoided. Tests conducted in the Warm Air Research

Residence 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 which throw the air downward toward the floor and diffuse it at the same time. For baseboard registers air temperatures under 125 F and air velocities over 500 fpm should be avoided as they may cause drafts.

High side wall registers must be of such type that the air is delivered horizontally or in a slightly downward direction, and must be so located as to avoid impingement of air on ceiling or wall. Directional flow diffusing type should be used to insure best results. Register air velocities

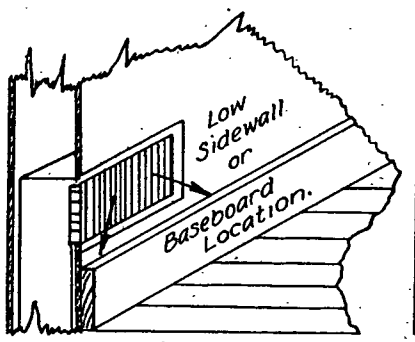


FIG. 1. RECOMMENDED TYPE OF BASEBOARD AND LOW SIDEWALL REGISTERS[a]

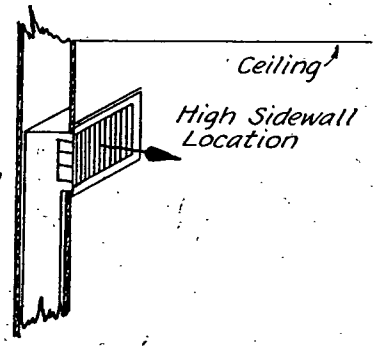


FIG. 2. RECOMMENDED TYPE OF HIGH SIDEWALL REGISTERS[b]

[a]Vertical bars with adjustable deflection, or fixed vertical bars with deflections to right and left not exceeding about 22 deg. For low sidewall location, the deflection for horizontal, multiple valve, registers should not exceed 22 deg. For baseboard locations, the deflection for horizontal, multiple valve, registers should not exceed about 10 deg.

[b]Horizontal valves, in back or front, to give downward deflections not to exceed from 15 to 22 deg.

should be such that the air stream carries to the opposite exposure. Velocities under 500 fpm are not recommended. Basic rules for the location and selection of registers are given in Section C of the Code and Manual (Textbook Section 7) of the *National Warm Air Heating and Air Conditioning Association.*

Registers should be well proportioned and decorated to harmonize with the trim. Air supply registers should be equipped with dampers and all registers should be sealed against leakage around edges. The register types shown in Figs. 1 and 2 have been recommended as standard by the *National Warm Air Heating and Air Conditioning Association.*

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.

Return air grilles may be located in hallways, near entrance doors, under windows, in exposed corners, or inside walls, depending on location of supply registers. Baseboard returns are preferable to floor grilles.

Dampers

Suitable dampers for air direction or volume control are essential to any trunk or individual duct system. Special care must be used in the design

of any system to avoid turbulence and 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 pipes. *Splitter dampers* are used where a branch is taken off from a main trunk. *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. It is essential that a damper with positive locking device be provided for each main or duct branch. Labels placed on ducts should indicate the room being served. Damper positions should be marked for summer and winter operation, and to avoid tampering.

Ducts

The ducts may be either round or rectangular in cross section. The radii of elbows should preferably be not less than one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts. Warm air ducts passing through cold spaces, or where located in exposed walls, should have ½ to 2 in. of insulation.

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 canvas. Motors and mountings must be carefully selected for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing. Installation of a fan directly under a cold air grille is usually not recommended.

STANDARD COMBINATIONS OF PARTS

The combinations of parts selected as a standard by the *National Warm Air Heating and Air Conditioning Association* are shown in Tables 1 and 2. A method for selecting these combinations is indicated in the section entitled *Simplified Method of Designing Domestic Forced-Air Systems.*

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[3].

Controls which are considered desirable for this system are:

1. A *thermostat* located in a living room where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. The thermostat location should not be on an outside wall, in a bed room, bath room or sun room, or in a location where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct, register or chimney.

2. A *fan switch* control located in the bonnet to start blower operations at temperatures between 110 and 130 F, and to stop the blower at about 25 to 30 F 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 results these settings should be as low as is feasible.

3. A *protective high limit switch* located in the bonnet to stop the system independently of the thermostat if the bonnet temperature exceeds 175 F.

4. On oil and gas burner installations, a protective control should be included which will stop the system if the fire is extinguished or if there is a failure of the ignition system.

5. On automatic stoker installations, a control is usually included which will start the operation regardless of thermostat settings whenever the bonnet temperature indicates that the fire is dying, or a time interval contactor is used that will start the stoker to run a few minutes out of each hour.

6. A *humidistat* to regulate the moisture supplied to the rooms, located either in one of the rooms or in the main return duct near the furnace.

TABLE 1. WARM AIR DUCT SYSTEM COMBINATIONS OF PARTS SELECTED AS STANDARD

Combination No.	Stack Size In.	Branch Pipe Size, In.		Register Size, In. (See Figs. 1 and 2)				Required Increase in Width of Trunk Duct, In.
					High Sidewall			
		Round	Rectangular	Baseboard or Low Sidewall	For throws less than 13 ft	For throws more than 13 ft	Floor Registers[a]	
1	2	3	4	5	6	7	8	9
41	10 x 3¼	6	4 x 8	10 x 6	10 x 6	10 x 4	8 x 10[a]	1
42	10 x 3¼	6	4 x 8	10 x 6	10 x 6	10 x 4	8 x 10[a]	2
43	12 x 3¼	7	5 x 8	12 x 6	12 x 6	12 x 4	9 x 12[a]	3
44	14 x 3¼	8	6 x 8	14 x 6	14 x 6	14 x 4	9 x 12[a] or longer	4
45	10 x 3¼ (2-Stacks)	9	8 x 8	(2) 10 x 6 or (1) 24 x 6	(2) 10 x 6 or (1) 24 x 6	(2) 10 x 4 or (1) 24 x 4	10 x 12[a]	6
46	12 x 3¼ (2-Stacks)	10	10 x 8	(2) 12 x 6 or (1) 30 x 6	(2) 12 x 6 or (1) 30 x 6	(2) 12 x 4 or (1) 30 x 4	12 x 14[a]	7

[a]Use these items only when the building construction or capacity requirements necessitate the use of floor registers. The sizes listed for floor registers correspond to the standard sizes for gravity warm-air systems, except for the sizes of the floor box collars. The use of standard blind boxes is suggested.

SIMPLIFIED METHOD OF DESIGNING DOMESTIC FORCED-AIR SYSTEMS

A simplified method for selecting the combinations of branches, boots, stacks, and registers, is given in the Code and Manual (Textbook section No. 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.

a. Location of room, that is, whether on first or second story.

b. Actual length of basement duct from bonnet to boot, in feet.

c. Btu loss from room to be heated.

d. Equivalent lengths in feet of all fittings and of the register. Fig. 3 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 150,000 Btu per hour. The capacities shown in Tables 3 and 4 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. water gage. The use of this method assumes 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.

TABLE 2. RETURN-AIR DUCT SYSTEM COMBINATIONS OF PARTS SELECTED AS STANDARD

Combination No.	Return-Air Intake Size, In.		Riser Size, In. Where Stack is Used in Stud Space	Branch Pipe Size, In.		When Joist Lining is Used[b] Number of Joist Spaces Lined and Minimum Depth of Space Required	Required Increase in Width of Trunk Duct (for 8 In. Depth of Duct), In.
	Baseboard	Floor[a]		Round	Rectangular		
1	2	3	4	5	6	7	8
51	10 x 6	6 x 10 or 4 x 14	10 x 3¼[c]	6	4 x 8	1 space of 3 in. depth	1
52	10 x 6	6 x 10 or 4 x 14	10 x 3¼[c]	6	4 x 8	1 space of 3 in. depth	2
53	12 x 6	6 x 12 or 6 x 14	12 x 3¼[d]	7	5 x 8	1 space of 4 in. depth	3
54	14 x 6	6 x 14	14 x 3¼[d]	8	6 x 8	1 space of 5 in. depth	4
55	24 x 6 or 30 x 6	6 x 30	Two stacks each 10 x 3¼[c]	9	8 x 8	1 space of 6 in. depth or 2 spaces of 3 in. depth	6
56	30 x 6	6 x 30	Two stacks 12 x 3¼[d]	10	10 x 8	1 space of 7 in. depth or 2 spaces of 4 in. depth	7
57	.....	8 x 30	.....	12	15 x 8	1 space of 9 in. depth or 2 spaces of 5 in. depth	11

[a]Use these items only when building construction, or capacities, requires the use of floor intakes. The sizes listed correspond to standard sizes for gravity installations, except floor box collars. The use of standard blind boxes is suggested.

[b]Based on 14 in. space between joists. Use full depth of joist, except when joist depth is less than minimum depth required, in which case a **drop pan must be used.** This may occur when two or more return ducts are connected to the same joist space.

[c]If it is desired to use 14 in. x 3⅝ in. stud space, it makes no difference whether this space has protruding keys or not.

[d]If it is desired to use 14 in. x 3⅝ in. stud space, the plaster base must be smooth, without any protruding plaster keys to interfere with the flow of air.

The combination numbers shown in the right hand column of Tables 3 and 4 correspond to those given in Tables 1 and 2.

Tables 3 and 4 are also applicable for the selection of the return air branches. A depth of 8 in. has been adopted as the standard for the trunk ducts. The width of a trunk duct serving two branches is determined by adding to the width of the remote branch the value shown in column 9 of Table 1, or column 8 of Table 2.

For buildings having a heat loss in excess of 150,000 Btu per hour the design procedure may be that given in the Technical Code, Fourth Edition, of the *National Warm Air Heating and Air Conditioning Association*. Air duct sizes and air distribution may also be calculated in accordance with data given in Chapters 40 and 41.

COOLING METHODS

A slight cooling effect may be obtained under certain conditions by the use of the cooler basement air. A more positive cooling effect may be

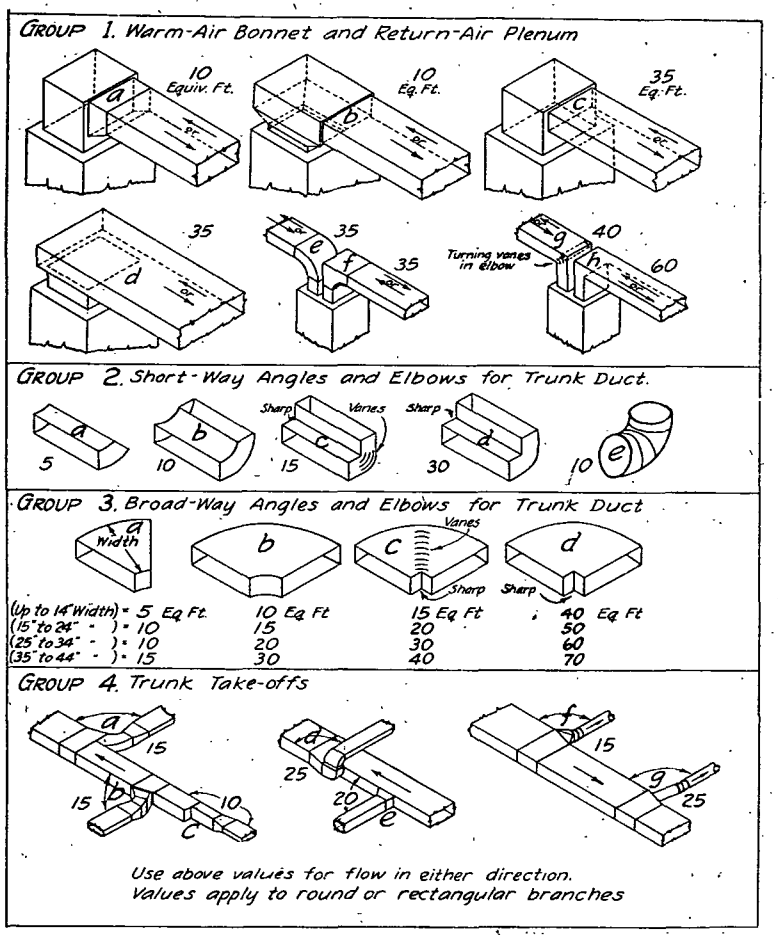


FIG. 3. EQUIVALENT LENGTH OF FITTINGS

obtained by the use of an air washer where the temperature of the city or well water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the temperature of the leaving water is below the dew-point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew-point. Ample coil surface and fan capacity are necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with a warm air system and to cool the building by this method,

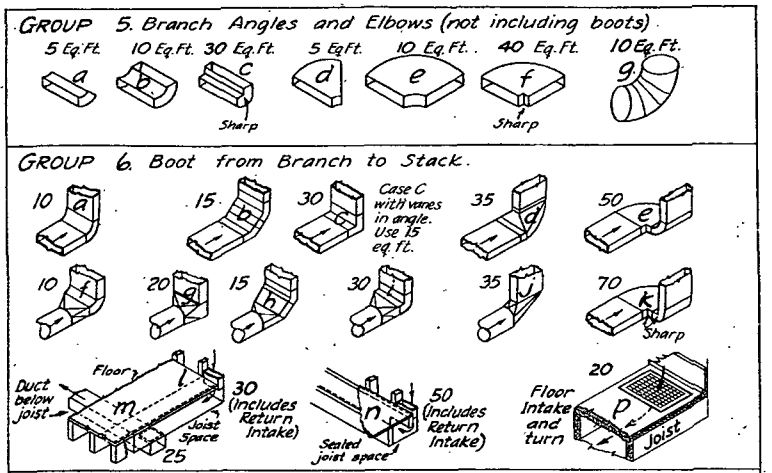


TABLE 7.
Equivalent Length for Registers.
(See diagrams at right)

Deflection Angle a.	Baseboard or Low Sidewall Register	High Sidewall Register For throw of air of: less than 13 feet	over 13 feet
0°	35 Eq Ft.	35 Eq Ft.	70 Eq Ft.
15°	40 " "	40 " "	80 " "
22°	45 " "	45 " "	90 " "
30°	60 " "	60 " "	115 " "
45°	115 " "	115 " "	230 " "

Notes for Table 7.

a) When turning vanes are used in stackhead use 0.7 of values in columns 2 to 4.

b) Values shown in 2nd and 3rd columns in Table are for 6 in. height registers (Standard height) and those in 4th column are for 4 in. height registers. For 5 in. height registers, multiply values in Col. 2 and 3 by 1.3 For 8 in. height registers, multiply values in Col. 2 and 3 by 0.8

For 2-way deflection registers, add the vertical and horizontal deflection angles together and multiply by 0.7. Select closest angle "a" in Col. 1 of Table 7

FIG. 3. EQUIVALENT LENGTH OF FITTINGS (Continued)

provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. (See also Chapters 38, 39 and 43.)

Conclusions drawn from studies[4] conducted in the University of Illinois Research Residence, subject to the limitations of the test are:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hr on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

TABLE 3. CAPACITY TABLES FOR WARM-AIR AND RETURN-AIR BRANCHES[a,b]

FIRST STORY

FOR UNINSULATED METAL DUCTS	ACTUAL LENGTH (FROM BONNET TO BOOT) OR, (FROM RETURN PLENUM TO BOOT) IN FEET								WARM AIR COMBINATION No.	RETURN AIR COMBINATION No.
	1 TO 7 FT	8 TO 12 FT	13 TO 17 FT	18 TO 24 FT	25 TO 34 FT	35 TO 44 FT	45 TO 54 FT	55 TO 64 FT		
	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G	COL. H		
Section A.	7200	6700	6100	5600	4800	4100	3500	3000	41	51
	12500	11700	10800	9900	8500	7400	6400	5500	42	52
40 to 69	16000	15000	14000	13000	11300	9900	8700	7700	43	53
Equivalent	19100	18000	17000	16000	14200	12500	11000	9600	44	54
Ft for										
Fittings	25000	23400	21600	19800	17000	14800	12800	11000	45[c]	55
and	32000	30000	28000	26000	22600	19800	17400	15400	46[c]	56
Register	80000	75000	70000	65000	56500	49500	43500	38500	....	57[c]
Section B.	5500	5100	4800	4500	3900	3400	3000	2600	41	51
	9900	9200	8600	8100	7100	6200	5400	4600	42	52
70 to 99	13100	12300	11600	10900	9700	8500	7500	6500	43	53
Equivalent	16300	15400	14500	13700	12200	10800	9500	8300	44	54
Ft										
	19800	18400	17200	16200	14200	12400	10800	9200	45[c]	55
	26200	24600	23200	21800	19400	17000	15000	13000	46[c]	56
	65500	61500	58000	54500	48500	42500	37500	42500	....	57[c]
Section C.	4400	4200	3900	3700	3200	2900	2600	2300	41	51
	8100	7600	7100	6600	5800	5100	4500	4000	42	52
100 to 139	10800	10100	9600	9000	8000	7100	6200	5400	43	53
Equivalent	13700	12900	12100	11300	10000	8900	7900	7000	44	54
Ft										
	16200	15200	14200	13200	11600	10200	9000	8000	45[c]	55
	21600	20200	19200	18000	16000	14200	12400	10800	46[c]	56
	54000	50500	48000	45000	40000	35500	31000	27000	....	57[c]
Section D.	3900	3700	3500	3300	2900	2500	2200	2000	41	51
	6900	6500	6100	5700	5100	4500	4000	3600	42	52
140 to 189	9300	8700	8100	7600	6800	6000	5300	4700	43	53
Equivalent	11600	11000	10400	9800	8700	7700	6800	6000	44	54
Ft										
	13800	13000	12200	11400	10200	9000	8000	7200	45[c]	55
	18600	17400	16200	15200	13600	12000	10600	9400	46[c]	56
	46500	43500	40500	38000	34000	30000	26500	23500	....	57[c]
Section E.	3300	3100	2900	2700	2500	2200	1900	1700	41	51
	5700	5400	5100	4800	4300	3800	3400	3000	42	52
190 to 250	7800	7300	6900	6500	5700	5000	4400	3900	43	53
Equivalent	9700	9100	8600	8100	7300	6500	5800	5100	44	54
Ft										
	11400	10800	10200	9600	8600	7600	6800	6000	45[c]	55
	15600	14600	13800	13000	11400	10000	8800	7800	46[c]	56
	39000	36500	34500	32500	28500	25000	22000	19500	.....	57[c]

FOR INSULATED DUCTS	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	FOR DUCTS THAT ARE COMPLETELY INSULATED WITH ½ IN. THICK INSULATION FROM BONNET TO BOOT, USE THESE COLUMN HEADINGS.
	1 TO 9 FT	10 TO 17 FT	18 TO 24 FT	25 TO 34 FT	35 TO 54 FT	55 TO 74 FT	

[a]These tables are for use in sizing **both the warm air and the return air** branches.

[b]Frictional resistances and temperature drops in ducts have both been accounted for in these tables.

[c]Use these items only when the building construction, or capacity requirements, necessitates the use of two adjoining stacks or floor registers.

TABLE 4. CAPACITY TABLES FOR WARM-AIR AND RETURN AIR BRANCHES[a,b]

SECOND STORY

FOR UNINSULATED METAL DUCTS	ACTUAL LENGTH (FROM BONNET TO BOOT) OR, (FROM RETURN PLENUM TO BOOT) IN FEET								WARM AIR COMBINATION No.	RETURN AIR COMBINATION No.
	1 TO 7 FT	8 TO 12 FT	13 TO 17 FT	18 TO 24 FT	25 TO 34 FT	35 TO 44 FT	45 TO 54 FT	55 TO 64 FT		
	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G	COL. H		
Section A.	6300	5700	5200	4800	4100	3500	3100	2800	41	51
	10900	10000	9200	8500	7300	6400	5600	5000	42	52
40 to 69	14000	13000	12100	11400	10000	8800	7800	6800	43	53
Equivalent	17000	15900	14900	13900	12400	11100	9900	9000	44	54
Ft for										
Fittings	21800	20000	18400	17000	14600	12800	11200	10000	45[c]	55
and	28000	26000	24200	22400	20000	17600	15600	13600	46[c]	56
Register	70000	65000	60500	57000	50000	44000	39000	34000	....	57[c]
Section B.	5000	4600	4300	4000	3400	3000	2700	2300	41	51
	9000	8200	7600	7100	6200	5400	4700	4200	42	52
70 to 99	11900	11000	10300	9600	8400	7500	6700	6000	43	53
Equivalent	14800	13900	13000	12200	10800	9600	8500	7500	44	54
Ft										
	18000	16400	15200	14200	12400	10800	9400	8400	45[c]	55
	23900	22000	20600	19200	16800	15000	13400	12000	46[c]	56
	59500	55000	51500	48000	42000	37500	33500	30000	....	57[c]
Section C.	4100	3800	3600	3400	2900	2600	2300	2000	41	51
	7400	6900	6400	6000	5200	4600	4000	3600	42	52
100 to 139	10000	9300	8700	8100	7100	6200	5500	4800	43	53
Equivalent	12500	11600	10800	10100	9000	8000	7200	6500	44	54
Ft										
	14800	13800	12800	12000	10400	9200	8000	7200	45[c]	55
	20000	18600	17400	16200	14200	12400	11000	9600	46[c]	56
	50000	46500	43500	40500	35500	31000	27500	24000	....	57[c]
Section D.	3700	3400	3100	2900	2600	2200	2000	1800	41	51
	6500	6000	5600	5300	4700	4100	3700	3300	42	52
140 to 189	8600	8000	7500	7000	6100	5400	4800	4300	43	53
Equivalent	10900	10100	9400	8800	7800	6900	6100	5400	44	54
Ft										
	13000	12000	11200	10600	9400	8200	7400	6600	45[c]	55
	17200	16000	15000	14000	12200	10800	9600	8600	46[c]	56
	43000	40000	37500	35000	30500	27000	24000	21500	....	57[c]
Section E.	3200	2900	2700	2500	2200	1900	1700	1500	41	51
	5500	5200	4800	4400	3900	3400	3100	2800	42	52
190 to 250	7300	6800	6300	5900	5100	4500	3900	3500	43	53
Equivalent	9100	8500	7900	7500	6600	5900	5200	4700	44	54
Ft										
	11000	10400	9600	8800	7800	6800	6200	5600	45[c]	55
	14600	13600	12600	11800	10200	9000	7800	7000	46[c]	56
	36500	34000	31500	29500	25500	22500	19500	17500	....	57[c]

FOR INSULATED DUCTS	COL. A	COL. B	COL. C	COL. D	COL. E	COL. F	COL. G	FOR DUCTS THAT ARE COMPLETELY INSULATED WITH ½ IN. THICK INSULATION FROM BONNET TO BOOT USE THESE COLUMN HEADINGS.
	1 TO 8 FT	9 TO 14 FT	15 TO 20 FT	21 TO 27 FT	28 TO 42 FT	43 TO 54 FT	55 TO 70 FT	

[a]These tables are to use in sizing **both the warm air and the return air** branches.

[b]Frictional resistance and temperature drops in ducts have both been accounted for in these tables.

[c]Use these items only when the building construction, or capacity requirements, necessitates the use of two adjoining stacks or floor registers.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The duct system in a forced-air heating installation can be successfully converted to a system for conveying cool air for the purpose of cooling the structure. No condensation of moisture was observed when the duct temperatures were not less than 65 F.

7. Cooling by means of water at a temperature of 60 F is not satisfactory unless an indoor temperature of less than 80 F is maintained.

8. In the selection of cooling coils, the additional frictional resistance of the coil to flow of air must be given consideration.

9. Cooling the structure by introducing large quantities of air from outdoors at night tended to reduce the amount of cooling required on the following day and was a practical means of providing more comfortable conditions in those homes where cooling systems were not available.

METHOD OF DESIGNING COOLING SYSTEM

The general procedure which may be used for the design of a summer cooling system in a forced-air installation is:

1. Calculate heat gain for each room or space to be conditioned. (See Chapters 6 and 15.) Allowance for addition of outside air must be included in this calculation.

2. Select a temperature of air leaving supply inlets. In Research Residence tests a value of from 65 to 70 F was found satisfactory.

3. Determine indoor conditions to be maintained. In Research Residence 80 F dry-bulb and 45 per cent relative humidity were found satisfactory.

4. Determine the quantity of air to be introduced into each room. (See Chapter 43.)

5. Estimate heat loss in duct system between cooling unit and supply registers.

6. Calculate the sensible and latent heat to be removed by the cooling unit.

7. Determine size of ducts in duct system and size of registers, as explained in Chapters 40 and 41.

8. Determine pressure loss in duct system and select fan as also explained in the same section.

9. Select cooling unit from manufacturer's data. Specify temperature and pressure of available cooling water, voltage and characteristics of electrical supply, and method of control of apparatus.

10. Select cooling coils from manufacturer's data to take care of latent heat load and to give required drop in air temperature with the weight of air flowing. (See Chapter 7.)

11. If system is to be used for both winter heating and summer cooling, duct sizes must be checked to insure that velocities and friction losses are reasonable for both conditions of operation. Adjustable dampers will be necessary to make changes in air distribution for the two seasons. Provision must also be made for changing fan speeds for summer and winter operation.

REFERENCES

1—Specifications for the furnace unit and the installed duct system are shown in The Yardstick (Textbook Section 8) and the Code and Manual (Textbook Section 7) published by *National Warm Air Heating and Air Conditioning Association.*

2—Performance of a Forced Warm-Air Heating System as Affected by Changes in Volume and Temperature of Air Recirculated, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 393).

3—Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 37).

4—Summer Cooling in the Research Residence, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick (University of Illinois *Engineering Experiment Station Bulletins* Nos. 290, 305 and 321). A.S.H.V.E RESEARCH REPORT No. 1177—Summer Cooling in the Research Residence with a Gas-Fired Dehydration Cooling Unit, by A. P. Kratz, S. Konzo and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 203).

CHAPTER 23

Steam Heating Systems and Piping

Classification, Piping for Steam Heating Systems, Steam Flow, Pipe Sizes, Indirect Heating Units, Types of Heating Systems, High Pressure Steam Systems, Boiler Connections, Condensation Return Pumps, Vacuum Pumps, Traps, Control Valves, Connections to Heating Units

STEAM heating systems may be classified according to any one of or combination of the following features: (1) the piping arrangement, (2) the method of returning the condensation to the boiler, (3) the accessories used, (4) the method of expelling or removing the air from the system, (5) the type of control used, and (6) the pressure or vacuum conditions obtained in operation.

In all heating systems, the condensation is returned to the boiler either by gravity or by mechanical means. In *gravity systems* the condensate is returned by gravity due to the static head of water in the return pipes or mains. The elevation of the boiler water line must be sufficiently below the lowest heating unit, steam pipe or dry return pipe to permit the return by gravity. The *water line difference* forming the *static head* must be sufficient to overcome the maximum pressure drop in the system, including the pressure drop due to the condensing effect of the radiation. When radiator and drip traps are used, as in two-pipe vapor systems, the static head must also exceed the operating pressure of the boiler. The pressure drop caused by condensing rate of the radiation is especially important during those portions of the operating periods, when changing pressure conditions prevail, as for example, when the system is being initially filled with steam. In systems where the condensate is wasted to the sewer, no water line difference is required. However, the waste of condensate may introduce conditions which warrant the use of an appropriate mechanical return system. Whenever the conditions of a heating system are such that the condensate cannot gravitate to the boiler, it must be returned by some mechanical means.

In *mechanical systems* the condensate flows to a receiver by gravity and is then forced into the boiler against its pressure. In all instances the preferable practice is to provide for gravity flow even when a vacuum pump is used. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system.

There are three general types of mechanical return devices in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return line pump.

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, as 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. Arranging the piping so the total distance from the boiler to the radiation is the same as the return piping distance from the heating unit back to the boiler tends to obtain such a result. The condensation which occurs in steam piping as well as in radiators must be drained to prevent impeding the ready flow of the steam and air. The effect of back pressure in the returns and excessive revaporization, such as occurs where condensation 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* outside 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

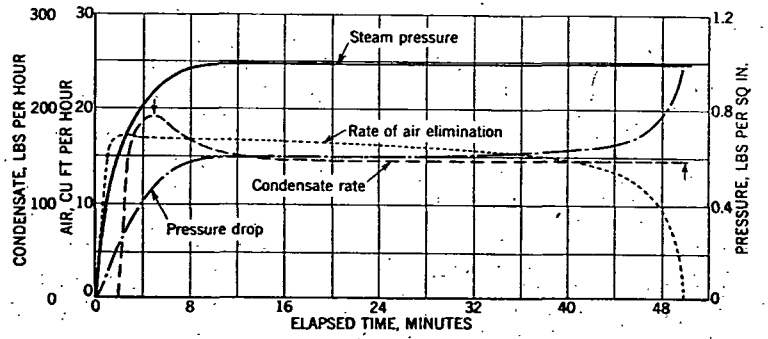


FIG. 1. RELATION BETWEEN ELAPSED TIME, STEAM PRESSURE, CONDENSATE AND AIR ELIMINATION RATES

return of condensation have revealed that as high as 143 per cent of the design condensation rate may exist under conditions of actual operation.

The piping design of a heating system is greatly influenced by its operating characteristics. Heating systems do not operate under constant conditions as they are continually changing 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 especial 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. 1 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 condensation flow during the initial warming-up period reaches a peak which is greater than the constant condensation rate eventually reached when the pressure becomes uniform. Moreover,

the peak condensation 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 has been established by Babcock in the formula given at the top of Table 1. In Columns 1, 2, 3, and 4 of this table, the numerical values of the factors for different pressure losses, pipe diameters, steam densities and lengths of pipe have been worked out in convenient form so that the steam flowing in any pipe may be calculated by multiplying together the proper factors in each column as shown in the example at the bottom of the table.

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 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 sub-atmospheric, which normally operate under controlled partial vacua, and orifice and vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; and (4) the equivalent head 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.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

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

Table 1. Flow of Steam in Pipes

P = loss in pressure in pounds per square inch.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

$$W = 5220\sqrt{\frac{PdD^5}{\left(1+\frac{3.6}{D}\right)L}}$$

$$P = 0.0000000367\left(1+\frac{3.6}{D}\right)\frac{W^2L}{dD^5}$$

Pressure Loss in Ounces	Col. 1 $5220\sqrt{\frac{P}{100}}$	Pipe Size Nominal	Pipe Size Actual Internal Diameter	Internal Area of Pipe Sq Inches	Col. 2 $\sqrt{\frac{D^5}{1+\frac{3.6}{D}}}$	Steam Press. psig	Col. 3 $\sqrt{d}$	Length of Pipe in Feet	Col. 4 $\sqrt{\frac{100}{L}}$
0.25	65.28	1	1.049	0.864	0.536	−1.0[a]	0.187	20	2.240
0.50	92.28	1¼	1.380	1.496	1.178	−0.5[a]	0.190	40	1.580
1.00	130.5	1½	1.610	2.036	1.828	0.0	0.193	60	1.290
2	184.6	2	2.067	3.356	3.710	0.3	0.195	80	1.120
3	226.0	2½	2.469	4.788	6.109	1.3	0.201	100	1.000
4	261.0	3	3.068	7.393	11.183	2.3	0.207	120	0.912
5	291.8	3½	3.548	9.887	16.705	5.3	0.223	140	0.841
6	319.7	4	4.026	12.730	23.631	10.3	0.248	160	0.793
7	345.3	4½	4.506	15.947	32.134	15.3	0.270	180	0.741
8	369.1	5	5.047	20.006	43.719	20.3	0.290	200	0.710
10	412.7	6	6.065	28.886	71.762	30.3	0.326	250	0.632
12	452.0	7	7.023	38.743	106.278	40.3	0.358	300	0.578
14	488.3	8	7.981	50.027	149.382	50.3	0.388	350	0.538
16	522.0	9	8.941	62.786	201.833	60.3	0.415	400	0.500
20	583.6	10	10.020	78.854	272.592	75.3	0.452	450	0.477
24	639.3	12	12.000	113.098	437.503	100.3	0.507	500	0.447
28	690.5	14	13.250	137.880	566.693	125.3	0.557	600	0.407
32	738.2	16	15.250	182.655	816.872	150.3	0.603	700	0.378
40	825.4					175.3	0.645	800	0.354
48	904.1					200.3	0.685	900	0.333
80	1167.2							1000	0.316
160	1650.7							1200	0.289
320	2334.5							1500	0.258
480	2859.1							2000	0.224

Column 1 × 2 × 3 × 4 = lb of steam per hour that will flow through a straight pipe for a given condition.

Example 1: 1 oz drop = 2 in. pipe = 1.3 lb press. = 100 ft equivalent length:

130.5 × 3.710 × 0.201 × 1 = 97.2 lb per hour.
97.2 × 4[b] = 388.8 sq ft equivalent radiation.

Table 1 does not allow for entrained water in low-pressure steam, condensation in covered pipe and roughness in commercial pipe as found in practice.

[a]Pounds per square inch gage = 2.04 in. Vacuum, Mercury Column.

[b]The factor 4 is the approximate equivalent in square feet of steam radiation of 1 lb of steam per hour.

factor is the velocity permissible without interfering with the condensate flow. A.S.H.V.E. Research Laboratory experiments limit this to the capacities given in Table 2 for horizontal pipes at varying grades.

Maximum Velocity

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation 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

Table 2. 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 Square Feet E D R															
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30
	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]Data from American Society of Heating and Ventilating Engineers Research Laboratory.

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.

Reaming Important

Three factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided. The second is the care used in reaming the ends of the pipe after cutting. The effect of both of these factors increases as the pipe size decreases. According to A.S.H.V.E. Research Laboratory tests, either of these factors may affect the capacity of a 1-in. pipe as much as 20 per cent. The third factor is the uniformity in grading the pipe line.

All of the capacity tables given in this chapter include a factor of safety. However, the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

Equivalent Length of Run

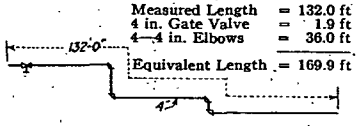
All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 3 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 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.

TABLE 3. 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	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.

Example of length in feet of pipe to be added to actual length of run.



TABLES FOR PIPE SIZING FOR LOW PRESSURE SYSTEMS[2]

Tables 4, 5, and 6 are based on the actual inside diameters of the pipe and the condensation of ¼ lb (4 oz) of steam per square foot of equivalent direct radiation (*abbreviated EDR*) per hour. The drops indicated are

TABLE 4. STEAM PIPE CAPACITIES FOR LOW PRESSURE SYSTEMS

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

PIPE SIZE IN.	CAPACITIES OF STEAM MAINS AND RISERS								SPECIAL CAPACITIES FOR ONE-PIPE SYSTEMS ONLY		
	DIRECTION OF CONDENSATION FLOW IN PIPE LINE										
	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only				
	⅛ psi or ½ Oz Drop	1/24 psi or 3/32 Oz Drop	⅛ psi or 1 Oz Drop	⅛ psi or 2 Oz Drop	¼ psi or 4 Oz Drop	½ psi or 8 Oz Drop	Vertical	Horizontal	Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[a]	*J*[b]	*K*	*L*[c]
Capacity Expressed in Square Feet E D R											
¾	.....	.....	30	.....	.....	.....	30	.....	25	.....	.....
1	39	46	56	79	111	157	56	34	45	28	28
1¼	87	100	122	173	245	346	122	75	98	62	62
1½	134	155	190	269	380	538	190	108	152	93	93
2	273	315	386	546	771	1,091	386	195	288	169	169
2½	449	518	635	898	1,270	1,800	635	395	464	.....	260
3	822	948	1,160	1,650	2,330	3,290	1,130	700	800	.....	475
3½	1,230	1,420	1,740	2,460	3,470	4,910	1,550	1,150	1,140	.....	745
4	1,740	2,010	2,460	3,480	4,910	6,950	2,040	1,700	1,520	.....	1,110
5	3,210	3,710	4,550	6,430	9,090	12,900	.....	3,150	.....	.....	2,180
6	5,280	6,100	7,460	10,550	14,900	21,100	.....	.....	.....	.....	.....
8	11,000	12,700	15,500	21,970	31,070	43,900	.....	.....	.....	.....	.....
10	20,000	23,100	28,300	40,100	56,700	80,200	.....	.....	.....	.....	.....
12	32,000	37,100	45,500	64,300	91,000	129,000	.....	.....	.....	.....	.....
16	61,000	69,700	84,800	121,000	170,000	242,000	.....	.....	.....	.....	.....
Capacity Expressed in Pounds per Hour											
¾	.....	.....	8	.....	.....	.....	8	.....	6	.....	7
1	10	12	14	20	28	40	14	9	11	7	7
1¼	22	25	31	43	61	87	31	19	20	16	16
1½	34	39	48	67	95	135	48	27	38	23	23
2	68	79	97	137	193	273	97	49	72	42	42
2½	112	130	159	225	318	449	159	99	116	.....	65
3	206	237	291	411	581	822	282	175	200	.....	119
3½	307	355	434	614	869	1,230	387	288	286	.....	186
4	435	503	614	869	1,230	1,740	511	425	380	.....	278
5	806	928	1,140	1,610	2,270	3,210	.....	788	.....	.....	545
6	1,320	1,520	1,870	2,640	3,730	5,280	.....	.....	.....	.....	.....
8	2,750	3,170	3,880	5,490	7,770	11,000	.....	.....	.....	.....	.....
10	5,010	5,790	7,090	10,000	14,200	20,000	.....	.....	.....	.....	.....
12	8,040	9,290	11,400	16,100	22,700	32,200	.....	.....	.....	.....	.....
16	15,100	17,400	21,200	30,300	42,400	60,500	.....	.....	.....	.....	.....
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Un-dripped Run-outs	Up-Feed Risers	Radiator Con-nections	Run-outs Not Dripped

Note.—All drops shown are in psi per 100 ft of equivalent run—based on pipe properly reamed.

[a]Do not use Column *H* for drops of 1/24 or 1/32 psi; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 psi except on sizes 3 in. and over; below 3 in. substitute Column *B*.

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 4.

TABLE 5. RETURN PIPE CAPACITIES FOR LOW PRESSURE SYSTEMS

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pipe Size Inches	CAPACITY OF RETURN MAINS AND RISERS																				
	MAINS																				
	1/32 Psi or 1/16 Oz Drop per 100 Ft			1/24 Psi or 3/32 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			1/4 Psi or 4 Oz Drop per 100 Ft			1/2 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.			
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*			
3/4	.....	.....	.....	.....	.....	326	.....	.....	400	.....	.....	568	.....	.....	800	.....	.....	1,130			
1	500	248	.....	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	.....	.....	1,980			
1 1/4	850	520	.....	990	595	976	1,200	670	1,200	1,700	868	1,700	2,400	962	2,400	.....	.....	3,390			
1 1/2	1,350	822	.....	1,570	943	1,550	1,900	1,060	1,900	2,700	1,360	2,700	3,800	1,510	3,800	.....	.....	5,370			
2	2,800	1,880	.....	3,240	2,140	3,260	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	.....	.....	11,300			
2 1/2	4,700	3,040	.....	5,300	3,470	5,450	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	.....	.....	18,900			
3	7,500	5,840	.....	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,200	21,400	10,000	21,400	.....	.....	30,200			
3 1/2	11,000	7,880	.....	13,200	8,800	13,000	16,000	10,000	16,000	22,000	12,900	22,700	32,000	14,300	32,000	.....	.....	45,200			
4	15,500	11,700	.....	18,300	13,400	18,000	22,000	15,000	22,000	31,000	19,300	31,200	44,000	21,500	44,000	.....	.....	62,190			
5	.....	.....	.....	.....	.....	31,500	.....	.....	38,700	.....	.....	54,900	.....	.....	77,400	.....	.....	109,000			
6	.....	.....	.....	.....	.....	50,450	.....	.....	62,000	.....	.....	88,000	.....	.....	124,000	.....	.....	175,000			
	RISERS																				
3/4	.....	190	.....	.....	190	570	.....	190	700	.....	190	994	.....	190	1,400	.....	.....	1,980			
1	.....	450	.....	.....	450	976	.....	450	1,200	.....	450	1,700	.....	450	2,400	.....	.....	3,390			
1 1/4	.....	990	.....	.....	990	1,550	.....	990	1,900	.....	990	2,700	.....	990	3,800	.....	.....	5,370			
1 1/2	.....	1,500	.....	.....	1,500	3,260	.....	1,500	4,000	.....	1,500	5,680	.....	1,500	8,000	.....	.....	11,300			
2	.....	3,000	.....	.....	3,000	5,450	.....	3,000	6,700	.....	3,000	9,510	.....	3,000	13,400	.....	.....	18,900			
2 1/2	.....	.....	.....	.....	.....	8,710	.....	.....	10,700	.....	.....	15,200	.....	.....	21,400	.....	.....	30,200			
3	.....	.....	.....	.....	.....	13,000	.....	.....	16,000	.....	.....	22,700	.....	.....	32,000	.....	.....	45,200			
3 1/2	.....	.....	.....	.....	.....	17,900	.....	.....	22,000	.....	.....	31,200	.....	.....	44,000	.....	.....	62,200			
4	.....	.....	.....	.....	.....	31,500	.....	.....	38,700	.....	.....	54,900	.....	.....	77,400	.....	.....	109,000			
5	.....	.....	.....	.....	.....	50,500	.....	.....	62,000	.....	.....	88,000	.....	.....	124,000	.....	.....	175,000			

TABLE 6. RETURN PIPE CAPACITIES FOR LOW PRESSURE SYSTEMS

Capacity Expressed in Pounds per Hour

(Reference to this table will be made by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pipe Size Inches	CAPACITY OF RETURN MAINS AND RISERS																				
	MAINS																				
	1/32 Psi or 1/16 Oz Drop per 100 Ft			1/24 Psi or 3/32 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			1/4 Psi or 4 Oz Drop per 100 Ft			1/2 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.			
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*			
3/4	.....	.....	.....	.....	.....	42	.....	.....	100	.....	.....	142	.....	.....	200	.....	.....	283			
1	125	62	.....	145	71	143	175	80	175	250	103	249	350	115	350	.....	.....	494			
1 1/4	213	130	.....	248	149	244	300	168	300	425	217	426	600	241	600	.....	.....	848			
1 1/2	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			
2 1/2	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			
3 1/2	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																				
3/4	.....	48	.....	.....	48	143	.....	48	175	.....	48	249	.....	48	350	.....	.....	494			
1	.....	113	.....	.....	113	244	.....	113	300	.....	113	426	.....	113	600	.....	.....	848			
1 1/4	.....	248	.....	.....	248	388	.....	248	475	.....	248	674	.....	248	950	.....	.....	1,340			
1 1/2	.....	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			
2 1/2	.....	.....	.....	.....	.....	2,180	.....	.....	2,680	.....	.....	3,800	.....	.....	5,350	.....	.....	7,560			
3	.....	.....	.....	.....	.....	3,250	.....	.....	4,000	.....	.....	5,680	.....	.....	8,000	.....	.....	11,300			
3 1/2	.....	.....	.....	.....	.....	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			

drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed and without unusual or noticeable defects.

Table 4 may be used for sizing piping for steam heating systems by pre-determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G*, inclusive, are used where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

Return piping may be sized with the aid of Tables 5 and 6 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 4. *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-psi gage?

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 1,000 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 1,000 ft, the drop per 100 ft would be $\frac{1}{10}$ psi, while if the total drop were $\frac{1}{2}$ psi, the drop per 100 ft would be $\frac{1}{20}$ psi. In the first instance the pipe could be sized according to Column *D* for $\frac{1}{16}$ psi per 100 ft, and in the second case, the pipe could be sized according to Column *C* for $\frac{1}{24}$ psi. 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.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_1 - t_e)}{55.2 \times 240} = \frac{Q \times (t_1 - t_e)}{220.8} \qquad (1)$$

where

EDR = equivalent direct radiation, square feet.
Q = volume of air, cubic feet per minute.
t_e = the temperature of the air entering the row of heating units under consideration, Fahrenheit degrees.
t_1 = the temperature of the air leaving the row of heating units under consideration, Fahrenheit degrees.
60 = the number of minutes in one hour.
55.2 = the number of cubic feet of air heated 1 Fahrenheit degree by 1 Btu.
240 = Btu equivalent of 1 sq ft of EDR.

Example 3. Assume that a 3-row heating unit shown in this chapter in Fig. 38 is handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

Solution. For row 1,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For row 2,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For row 3,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	Total Load (EDR)	Stack Load[a] (EDR)	Connection Load[b] (EDR)
1	9058	2265	2265 or 1132
2	5661	1415	1415 or 708
3	3397	849	849 or 425

[a]One quarter of total row load.
[b]One half of stack load if two steam connections are made; otherwise, same as stack load.

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system.

Most manufacturers of heating coils now publish condensing capacities of their coils for all common operating conditions in pounds of steam per hour per square foot of face area (or free area for some designs). These rates may be used in connection with Tables 4 and 6 to size the piping.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

This system is the most common of all methods of steam heating, especially for small size installations, due largely to its low cost and simplicity.

The downward pitch of a one-pipe air-vent system is indicated in Fig. 2. Low points and ends of steam mains pitched down from the boiler should be dripped. All drips should be sealed below water line before connecting together. In the risers and radiator connections, steam and condensation flow in opposite directions. In long steam mains it flows in the same direction as the steam and is removed from the main through the drip. Short mains may be arranged for the condensate to flow in a direction opposite the steam by sizing them so the critical velocity is not exceeded. It is customary to drip the heel of each riser in buildings of several stories to avoid counter-flow of the steam and condensate in the riser runout. In buildings of one or two stories the condensate is returned to the steam main instead of being dripped. Both types of risers are shown in Fig. 2, and riser connections are shown in Figs. 3 and 4. A typical overhead down-feed system is illustrated in Fig. 5. While wet

return mains need not be pitched toward the boiler to maintain steam circulation, they should be pitched for drainage.

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. It is desirable to install the air-vent valves about a foot ahead of the drips, as indicated in Fig. 2, to prevent possible damage to their mechanisms by water.

The radiator valves may be the angle-globe, offset-corner pattern or gate type. Straight-globe and straight-corner types should not be used since the raised valve seat would interfere with the flow of condensation through the valve. Graduated valves cannot be used since the steam valves on this system must be fully open or fully closed to prevent the radiators filling with water and creating a dangerous water line condition.

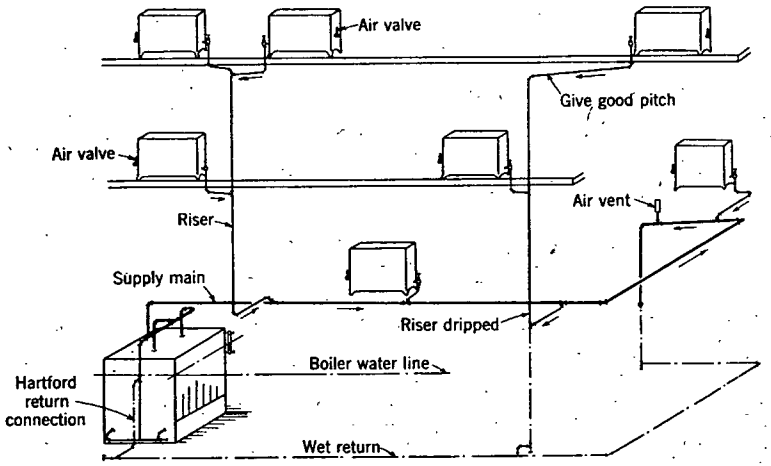


FIG. 2. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

With a one-pipe system the heat cannot be modulated at the radiator, the steam being either *all on* or *all off*. Systems and devices are available which make it possible to obtain a partial modulating effect from one-pipe heating systems.

It is important to keep the lowest points of the steam mains and heating units sufficiently above the water line of the boiler to prevent flooding. The minimum water line difference depends on the initial steam pressure and piping pressure drop plus a safety factor for heating up.

Referring to Fig. 6 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 $\frac{1}{8}$ 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 $\frac{1}{8}$ of 28 in. (28 in.

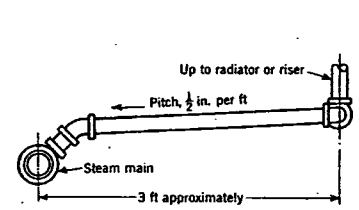


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

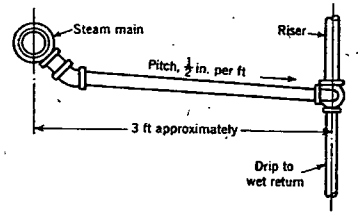


FIG. 4. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

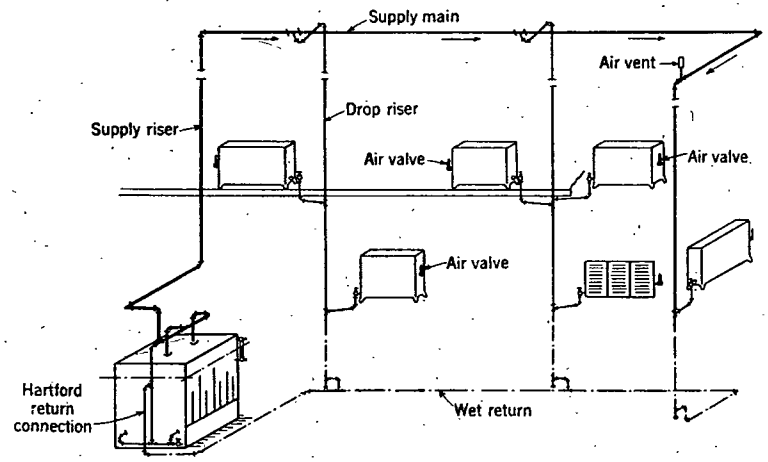


FIG. 5. TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

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 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, 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.

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 4, 5 and 6 as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use $\frac{1}{16}$-psi drop (Column *D*).
2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.
3. *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.
4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.
5. *For the radiator valve size and the stub connection*, use Column *K*.
6. For the *dry return main*, use Column *U*.
7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over $\frac{1}{4}$ psi. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where $\frac{1}{24}$-psi drop is being used, the steam main and dripped runouts would be sized from Column *C*; radiator runouts and undripped riser runouts from Column *L*; up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of $\frac{1}{24}$ psi); the dry return from Column *R*; and the wet return from Column *Q*.

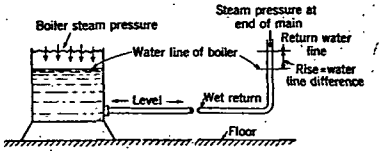


FIG. 6. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

With a $\frac{1}{32}$-psi drop the sizing would be the same as for $\frac{1}{24}$ psi except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should not be less than $\frac{1}{4}$ in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should not be less than $\frac{1}{2}$ in. in 10 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 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 4. Size the one-pipe gravity steam system shown in Fig. 7 assuming that this is all there is to the system or that the riser and run 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 $\frac{1}{4}$ psi the drop per 100 ft will be slightly less than $\frac{1}{16}$ psi. It would be well in this case to use $\frac{1}{24}$ psi, and this would result in the theoretical sizes indicated in Table 7. These theoretical sizes, however, should be modified by not using a wet return 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.

TABLE 7. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 7

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

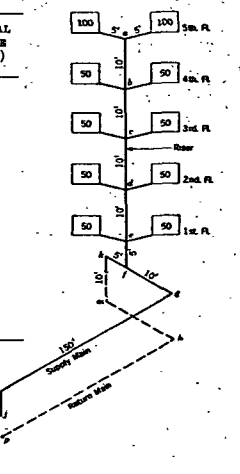


FIG. 7. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

ONE-PIPE VAPOR SYSTEM

The one-pipe vapor system operates under pressures at or near atmospheric and returns its condensation to the boiler by gravity. In this

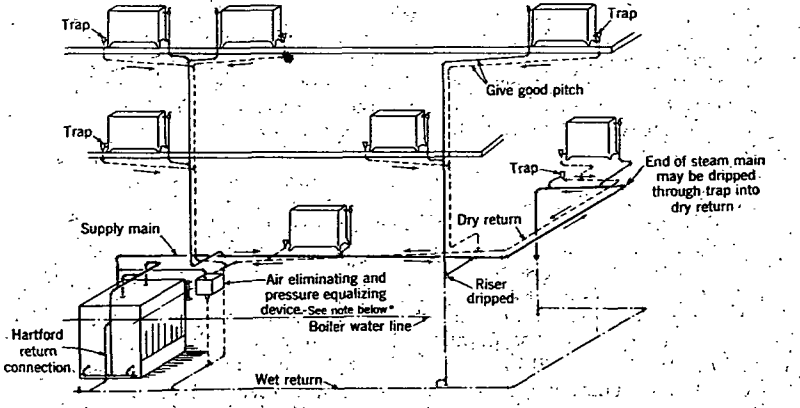


FIG. 8. TYPICAL UP-FEED SYSTEM WITH AUTOMATIC RETURN TRAP*

*Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

system the automatic air valves are of special design to permit the ready release of air and prevent its ready return after it is expelled. The steam radiator valves are a type which, when opened, give a free and unobstructed passageway for water. The piping is the same as for the one-pipe gravity system but sized so as to permit operation at a few ounces pressure.

TWO-PIPE VAPOR SYSTEM

A two-pipe up-feed vapor system using separate supply and return pipes is shown in Fig. 8. The radiators discharge their condensation through thermostatic traps to the dry return pipe. These systems operate at a few ounces pressure and above, but those with mechanical condensate return devices may operate at pressures upward of 10 psi. The simplest

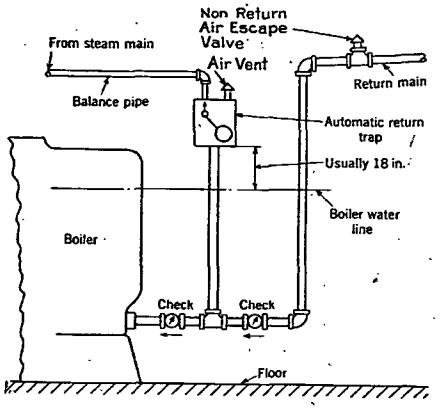


FIG. 9. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

method of venting the system consists of a ¾-in. pipe with a check valve opening outward. Most systems employ various forms of vent valves, designed to allow the air to readily pass out of the system and to prevent its return. These systems permit control of the heat in the radiator by varying the opening of the graduated radiator valves. The boiler pressure is maintained at substantially constant pressure slightly above atmospheric pressure.

These systems may be classified as (1) *closed systems*, consisting of those which have a device to prevent the return of air after it has once been expelled from the system, and which can operate at both super and subatmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems*, comprising those which have the return line constantly open to the atmosphere without a check or other means to prevent the return of air. The open systems are not so popular because they have the disadvantage of not holding heat when the rate of steam generation is diminishing.

Closed systems should preferably be equipped with an automatic return trap to prevent water from backing out of the boiler. In installing the return trap a check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap,

which is usually located with the bottom about 18 in. above the boiler water line. Some traps are constructed so that they will operate when they are installed with their bottom as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler. Fig. 9 shows a typical connection for an automatic return trap.

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The runouts are taken off the main from the bottom or at a 45-deg angle downward, and sloped toward the drops. Thus each runout from the main forms a drip and no accumulation of water is carried down any one drop.

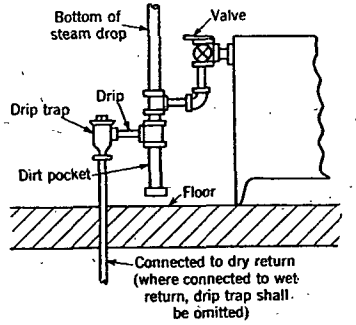


FIG. 10. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small and the normal size of drop required is 1 in. or less. The bottom of each steam drop should terminate with a dirt pocket and be dripped as shown in Fig. 10. The returns on a down-feed vapor system are the same as on an up-feed system. The runouts to the radiators and the radiator connections of the down-feed system are the same as those for the up-feed system already described.

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 condensation 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) because with large variations in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where 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 from Column *D*, Table 4, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Tables 5 and 6, Column *U*, (lower portion) and the mains from Column *U* (upper portion). 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.

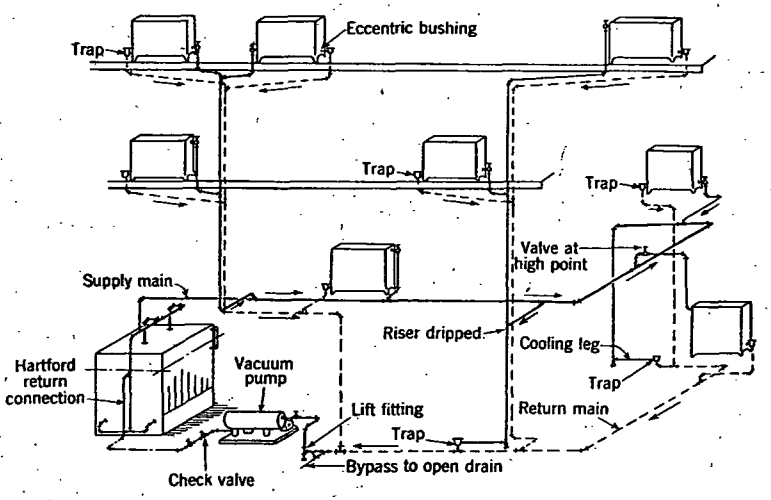


FIG. 11. TYPICAL UP-FEED VACUUM PUMP SYSTEM

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* for small systems correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed ⅛ psi to ¼ psi, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over ⅛ psi divided by 4, or 1/32 psi. In this case the steam mains would be sized from Column *B*, the radiator and undripped riser runouts from Column *I*; the risers from Column *B*, because Column *H* gives a drop in excess of 1/32 psi. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over 1/32 psi. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor 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. in 10 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 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.

VACUUM SYSTEMS

In the vacuum system, a vacuum is maintained in the return line practically at all times. The pump is usually controlled by a vacuum regulator which operates the pump to maintain the vacuum within limits and operates in response to a pressure difference between the atmosphere and the return to control the vacuum in the return main. The source of steam supply may be a low pressure boiler as shown in Fig. 11, or a high pressure line through a pressure reducing valve. The piping and other details are the same as for the vapor systems.

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 expells 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 desirable practice demands a return flowing to the vacuum pump by an uninterrupted downward slope. In some instances local conditions make it necessary to drop the return below the level of the vacuum pump inlet, before the pump can be reached. In such an event one of the advantages of the vacuum system is the ability to raise the condensate to a considerable height by the suction of the vacuum pump by means of a lift connection or fitting inserted in the return. The height the condensate can be raised depends on the amount of vacuum maintained. It is preferable to limit lift connections to a single lift at the vacuum pump. A still more preferable arrangement is the use of an accumulator tank, or receiver tank, with a float control for the pump, at the low point of the return main located adjacent to the vacuum pump.

When the vertical lift is considerable, several lift fittings should be used in steps as shown in Fig. 12. This permits a given lift to be secured with a somewhat lower vacuum than where the vertical distance is served by a single lift. Where several lifts are present in a given system at different locations, the lifting cannot occur until the entire system is filled with steam. A lift connection for location close to the pump, where the size may be above the commercial stock sizes, is shown in Fig. 13. It is desirable that means be provided for manually draining the low point of the lift fittings to eliminate danger of freezing.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return

side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45-deg angle down from the steam main so that they may serve as steam main drips. When this is done it is practical to run the steam main level, if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 14). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 15.

Vacuum, atmospheric, sub-atmospheric and orifice systems are usually employed in large installations and have total drops varying from ¼ to ½ psi. Systems where 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 1200 ft longest equivalent length of run would employ a drop per 100 ft of ½ psi divided by 12, or ½₄ psi. In this case, the steam main would be sized from Column *C*, Table 4, and the risers also from Column

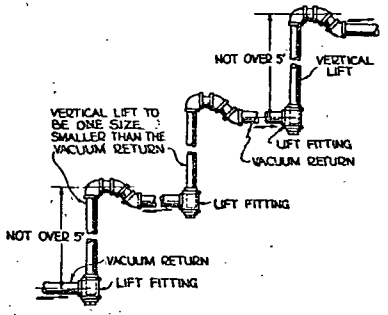


FIG. 12. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

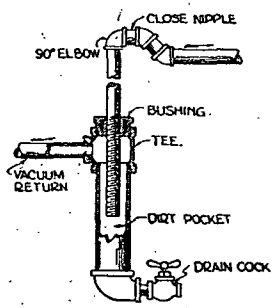


FIG. 13. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP

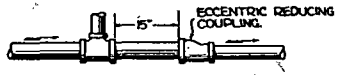


FIG. 14. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

C (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of ½₄ psi). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*, Tables 5 and 6; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed ⅛ psi drop per 100 ft of equivalent run nor to exceed 1 psi total pressure drop in any system.

2. Pitch of mains should not be less than ¼ in. in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft. Where this pitch cannot be obtained runouts over 8 ft in length should be one size larger than called for in the table.

4. In general it is not considered desirable to have a supply main smaller than 2 in.

5. 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.

6. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in this chapter.

7. No lifts can be used in orifice and atmospheric systems. In sub-atmospheric systems the lift must be at the vacuum pump.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems but, in contrast, provide control of building temperature by variation of the heat

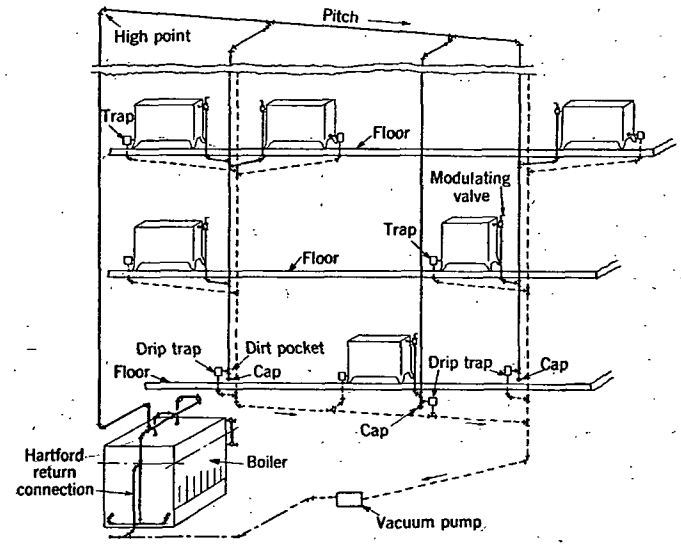


FIG. 15. TYPICAL DOWN-FEED VACUUM SYSTEM

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 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 sub-atmospheric 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 sub-atmospheric pressures by continuous flow into the heating main. All radiator supply valves have incorporated 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. These orifices are omitted on some systems, depending upon the type of control. Radiator traps and drips are designed to operate at any pressure from 15 lb gage to 26 in. Hg. A vacuum pump capable of operating at high vacuum is preferable to promote accuracy in the distribution of steam throughout the system, particularly in mild weather. This vacuum is partially self-induced by the condensation of the steam in the system under conditions of restricted supply for reduction of the radiator heat emission.

The returns must grade downward constantly and uninterruptedly from the radiator return outlets to the inlet of the receiver of the vacuum pump. One radical difference between this and the ordinary vacuum system is that no lifts should be made in the return line, except at the vacuum pump. The receivers are placed at a lower level than the pump and equipped with float control so that the pump may operate as a return pump under night conditions. The system may be operated in the same manner as the ordinary vacuum system when desired.

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 sub-atmospheric method of 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 by-pass around the heaters. Some applications of sub-atmospheric systems are proprietary.

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 condensation pump with receiver vented to atmosphere, a return line vacuum pump, or a return trap, is generally used to return the condensation 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 based on the fact that the steam flow through an orifice will vary when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent 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 psi gage on one side and ¼ psi gage on the other, the absolute pressure relation is:

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 0.90 \text{ or } 90 \text{ per cent.}$$

Should the steam pressure be dropped to ¼ lb 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 may be controlled sufficiently close. 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, which maintains a deter-

TABLE 8. ORIFICE CAPACITIES FOR LOW PRESSURE STEAM SYSTEMS

This table is based on data from actual tests*

Orifice Diameter 64ths of an Inch	6 in. Hg Differential	5 in. Hg Differential	4 in. Hg Differential	2 in. Hg Differential	1 in. Hg Differential
	Capacity Expressed in Square Feet E D R				
7	18–23	16–21	15–19	10–13	
8	23–29	21–27	19–25	13–17	8–11
9	29–36	27–33	25–30	17–21	11–14
10	36–44	33–40	30–37	21–26	14–17
11	44–52	40–48	37–44	26–31	17–20
12	52–62	48–57	44–51	31–37	20–24
13	62–72	57–66	51–59	37–43	24–28
14	72–83	66–76	59–67	43–49	28–32
15	83–94	76–86	67–76	49–56	32–37
16	94–106	86–97	76–86	56–64	37–42
17	106–119	97–109	86–97	64–72	42–47
18	119–133	109–122	97–108	72–80	47–52
19	133–148	122–135	108–120	80–88	52–58
20	148–163	135–149	120–133	88–98	58–64
21	163–179	149–164	133–145	98–107	64–71
	Capacity Expressed in Pounds per Hour				
7	4.5–5.8	4.0–5.3	3.8–4.8	2.5–3.3	
8	5.8–7.3	5.3–6.8	4.8–6.3	3.3–4.3	2.0–2.8
9	7.3–9.0	6.8–8.3	6.3–7.5	4.3–5.3	2.8–3.5
10	9.0–11.0	8.3–10.0	7.5–9.3	5.3–6.5	3.5–4.3
11	11.0–13.0	10.0–12.0	9.3–11.0	6.5–7.8	4.3–5.0
12	13.0–15.5	12.0–14.3	11.0–12.8	7.8–9.3	5.0–6.0
13	15.5–18.0	14.3–16.5	12.8–14.8	9.3–10.8	6.0–7.0
14	18.0–20.8	16.5–19.0	14.8–16.8	10.8–12.3	7.0–8.0
15	20.8–23.5	19.0–21.5	16.8–19.0	12.3–14.0	8.0–9.3
16	23.5–26.5	21.5–24.3	19.0–21.5	14.0–16.0	9.3–10.5
17	26.5–29.8	24.3–27.3	21.5–24.3	16.0–18.0	10.5–11.8
18	29.8–33.3	27.3–30.5	24.3–27.0	18.0–20.0	11.8–13.0
19	33.3–37.0	30.5–33.8	27.0–30.0	20.0–22.0	13.0–14.5
20	37.0–40.8	33.8–37.3	30.0–33.3	22.0–24.5	14.5–16.0
21	40.8–44.8	37.3–41.0	33.3–36.3	24.5–26.8	16.0–17.8

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.

*Flow of steam through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 371).

TABLE 9. STEAM PIPE CAPACITIES FOR 30 PSI STEAM SYSTEMS

Capacity Expressed in Pounds per Hour

(Steam and Condensate Flowing in Same Direction)

PIPE SIZE INCHES	DROP IN PRESSURE—POUNDS PER 100 FT IN LENGTH					
	⅛	¼	½	¾	1	2
¾	15	22	31	38	45	63
1	31	46	63	77	89	125
1¼	69	100	141	172	199	281
1½	107	154	219	267	309	437
2	217	313	444	543	627	886
2½	358	516	730	924	1,033	1,460
3	651	940	1,330	1,628	1,880	2,660
3½	979	1,414	2,000	2,447	2,825	4,000
4	1,386	2,000	2,830	3,464	4,000	5,660
5	2,560	3,642	5,225	6,402	7,390	10,460
6	4,210	6,030	8,590	10,420	12,140	17,180
8	8,750	12,640	17,860	21,865	25,250	35,100
10	16,250	23,450	33,200	40,625	46,900	66,350
12	25,640	36,930	52,320	64,050	74,000	104,500

mined pressure by varying the vacuum in the return lines, or by varying the pressure in the supply lines and the vacuum in the returns. The valves are frequently manually set 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 outside and inside temperatures. Since the range through which the pressures may be varied is usually from 0 to 4 psi gage, the control should be capable of maintaining close regulation to maintain the desired space temperatures, particularly in mild weather.

A recommended orifice schedule is shown in Table 8. 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

TABLE 10. STEAM PIPE CAPACITIES FOR 150 PSI STEAM SYSTEMS

Capacity Expressed in Pounds per Hour

(Steam and Condensate Flowing in Same Direction)

PIPE SIZE INCHES	DROP IN PRESSURE—PSI PER 100 FT IN LENGTH						
	⅛	¼	½	¾	1	2	5
¾	29	41	58	71	82	116	184
1	58	82	117	143	165	233	369
1¼	130	185	262	320	370	523	827
1½	203	287	407	497	575	813	1,290
2	412	583	825	1,010	1,167	1,650	2,600
2½	683	959	1,359	1,650	1,920	2,710	4,290
3	1,237	1,750	2,476	3,020	3,500	4,940	7,820
3½	1,855	2,626	3,715	4,550	5,250	7,420	11,740
4	2,625	3,718	5,260	6,430	7,430	10,500	16,620
5	4,858	6,875	9,725	11,900	13,750	19,450	30,750
6	7,960	11,275	15,950	19,500	22,550	31,850	50,400
8	16,590	23,475	33,200	40,600	46,950	66,400	105,000
10	30,820	43,430	61,700	75,600	87,250	123,400	195,000
12	48,600	68,750	97,250	119,000	137,600	194,400	307,400

TABLE 11. RETURN PIPE CAPACITIES FOR 30 PSI STEAM SYSTEMS

Capacity Expressed in Pounds per Hour

PIPE SIZE INCHES	DROP IN PRESSURE—POUNDS PER 100 FT IN LENGTH				
	⅛	¼	½	¾	1
¾	115	170	245	308	365
1	230	340	490	615	730
1¼	485	710	1,025	1,285	1,530
1½	790	1,155	1,670	2,100	2,500
2	1,575	2,355	3,400	4,300	5,050
2½	2,650	3,900	5,600	7,100	8,400
3	4,850	7,100	10,250	12,850	15,300
3½	7,200	10,550	15,250	19,150	22,750
4	10,200	15,000	21,600	27,000	32,250
5	19,000	27,750	40,250	55,500	60,000
6	31,000	45,500	65,500	83,000	98,000

pressure in the steam main may be compensated to a great extent. For example, if the initial pressure was 3 psi gage and the pressure at the end of the main was 2 psi, 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.

HIGH PRESSURE STEAM SYSTEMS

Many of the recent 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 fan outlet.

Tables 9 to 12 may be used for the sizing of steam and return piping for systems of 30 and 150 psi pressure at various pressure drops. These

TABLE 12. RETURN PIPE CAPACITIES FOR 150 PSI STEAM SYSTEMS

Capacity Expressed in Pounds per Hour

PIPE SIZE INCHES	DROP IN PRESSURE—PSI PER 100 FT IN LENGTH					
	⅛	¼	½	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

tables are based on Babcock's formula, and have been used as the basis of design for a number of years. Capacities at other pressures may be computed by means of Table 1.

The steam lines can be sized for greater pressure drops than the return piping. For a system using steam at 30 psi pressure, the total pressure drop can be 5 to 10 psi, and for 150-psi systems 25 to 50 psi.

It has been observed that the maximum pressure in the returns of a 30-psi system is about 5 psi, and that of a 150-psi system is about 20 psi. The pressure in the return mains is, of course, caused by the discharge of traps, or by leaky traps and by flashing due to the lower pressure in the return line. The total pressure drop in the returns can be approximately 2 psi for the 30-psi returns, and about 10 psi for the 150-psi returns. 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-psi systems, and 1 psi per 100 ft for 150-psi systems. This is an average figure which corresponds generally to several of the previously published tables for the design of high pressure return piping.

The returns generally discharge to a vented receiver. It is, of course, necessary to provide for the elimination of air from high pressure systems, the same as in low pressure systems. If possible, it is desirable to design the system so as to condense some of the steam escaping through the vents by passing it through a heat exchanger.

When high pressure steam is being supplied and lower steam pressures are required for heating, for domestic hot water, for utility services, etc., 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 the low pressure building up at time of no load, and the other where the low pressure lines will condense enough steam to offset normal leaking through the valve. In the latter case, double seated valves may be used in a manner that reduces the work required of the diaphragm in closing the valve and consequently the size of the diaphragm. These valves also control the low pressures more closely under conditions of varying high pressures.

Valves that shut off all steam are called *dead end* type. They are single seated, and some of them have pilot operation that provides 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 should be used.

It is common practice when the initial steam pressure is 100 psi or higher to install two-stage reduction. If the radiation served is cast-iron, the *A.S.M.E.* code requires two reducing valves when the inlet pressure exceeds 50 psi. This makes a quieter condition of steam flow, as it is apparent that with one reduction, as for example from 150 to 2 psi, there is a smaller opening with greater velocity across the reducing valve and, consequently, more noise. A two-stage reduction also introduces a source of safety, since if one reducing valve were to build up its discharge pressure, this excess pressure would not be so great as the case might be in a one-stage reduction.

If an installation requires single seated valves and the pilot type cannot be used, it is necessary to use two-stage reduction, as single seated valves require sufficient diaphragm area to overcome the unbalanced pressure underneath the single valve. In many cases the large diameter

of diaphragm required would make it impractical in construction. With a two-stage reduction the diaphragm diameter required would be reduced. If a one-stage reduction is desired, it is necessary to use a pilot controlled pressure reducing valve, where low pressures are to be maintained closely.

In making a two-stage reduction, allowance for expansion of steam on the low pressure side of the valve should be made by increasing the pipe size. This also allows steam flow to be at a more nearly uniform velocity. Separating the valves by a distance up to 20 ft is recommended to reduce excessive hunting action of the first valve.

When the reduced pressure is approximately 15 psi or lower, the weight and lever diaphragm valve gives the best results with minimum maintenance. Above 15 psi, spring loaded diaphragm valves should be used, because of the extra weights required on weight and lever type. Pressure equalizing lines should not be connected too close to the valve. They should be connected into the bottom of the reduced pressure steam main, to allow maximum condensation to exist in the equalizing lines, or the connection can be made into the top of the main if a water accumulator is used to reduce the variation of the head of water on the diaphragm.

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 bring about wire-drawing of valve and seat, due to small lift of the valve seat.

On installations where the steam requirements are relatively large and variable in mild weather or reduced demand periods, wire-drawing may occur. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30 per cent proportion of maximum flow. For example, if 50,000 lb of steam per hour are required, the size of one valve is on the basis of $0.7 \times 50{,}000$ lb, or 35,000 lb, and the other on the basis of $0.3 \times 50{,}000$ lb, or 15,000 lb. During the mild or reduced demand periods, steam will flow through the smaller valve only. During the remainder of the season, the larger valve is set to control at whatever low pressure is desired, and the smaller one at a somewhat lower pressure. Thus, when steam flow is not at its maximum, the smaller valve is closed, but it opens automatically when the maximum steam demand occurs, because this maximum demand creates a slight pressure drop in the service line.

The installation of reducing valves in pipe lines requires detailed planning. They should be installed to give ease of access for inspection and repair, and wherever possible with diaphragm downward, except in cases of pilot operated valves.

There should be a by-pass around each reducing valve of size equal to one-half the size of reducing valve. The globe valve in by-pass line should be of a better 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. Safety valves located on the low pressure side should be set 5 psi higher than the final pressure but may be 10 psi higher than the reduced pressure if this reduced pressure is that of the first stage reduction of a double reduction. Strainers are sometimes installed on the inlet to the reducing valve but are not required before a second-stage reduction. 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 also. In sizes 3 in. and above, it is advisable to install a drip trap between the two reducing valves.

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 water being carried over into the steam main.

Return

Cast-iron boilers are generally provided with return tappings on both sides, while steel boilers are generally equipped with only one return

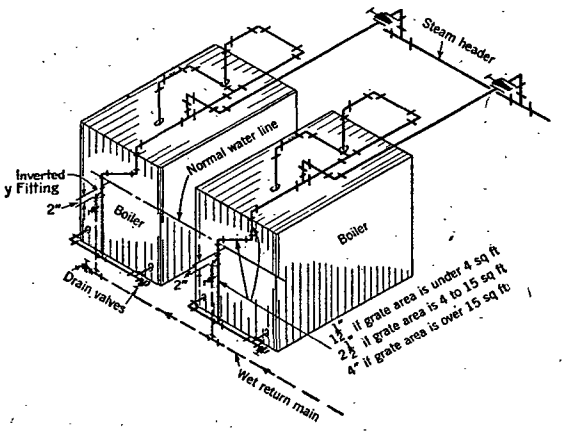


FIG. 16. THE HARTFORD RETURN CONNECTION

tapping. Where two tappings are provided, both should be used to effect proper circulation through the boiler. The return connection should include either a Hartford return connection or a check valve to prevent the accidental loss of boiler water to the returns with consequent danger of boiler damage. The Hartford return connection is to be preferred over the check valve because the latter is apt to stick or not close tightly and, furthermore, because the check valve offers additional resistance to the condensate coming back to the boiler, which in gravity systems would raise the water line in the far end of the wet return several inches.

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford return connection is recommended. This connection for a one- or two-boiler installation is shown in Fig. 16. The essential features of construction of a Hartford return connection are: (1) a direct connection (made without valves) between the steam side of the boiler and the return side of the boiler, and (2) a close nipple, or preferably an inverted Y-fitting connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return pressure balance connection. Equalizing pipe connections between the

steam and return are given in Fig. 16, based on grate areas, but in no case shall this pipe size be less than the main return piping from the system.

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 runout 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 operating, 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.

CONDENSATION RETURN PUMPS

Condensation return pumps are used for gravity systems when the local conditions do not permit the condensation to return to the boiler under the existing static head. The return of the condensate permits the water to repeatedly go 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 condensation is wasted and fresh make-up water is continually being introduced.

The most generally accepted condensation 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 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 3 to 4 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 condensation unit is illustrated in Fig. 17.

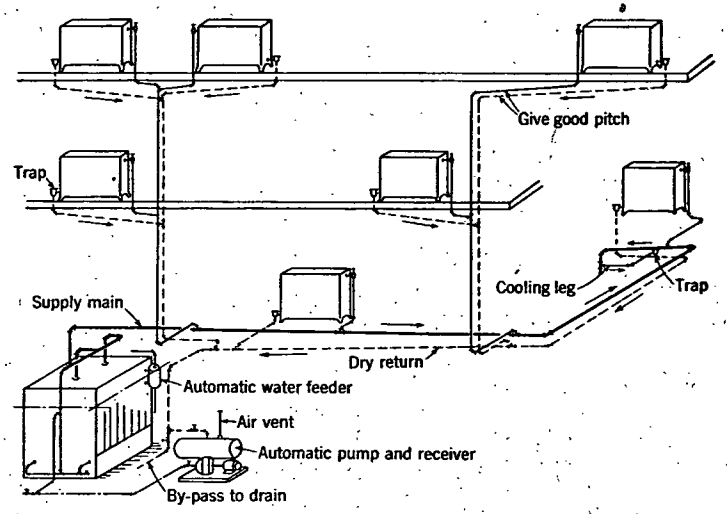


FIG. 17. TYPICAL INSTALLATION USING CONDENSATION PUMP

VACUUM HEATING PUMPS

On vacuum systems, where the returns are under a vacuum, and sub-atmospheric 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 non-condensable gases to atmosphere and to dispose of the condensation. 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[3] 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.

Manufacturers of vacuum pumps specify that the standard capacity of pumps shall be 0.3 to 0.5 cfm of air removal and 0.5 gpm of water per

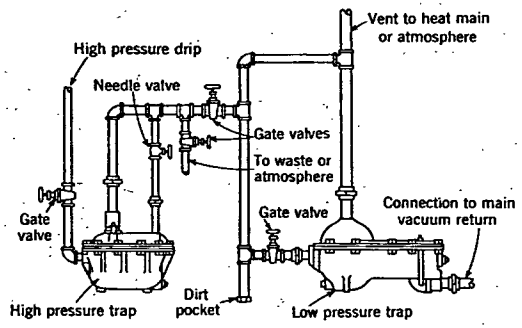


FIG. 18. METHOD OF DISCHARGING HIGH-PRESSURE APPARATUS INTO LOW-PRESSURE HEATING MAINS AND VACUUM RETURN MAINS THROUGH A LOW PRESSURE TRAP

1000 *EDR* served. This capacity is at 5½ in. of vacuum and with condensate at 160 F. The larger air capacity is for smaller systems and the smaller capacity for the larger systems.

Some manufacturers, however, specify more air capacity than standard where higher vacuums are desired and where air leakage is suspected.

The vacuum that can be maintained on a system depends upon the relationship of the air leakage rate into the system to the operating air capacity of the hydraulic evacuator when operating at any given return line temperature. The hotter the returns, the lower will be the possible vacuum for a given air leakage rate into the system. It is particularly essential 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 see that relatively higher temperature steam is prevented from entering the vacuum return lines through leaky traps, high pressure drips, etc. It is for this reason that the condensate from equipment using steam at high pressures should not be connected directly to a vacuum return line, but should drain to a receiver 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 in Fig. 18.

Vacuum Pump Controls

In the ordinary vacuum system, the vacuum pump is controlled 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 automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum on the system. A selector switch is usually provided to allow operation at night as a condensation 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 sub-atmospheric 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 sub-atmospheric 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 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 condensation is used for average vacuums and systems.

TRAPS

Traps are generally classified as to function as (*a*) separating traps, (*b*) return, lifting or vacuum traps, and (*c*) air traps. Separating traps may be either float operated, thermostatically operated, or float and thermostatically operated. Return traps for low pressure service are referred to later as *alternating receivers* in this chapter. Return traps may also operate to receive condensate under a vacuum and return it to atmosphere or a higher pressure. Air traps are generally float operated.

Separating traps are used to release water of condensation but to retain steam. The thermostatic, and float and thermostatic types release both condensate and air but retain steam. Separating traps are used for draining condensate from radiators, indirect air heaters, steam piping systems, kitchen equipment, laundry equipment, hospital equipment,

drying equipment and many other kinds of apparatus. Air traps release air but retain water. Devices known as air vents are, in principle, traps which allow the passage of air but prevent the passage of either water or steam.

Return traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap.

Traps may also be classified according to the principle of operating device which supplies the power to cause them to function as (1) float, (2) bucket, (3) thermostatic, (4) float and thermostatic, (5) impulse, or (6) tilting traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass may be used to indicate the height of water in the 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. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the *upright bucket* trap, the water of condensation enters the trap 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 water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a 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.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the valve. Water entering the trap fills the bucket, which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used, but they are unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators are called *blast* traps.

Float and thermostatic traps have both a thermostatic element to release air and a float element to release the water.

Impulse traps operate with a moving valve actuated by a control cylinder. When the trap is handling condensate, the pressure required to lift the valve is greater than the reduced pressure in the control cylinder and consequently the valve opens allowing a free discharge of condensate. As the remaining condensate approaches steam temperature, flashing results, flow through the valve orifice is choked and the pressure builds up in the control chamber closing the valve.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct-return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct-return trap assures safety for such systems, and guarantees the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counter-balanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight overbalances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened, thus admitting live steam pressure on the surface of the water, and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying. The air relief of tilting traps is taken care of by the valves of the trap.

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. Any steam main in any heating system can be elevated if dripped. Fig. 19 shows a connection where the steam main is raised and the drain is to a wet return. If the elevation of the low point is above a dry return, it may be drained through a trap to the dry return in two-pipe vapor, vacuum and sub-atmospheric 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 condensation (Fig. 20). Horizontal return pipes may be carried past doorways and other ob-

structions by using the scheme illustrated in Fig. 21. It will be noted that the large pipe, in this case, runs below the obstruction and the smaller one over it.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 22, but where radiator condensation 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

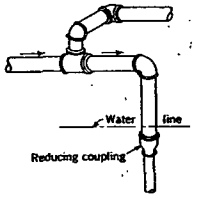


FIG. 19. DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

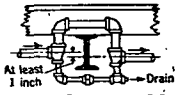


FIG. 20. LOOPING MAIN AROUND BEAM

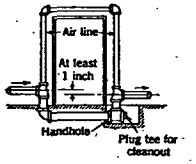


FIG. 21. LOOPING DRY RETURN MAIN AROUND OPENING

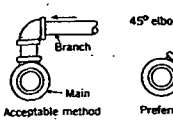
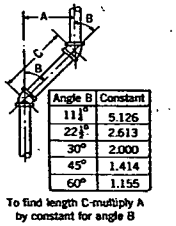


FIG. 22. METHODS OF TAKING BRANCH FROM MAIN



Angle B	Constant
11¼°	5.126
22½°	2.613
30°	2.000
45°	1.414
60°	1.155

FIG. 23. CONSTANTS FOR DETERMINING LENGTH OFFSET PIPE

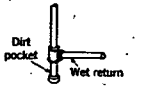


FIG. 24. DIRT POCKET CONNECTION

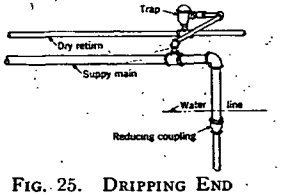


FIG. 25. DRIPPING END OF MAIN INTO WET RETURN

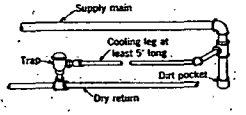


FIG. 26. DRIPPING END OF MAIN INTO DRY RETURN

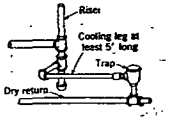


FIG. 27. DRIPPING HEEL OF RISER INTO DRY RETURN

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 from the steam main are usually made about 5 ft long to provide flexibility for movement in the main.

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. 23.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck 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. They are constructed as shown in Fig. 24.

On vapor systems where the end of the steam main is dripped down 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. 25. On vacuum systems the ends of the steam mains are dripped and vented into the return through drip traps opening into the return line. The same method may be used in atmospheric systems.

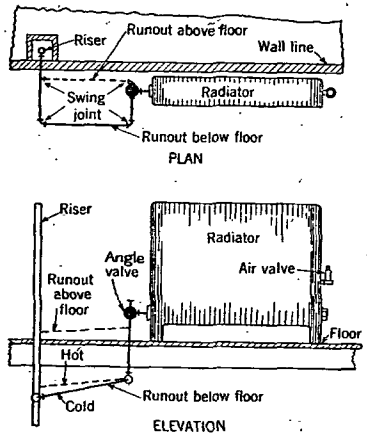


FIG. 28. ONE-PIPE RADIATOR CONNECTIONS

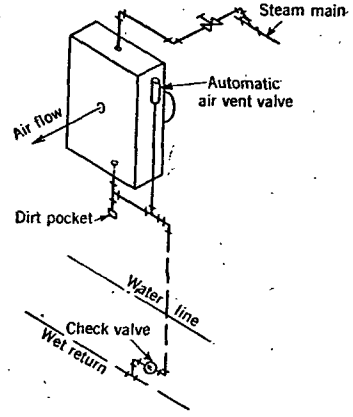


FIG. 29. UNIT HEATER CONNECTED TO ONE-PIPE AIR-VENT SYSTEM

A float type trap is preferable to a thermostatic trap for dripping steam mains and large risers. If thermostatic traps are used, a cooling leg (Fig. 26) should always be provided. The cooling leg is for cooling the condensation sufficiently before it reaches the trap, so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottoms of the steam risers are dripped in the manner shown in Fig. 27. On large systems it is desirable to install a gate valve in the cooling leg ahead of the trap.

CONNECTIONS TO HEATING UNITS

Riser, radiator and convector connections must not only be properly pitched at the time they are installed but must 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 shown in Fig. 28. Where the vertical distance is limited and the runouts must run above the floor,

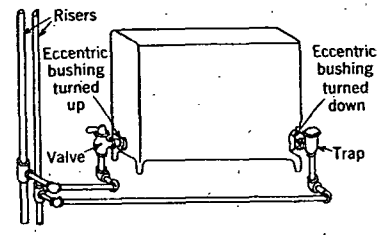


FIG. 30. TYPICAL CONNECTIONS FOR TWO-PIPE SYSTEM

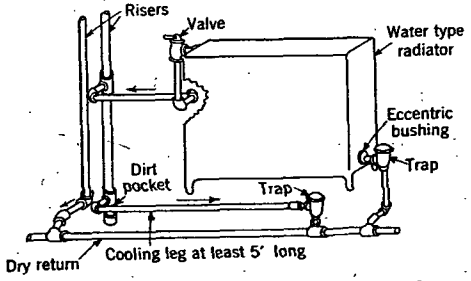


FIG. 31. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS

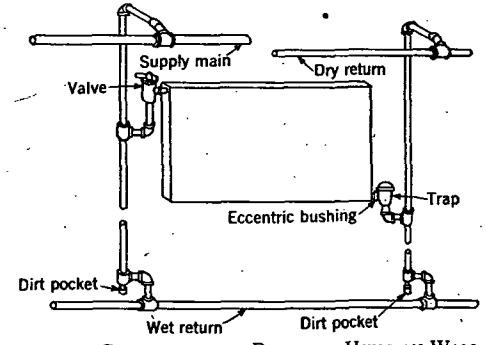


FIG. 32. CONNECTIONS TO RADIATOR HUNG ON WALL

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. 29.

Typical two-pipe radiator or convector connections are shown in Figs. 30, 31 and 32. While the top is the preferred location for the control

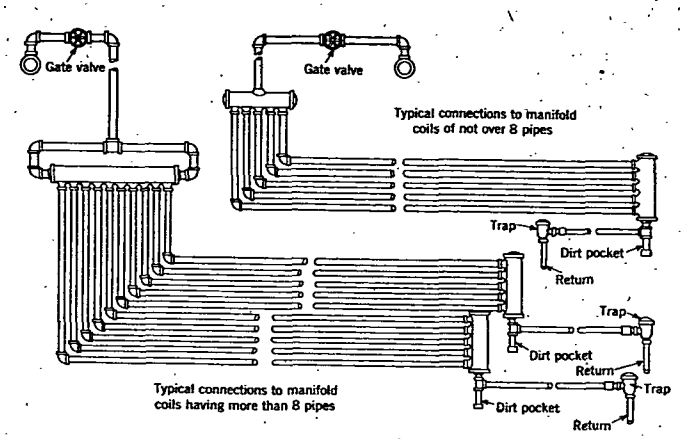


FIG. 33. TYPICAL PIPE COIL CONNECTIONS

valve, it may be located at the bottom. Short radiators may be top supply and bottom return on same end. With convectors the control valve is sometimes omitted and a damper in outlet grille used forheat control. The typical method of connecting pipe coils is shown in Fig. 33 and is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

Typical pipe connections for indirect radiators and tempering or heating stacks are shown in Figs. 34, 35, 36, 37 and 38.

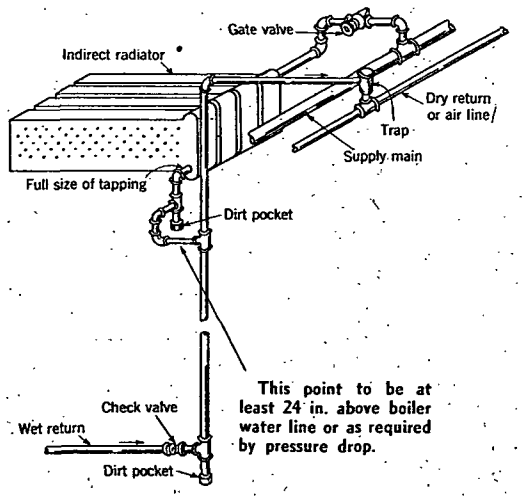


FIG. 34. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

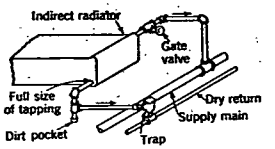


FIG. 35. PIPING CONNECTIONS TO INDIRECT RADIATORS WITH DRY RETURNS

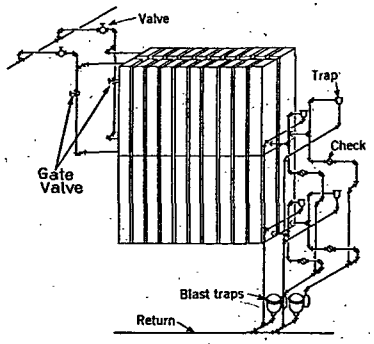


FIG. 36. SUPPLY AND RETURN CONNECTIONS FOR HEATING UNITS OF CENTRAL FAN SYSTEMS

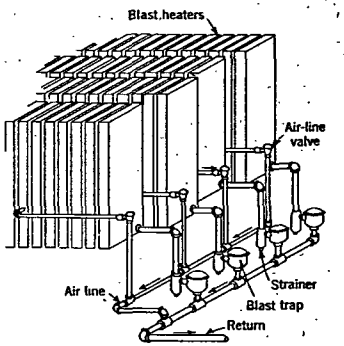


FIG. 37. TYPICAL CONNECTIONS TO CENTRAL FAN SYSTEM HEATING UNITS EXCEEDING 12 SECTIONS

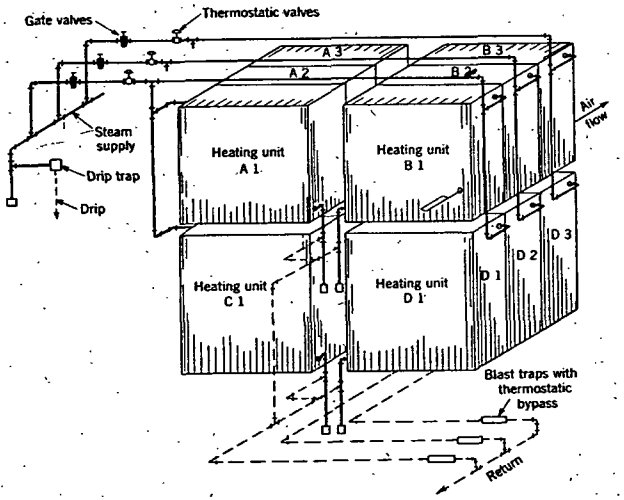


FIG. 38. TYPICAL PIPING FOR ATMOSPHERIC AND VACUUM SYSTEMS WITH THERMOSTATIC CONTROL (CENTRAL FAN SYSTEM)

Where a building is served by a vacuum system or a sub-atmospheric system the stacks should be piped in the usual manner and traps of large capacity, preferably of the combination float and thermostatic type, should be used. In the orifice and *closed* two-pipe systems, traps should be used on the returns so that a pressure above that of the atmosphere may be secured on the heaters.

Each stack should have a separate steam and return connection and trap. Wide stacks are more evenly heated if divided and supplied with two steam connections, one at each end, and a return connection for each steam connection. For stacks of large capacity it is sometimes desirable to run a separate steam main direct from the boiler to the stacks.

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, as done on by-passes around pressure reducing valves or on by-passes around traps.

REFERENCES

1—A.S.H.V.E. RESEARCH REPORT No. 954—Condensate and Air Return in Steam Heating Systems, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 199).

2—Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained. Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

3—A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 33).

CHAPTER 24

Hot Water Heating Systems and Piping

Systems of Piping, Forced Circulation Pipe Sizes, Gravity Circulation Pipe Sizes, Expansion Tanks, Installation Details, Examples of Piping Design

THERE are two general systems of piping used for either gravity or forced hot water systems: (a) Two-pipe system, (b) One-pipe system.

With either of these piping systems the distributing mains may be located in the basement with up-feed to the radiators and risers, or the supply main may be located in the attic with the return main located in the basement. For radiators located on the basement floor the mains may be run at the ceiling, as one of the advantages of a forced hot water heating system is that the returns need not be below the radiators as required with a steam system. In some one-pipe systems there is one supply main in the basement with separate flow and return riser connections to the radiators.

In the two-pipe system there are separate supply and return pipes throughout so that the radiators are connected in parallel, resulting in the same water temperature in all radiators. With the one-pipe system part of the water flows through more than one radiator, so that the water temperature toward the end of the main is not as high as near the boiler. However, with the one-pipe system, by maintaining a rapid circulation and small difference in temperature between the water leaving and returning to the boiler or other heat generator, the variation in the radiator water temperature is reduced.

The two-pipe system for larger buildings should, if possible, be arranged for reversed return. The direct and reversed return systems are shown in Figs. 1 and 2. With the reversed return system, the length of the water circuit for any one radiator is the same as for any other radiator and, therefore, the friction and temperature losses to all radiators should be nearly the same.

In some cases the reversed return system involves no more piping than the direct return system. In the case of large buildings, it is often advisable to zone the piping.

The quantity of water which must be circulated may be determined from the following equation:

$$W = \frac{H}{C\Delta t} \qquad (1)$$

where

W = weight of water circulated in pounds per hour.

H = heat required in Btu per hour.

C = specific heat (= 1 for water).

Δt = change in water temperature between supply and return line in Fahrenheit degrees.

Weight may be changed to gallons per minute by dividing W by 8 x 60 where 8 is average weight of a gallon of water at temperatures existing in heating systems.

Mechanical Circulators

Circulating pumps are usually of the centrifugal type. The required capacity of the pump is calculated as shown by Equation 1. For example, for 100 Mbh and 20 deg drop a pump having a capacity of 5000 lb water per hour or 10 gpm should be used. The resistance head is based on the system as designed. In large systems the economical size of pump may be determined by comparing the cost of power for operation with the annual charges on the capital cost of the piping system, as larger pipe sizes mean less pump power. Velocities through piping in excess of 4 fps are likely to cause disturbing noises in buildings other than factories. In large systems the pumps are run continuously while in small ones they are run either continuously or intermittently depending on the type of automatic temperature control selected. Small circulating pumps are usually driven by direct-connected electric motors. Under certain conditions a valved by-pass should be provided and the piping so designed that in case of breakdown of the pump or failure of electric current there will be sufficient gravity circulation to keep the building reasonably warm. In large buildings or groups of buildings, it is often advisable to have two pumps, each of about 70 per cent of the total capacity, to take care

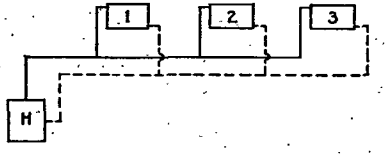


FIG. 1. A DIRECT RETURN SYSTEM

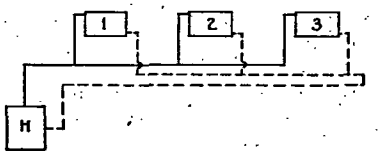


FIG. 2. A REVERSED RETURN SYSTEM

of breakdown service. During mild weather, variations in water temperature may be utilized to balance the required heat loss. In the larger systems steam turbines are sometimes used to drive the pumps, the exhaust steam being used for heating the water, and in such buildings as hospitals this may be the most economical method.

As the average pump used for water circulation is not over 60 per cent efficient, the cost of power on a large installation should be computed and comparisons made between the savings in capital cost of piping and the annual cost of power.

FORCED CIRCULATION PIPE SIZES

The heads available to cause circulation are much greater in forced circulation systems than in gravity circulation systems. Consequently, higher velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipe sizes of a heating system are reduced, the necessary increase in the velocity of the water increases the friction losses and thus the cost of operation and the initial cost of the circulating equipment. The increased velocity of a forced circulation system offers a number of advantages, such as a much shorter heating-up period and a more flexible control of hot water circulation. This improved performance merits the small increase in operating cost necessary to circulate the water mechanically. The velocities required should be determined by calculation for the particular system under consideration. As forced circulation velocities

are higher than those in gravity systems, and as the friction in a heating system varies almost as the square of the velocity, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in a gravity circulation system, and, consequently, it is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

In forced circulation systems, it is common practice to use a temperature drop of 20 to 30 deg across the boiler or heater. Next, the allowable friction in the water circuit must be decided. The allowable friction is

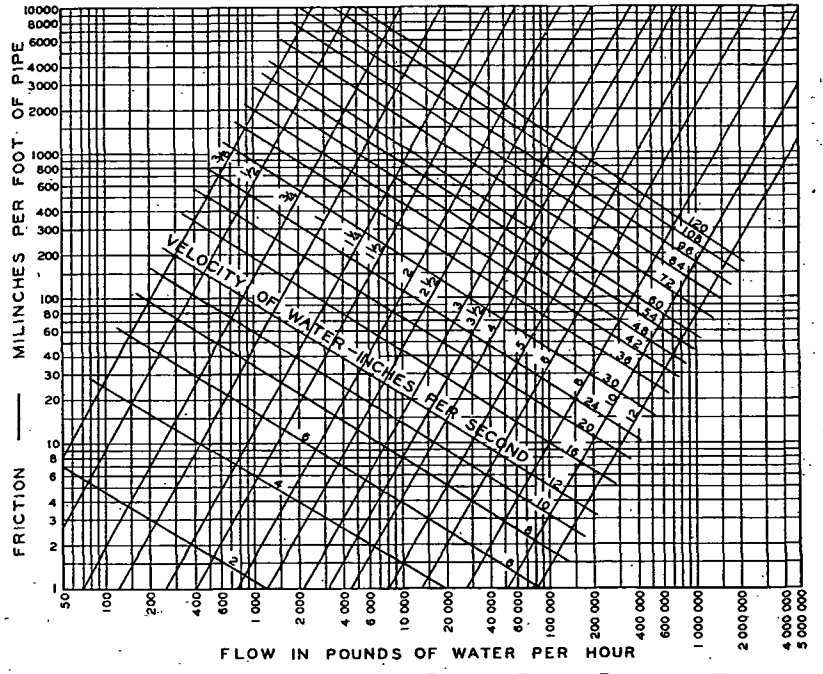


FIG. 3. FRICTION IN BLACK IRON PIPES BASED ON FLOW IN POUNDS PER HOUR

controlled partly by the characteristics of the pumps available for this service. The heads developed by these pumps range from 2 to 5 ft for residential systems up to 100 ft for institutional heating plants.

The problem in sizing of pipe is one of selecting sizes which will cause a circuit resistance approximately equal to the head available to produce circulation when the desired quantity of water is flowing.

The friction loss in black iron pipes for a determined weight of water may be obtained from Fig. 3. For the frequently used temperature drop of 20 deg the friction loss may be determined directly from the heat requirement by means of Table 1 or Fig. 4 for black iron pipes, or by means of Table 2 for copper tubing.

The resistance of various types of fittings expressed in equivalent elbow resistance is shown in Table 3.

Table 1. Heat-carrying Capacity of Standard Black Pipes with Temperature Drop of 20 Deg[a]

Nominal Pipe Sizes ⅜ in. to 12 in., and Friction 4 to 800 milinches per foot (A = Capacity, Mbh. B = Velocity, inches per second) (One milinch equals 0.001 in.)

Milinch Friction Loss per Foot of Pipe		Nominal Pipe Size, Inches															
		⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12
4	A	0.75	1.35	2.85	5.4	11.3	17.0	33.0	53.1	95	141	197	363	596	1250	2320	3730
	B	1.5	1.7	2.1	2.4	2.9	3.2	3.8	4.3	5.0	5.5	6.0	7.0	7.9	9.6	11	12
6	A	0.9	1.7	3.6	6.75	14.0	21.2	41.3	66.4	119	176	248	456	748	1570	2920	4690
	B	1.8	2.1	2.6	3.0	3.6	4.0	4.7	5.3	6.2	6.9	7.5	8.8	10	12	14	16
8	A	1.05	2.0	4.2	7.9	16.4	24.8	48.4	77.9	140	207	291	535	879	1850	3440	5520
	B	2.1	2.5	3.0	3.5	4.2	4.7	5.6	6.3	7.3	8.0	8.8	10	12	14	17	19
10	A	1.2	2.2	4.7	8.9	18.6	28.0	54.7	88.1	158	234	329	605	997	2100	3910	6270
	B	2.4	2.8	3.4	4.0	4.8	5.3	6.3	7.1	8.2	9.1	9.9	12	13	16	19	27
12	A	1.35	2.45	5.2	9.8	20.5	31.0	60.4	97.4	175	259	364	671	1100	2320	4330	6950
	B	2.7	3.1	3.7	4.4	5.3	5.9	6.9	7.8	9.1	10	11	13	15	18	21	24
14	A	1.45	2.65	5.65	10.7	22.3	33.7	65.8	106	190	282	397	731	1200	2530	4730	7590
	B	2.9	3.4	4.1	4.8	5.7	6.4	7.6	8.5	9.9	11	12	14	16	20	23	26
16	A	1.55	2.85	6.05	11.5	24.0	36.3	70.8	114	205	303	428	787	1300	2730	5100	8190
	B	3.1	3.6	4.4	5.1	6.2	6.9	8.1	9.7	11	12	13	15	17	21	25	28
20	A	1.75	3.25	6.85	13.0	27.1	41.0	80.0	129	232	344	484	892	1470	3100	5790	9300
	B	3.5	4.1	4.9	5.8	7.0	7.7	9.2	10	12	13	15	17	20	24	28	32
25	A	2.0	3.65	7.75	14.7	30.6	46.3	90.5	146	263	389	548	1010	1670	3510	6570	10560
	B	4.0	4.6	5.6	6.5	7.9	8.8	10	12	14	15	17	19	22	27	32	36
30	A	2.2	4.0	8.55	16.2	33.8	51.2	100	162	290	430	607	1120	1850	3900	7280	11710
	B	4.4	5.1	6.1	7.2	8.7	9.7	11	13	15	17	18	22	25	30	35	40
35	A	2.35	4.4	9.3	17.6	36.8	55.7	109	176	316	469	661	1220	2010	4250	7940	12780
	B	4.7	5.5	6.7	7.9	9.5	11	13	14	16	18	20	23	27	33	39	44
40	A	2.55	4.7	10.0	18.9	39.6	59.9	117	189	341	505	712	1320	2170	4580	8570	13780
	B	5.1	5.9	7.2	8.4	10	11	13	15	18	20	22	25	29	35	42	47
50	A	2.85	5.3	11.3	21.4	44.7	67.7	133	214	386	572	807	1490	2460	5190	9720	15650
	B	5.7	6.7	8.1	9.5	12	13	15	17	20	22	24	29	33	40	47	54
60	A	3.15	5.85	12.4	23.6	49.4	74.9	147	238	427	633	893	1650	2730	5760	10780	17360
	B	6.3	7.4	8.9	11	13	14	17	19	22	25	27	32	36	44	52	60
70	A	3.45	6.35	13.5	25.7	53.8	81.4	160	258	465	690	973	1800	2970	6280	11760	18950
	B	6.9	8.0	9.7	11	14	15	18	21	24	27	29	35	40	48	57	65
80	A	3.7	6.8	14.5	27.6	57.9	87.6	172	278	500	743	1050	1940	3200	6770	12690	20440
	B	7.4	8.6	10	12	15	17	20	22	26	29	32	37	43	52	62	70
100	A	4.15	7.7	16.4	31.1	65.4	99.0	194	314	566	840	1190	2200	3630	7680	14400	23200
	B	8.3	9.7	12	14	17	19	22	25	30	33	36	42	48	59	70	80
150	A	5.2	9.6	20.4	38.8	81.6	124	243	393	709	1050	1490	2760	4560	9650	18120	29220
	B	10	12	15	17	21	23	28	32	37	41	45	53	61	74	88	101
200	A	6.05	11.2	23.9	45.4	95.5	145	285	461	832	1240	1750	3240	5360	11350	21320	34400
	B	12	14	17	20	25	27	33	37	43	48	53	62	71	87	104	118
300	A	7.5	13.9	29.7	56.6	119	181	356	577	1040	1550	2190	4060	6730	14270	26830	43300
	B	15	18	21	25	31	34	41	46	54	60	66	78	90	110	131	149
400	A	8.75	16.2	34.7	66.2	140	212	417	676	1220	1820	2570	4780	7910	16790	31580	51000
	B	18	21	26	30	36	40	48	54	64	71	78	92	105	129	154	175
500	A	9.85	18.3	39.2	74.8	158	239	471	765	1380	2060	2910	5410	8970	19040	35840	57880
	B	20	23	29	33	41	45	54	62	72	80	88	104	119	147	174	199
600	A	10.9	20.2	43.2	82.5	174	264	521	846	1530	2280	3220	5990	9930	21100	39740	64210
	B	22	26	32	37	45	50	60	68	80	89	97	115	132	162	193	221
800	A	12.7	23.6	50.5	96.5	204	310	610	992	1790	2670	3780	7030	11670	24820	46780	75620
	B	25	30	37	43	52	59	70	80	94	104	114	135	155	191	228	260

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with a temperature drop of 30 deg the capacities shown in this table are to be multiplied by 1.5.

Table 2. Heat-carrying Capacity of Type L Copper Tubing with Temperature Drop of 20 Deg[a]

Nominal Tube Sizes ⅜ in. to 4 in., and Friction 60 to 720 milinches per foot. (A = Capacity, Mbh. B = Velocity, inches per second) (One milinch equals 0.001 in.)

Nominal Tube Size, In.		Milinch Friction Loss per Foot of Tube											
		720	600	480	360	300	240	180	150	120	90	75	60
⅜	A	10	9	8	6.8	6.2	5.4	4.6	4	3.6	3	2.8	2.4
	B	27	24	21	18	16.5	14	13	11	10	8.5	8	7
½	A	20	18	16	13.5	12	10.8	9	8	7	6	5.4	4.7
	B	35	30	25	21	19	17	15	13	12	10	9	8
⅝	A	36	30	26	22.1	20	17.8	15	13.1	11.8	9.9	9	7.9
	B	37	34	30	24	21	19	17	15	13	11	10	9
¾	A	51	46	40	34	31	28	23.2	20.5	18.1	15.3	13.9	12.1
	B	42	38	33	27	24	21	19	17	14	12	11.5	10
1	A	104	94	82	70	63	56	47	42	37	32	28	25
	B	48	45	39	34	30	25	22	19	17	14.5	13	12
1¼	A	185	169	149	125	112	100	84	75	66	56	50	44
	B	55	51	45	39	35	30	25	22	19	17	15	13
1½	A	300	270	235	200	180	160	134	120	105	90	81	71
	B	62	57	51	43	39	35	30	25	22	19	17	15
2	A	625	560	495	420	375	335	280	250	200	188	170	150
	B	76	68	59	51	47	42	36	32	27	22	20	18
2½	A	1130	1010	890	750	680	600	500	450	395	335	305	270
	B	90	80	69	58	49	47	42	37	33	26	23	21
3	A	1840	1650	1450	1210	1100	980	820	740	650	550	490	420
	B	98	90	80	66	59	52	47	42	36	30	27	23
3½	A	2750	2480	2170	1840	1650	1450	1210	1100	980	820	740	650
	B	110	100	89	75	66	57	51	45	40	35	30	26
4	A	3900	3505	3100	2600	2350	2090	1760	1580	1390	1180	1080	950
	B	120	108	96	83	73	63	55	49	44	37	34	29

[a]For other temperature drops the pipe capacities may be changed correspondingly. For example, with temperature drop of 30 deg, the capacities shown in this table are to be multiplied by 1.5.

Table 3. Iron and Copper Elbow Equivalents[a]

Fitting	Iron Pipe	Copper Tubing	Fitting	Iron Pipe	Copper Tubing
Elbow, 90-deg	1.0	1.0	Angle radiator valve	2.0	3.0
Elbow, 45-deg	0.7	0.7	Radiator or convector	3.0	4.0
Elbow, 90-deg long turn	0.5	0.5	Boiler or heater	3.0	4.0
Elbow, welded, 90-deg	0.5	0.5			
Reduced coupling	0.4	0.4	Tee, per cent flowing		
Open return band	1.0	1.0	through branch:		
Open gate valve	0.5	0.7	100	1.8	1.2
Open globe valve	12.0	17.0	50	4.0	4.0
			25	16.0	20.0

[a]The friction in one 90 deg standard elbow is approximately equal to the friction of a length of straight pipe of the same nominal size and 25 diam long. Hence one elbow equivalent equals 25 *D* divided by 12 ft of straight pipe or tubing.

When a reversed return system is used, it is usually found that but little adjustment is required to attain uniform distribution to all radiators. In larger systems, orifices may become desirable to assist in balancing flow. The resistance introduced by various sizes of orifices is listed in Table 4.

As systems increase in size, it becomes desirable to provide two valves

for each radiator or heating unit, to permit its removal without requiring the draining of the entire system. One valve may be of the lock shield type so that it may be used for balancing flow.

All large systems should be provided with extra stop and drain valves, suitably located so that parts of the system may be isolated for repairs without making it necessary to drain the water from the entire system.

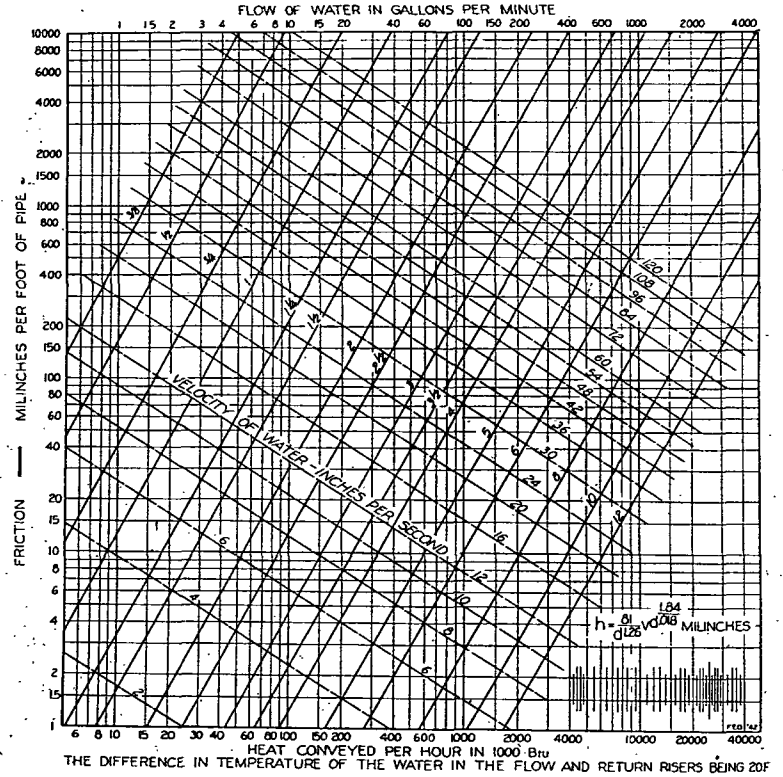


FIG. 4. FRICTION IN BLACK IRON PIPES

To find friction when temperature drop is other than 20 deg, multiply the actual heat conveyed by $\left(\frac{20}{\text{actual temp drop}}\right)$ and read the corresponding friction.

GRAVITY CIRCULATION PIPE SIZES

In gravity hot water heating systems the difference in density between the flow and return water produces the required natural circulation of the water. The design temperature difference is usually assumed from between 20 to 35 deg. After having determined the temperature difference and the temperature of the flow water, data given in Fig. 5 can be used to obtain the head available for circulation. With this information determined, pipe sizes which will have a resistance equal to the

TABLE 4. FRICTION (IN MILINCHES) OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN UNIONS

(One milinch equals 0.001 in.)

DIAMETER OF ORIFICES (INCHES)	VELOCITY OF WATER IN PIPE IN INCHES PER SECOND									
	2	3	4	6	8	10	12	18	24	36
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1½-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1¼-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois, (*Bulletin* 109, Table 6, p. 38, Davis and Jordan).

head available for circulation may be selected from the charts or tables in the same manner as for forced circulation systems.

Radiator heat emission rates from 150 to 200 Btu per square foot are commonly used so that flow temperatures generally range from 180 to 200 F or higher. Assuming a flow temperature of 200 F and a 35 deg drop, and with the mains located 4 ft above the top of the boiler, a head of 600 milinches results. This is obtained by following the 200 F floor riser line in Fig. 5 to where it intersects the 165 F return riser line and

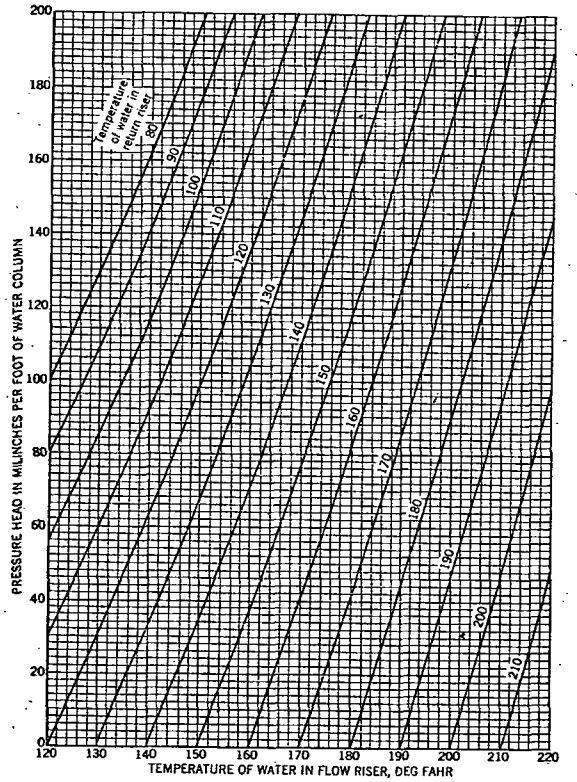


FIG. 5. [HEADS RESULTING FROM TEMPERATURE DIFFERENCE (GRAVITY SYSTEMS)

reading horizontally a head of 150 milinches per foot or 600 milinches for 4 ft. Assuming first floor radiators are located 3 ft above the mains and second floor radiators 12 ft above the mains, third floor 21 ft, and fourth floor 30 ft, the heads are 450, 1800, 3150 and 4500 milinches respectively.

EXPANSION TANKS

Water heated from 40 F to 200 F expands about 0.04 of the original volume. The expansion tank permits the change in volume of the water in the heating system to take place without producing undesirable stresses due to pressure in any part of the system. Expansion tanks may be open,

as illustrated in Fig. 6, or closed as shown in Fig. 7. An open expansion tank has free vent to the atmosphere and consequently the pressure on the surface of the water is always that of one atmosphere. The minimum contents of an open tank should be 0.06 of the volume of the water in the system including that in the boiler, heat transmitters, pipes, etc. This capacity is 50 per cent in excess of the actual increase in volume of water due to increase in temperature from 40 F to 200 F. The tank should be located at least 3 ft above the highest radiator. Provision must be made to prevent freezing of the water in the tank as well as in the pipe leading to the tank.

In a gravity circulation system, the pipe to the open expansion tank should be connected to the supply riser from the boiler, so that the air liberated from the water in the boiler will enter the expansion tank.

In a forced circulation system, the pipe to the open expansion tank

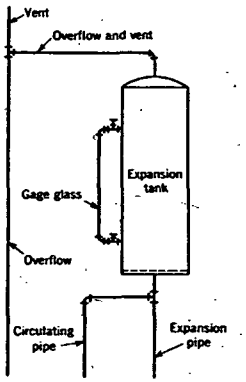


FIG. 6. AN OPEN EXPANSION TANK

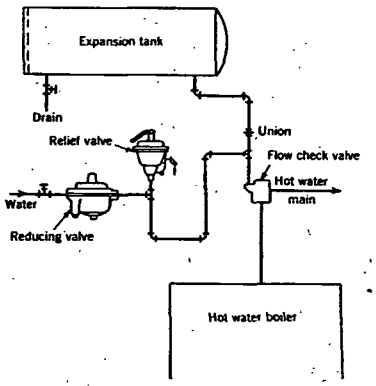


FIG. 7. A CLOSED EXPANSION TANK

should be connected on the suction side of the circulating pump, so that the head on the suction side of the pump will remain practically constant.

A closed expansion tank is sealed against free venting to the atmosphere. The tank may be above the highest radiator or heat transmitter, or may be below the lowest one. The minimum contents of a closed expansion tank must be such that the expansion of the water due to increase in temperature will be cushioned against a reservoir of compressed air above the water level in the expansion tank. The tank must provide space not only for the change in water volume, but also for variations in air volume within the tank due to changes in air pressure. If the closed expansion tank is below the heat transmitters, the tank should be larger than if it is above them, and the higher the building, under such circumstances, the larger should be the air capacity in excess of that required for increase in water volume due to temperature rise.

The size of a basement-located closed expansion tank should be at least equal to the following:

One story buildings: $x = 0.10\ V$ Three story buildings: $x = 0.17\ V$
Two story buildings: $x = 0.13\ V$ Six story buildings: $x = 0.28\ V$

where x = expansion tank size in gallons.
V = water volume in heating system in gallons.

This condition favors, especially in tall buildings, the placing of the closed expansion tank above the highest heat transmitter.

Any closed expansion tank located above the heat transmitters of a hot water heating system should be connected by a direct pipe with the flow main leaving the boiler, in order to enable the air to pass easily to the expansion tank. In a closed hot water heating system the water under pressure tends to absorb air at a rate increasing with pressure increase and decreasing with temperature increase.

Means must be provided to adjust and to observe the proportion of air within any closed expansion tank. This involves the provision of an air inlet valve, a water gage and a relief valve. A source of supply of compressed air for renewing the air cushion is highly desirable, especially in large, high pressure hot water heating systems where it is inconvenient, if not impracticable, to drain down the water in the system so as to permit introduction of atmospheric pressure air.

For every hot water heating system the designer should calculate the volume of water contained in the radiators, piping system, boiler, etc., in order to select the proper size of expansion tank. The water content

TABLE 5. VOLUME OF WATER IN STANDARD PIPE

PIPE SIZE, IN.	LINEAL FT OF PIPE CONTAINING 1 GAL	PIPE SIZE, IN.	LINEAL FT OF PIPE CONTAINING 1 GAL
½	63.1	2	5.75
¾	36.1	2½	4.02
1	22.2	3	2.60
1¼	12.8	4	1.52
1½	9.47	5	0.96
		6	0.67

of the piping can be obtained from Table 5. For a rough selection of size, however, it is sometimes assumed that 50 per cent of the volume of water is contained in the radiators, and that the water content per square foot of radiator heating surface is 0.2 gal for column radiators and 0.13 gal for tube type radiators.

Another rough method for determining the size of an expansion tank to be located above the highest radiator is to divide the square feet of radiation by the factor 40 to obtain the required capacity in gallons of the expansion tank.

INSTALLATION DETAILS

Items that should be considered in the design of this type of system are:

All piping must be so pitched that all air in the system can be vented either through an open expansion tank, radiators or automatic relief valves.

All piping must be arranged so that the entire system can be drained. Sections of piping individually valved shall have corresponding drain valves.

In large buildings, the piping may be zoned according to exposure of building, usage of building, or method of control.

All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

The pipe system should be designed so that each circuit has its correct friction for balanced water distribution. This may be done by change of pipe size or change in piping detail.

The connections from the boiler to the mains should be short and direct, to reduce the

friction and allow for expansion. It is frequently possible to avoid an elbow and to reduce the length of the pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

Generally, connections to risers or radiators are taken out of the top of mains either 45 or 90 deg. In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side.

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end. With gravity circulation

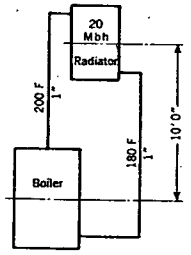


FIG. 8. GRAVITY SYSTEM

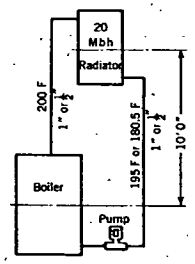


FIG. 9. FORCED CIRCULATION SYSTEM

the flow connection may be either at the top or at the bottom of the radiator. With short radiators both flow and return may be at the same end, but at top and bottom.

Unless used as heating surface, all piping, both flow and return, should be insulated.

EXAMPLES OF PIPING DESIGN

The following graded series of examples of the design of hot water piping systems will illustrate the fundamental principles and methods. The differences between reversed return and direct return systems are shown, and the methods of balancing the several radiators or circuits are illustrated. A simple gravity system is shown in Fig. 8 and an elementary forced circulation system is diagrammed in Fig. 9.

Elementary Gravity System

Example 1: A simple gravity-circulation system is illustrated in Fig. 8 with one radiator that is giving off heat at the rate of 20,000 Btu per hour or 20 Mbh. The boiler imparts heat to the water at the same rate, and the water circulates at a uniform velocity. This uniform velocity is such that the friction of the circuit is equal to the head developed by the difference in density between the supply and return water and the height of the system. The circuit consists of 1 boiler, 1 radiator, 2 ells, 1 radiator valve and a total of 24 ft of pipe.

Solution. With the average water temperatures of 200 and 180 F in the supply and return risers, respectively, the head will be 90 milinches per foot of water column. This head may be found from Fig. 5. Since the center of the radiator is 10 ft above the center of the boiler, the total head of the circuit is 10 x 90, or 900 milinches, or 0.9 inches of 190 F water. The friction of the circuit must then also be 900 milinches. The friction

of 1 ft of 1 in. pipe is found from Fig. 4 to be about 46 milinches at 20 Mbh, and the corresponding velocity 9 in. per second. (Note that all values in Fig. 4 are based on a temperature difference of 20 deg).

Similarly, if a 1¼ in. pipe were to be used, the friction head would be about 12 milinches per foot and the corresponding velocity about 5 in. per second, from Fig. 4.

To find the friction in the elbows, boiler, radiator and valve, Table 3 is used, and the entire circuit is found to be equal to 10 elbow-equivalents plus 24 ft of pipe. Each elbow-equivalent is equal to a pipe length of 25 times the nominal diameter. Then the equivalent lengths of straight pipe are 45 ft of 1 in. pipe or 50 ft of 1¼ in. pipe.

Hence, if 1 in. pipe is used, the friction of the circuit will be 45 x 46, or 2070 milinches, and if 1¼ in. pipe is used, the friction will be 50 x 12, or 600 milinches. A 1 in. pipe would, therefore, be too small and a 1¼ in. pipe too large to permit the desired circulation with a flow-return temperature difference of 20 deg.

If the circuit is of 1 in. pipe, the circulation will take place with a temperature difference greater than 20 deg, and if the circuit is of 1¼ in. pipe, the circulation will take place with a temperature difference smaller than 20 deg. To find, for example, the temperature difference at which a circuit of 1 in. pipe would transmit the required 20 Mbh, assume the difference to be 40 deg.

From Fig. 5, the head available for producing circulation would be 175 milinches per foot or 1750 for the system for a temperature drop from 200 to 160 F. The friction of the

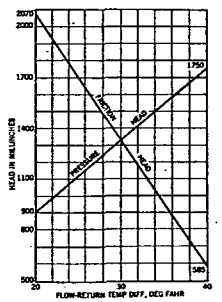


FIG. 10. DETERMINATION OF REQUIRED TEMPERATURE DIFFERENCE

system may be found from Fig. 4; the chart of this figure is based on a temperature difference of 20 deg; if the temperature difference were 40 deg, the heat conveyed would be twice that shown in the chart. Hence, find 10 Mbh on the lower scale, proceed vertically upward to the intersection with the 1 in. line, and from there to the left scale read 13 milinches per foot. Note that the velocity would then be only about 5 in. per second. The total friction would then be 45 x 13 or 585 milinches. Since the head would be 1750, circulation would take place with a temperature difference less than 40 deg. The required temperature difference may be determined by constructing the diagram of Fig. 10, from which it appears that the temperature difference with which the 1 in. pipe circuit would function is about 30 deg. Hence, if the flow riser temperature is 200, the return riser temperature will be 170, and the average water temperature in the radiator about 185 F.

Elementary Forced Circulation System

Example 2. Design a system for the piping arrangement shown in Fig. 9, according to one of the outlined procedures. The procedure may be as follows: Assume the head developed by the circulating pump and the pipe size and find the flow-return temperature difference; or, assume the head developed by the pump and the flow-return temperature difference and find the pipe size; or, assume the pipe size and the flow-return temperature difference and find the head which the circulating pump must develop.

Solution. Assume that the circulating pump will develop a head of 2 ft or 24,000 milinches and that a 1 in. pipe is to be used. The equivalent length of the circuit will then be 45 ft, as in Fig. 8, and the available head will be 24,000/45, or 533 milinches per foot. In Fig. 4, find 533 on the left scale, move horizontally to the intersection with

the 1 in. pipe line, and read about 77 Mbh delivered by the pipe (with a velocity of about 35 in. per second) for a temperature difference of 20 deg. Since the circuit is to deliver only 20 Mbh, the temperature difference will be 20 divided by 77 and multiplied by 20, or 5.2 deg. Hence, if the flow riser temperature is 200, the return riser temperature will be about 195, and the average water temperature in the radiator about 197.5 F.

If a ½ in. pipe were used instead of a 1 in., the equivalent length of circuit would be 35 ft instead of 45; the unit head, 686 milinches instead of 533; the velocity, 27 in. per second instead of 35; the temperature difference, 19.5 instead of 5.2; and the average water temperature in the radiator, about 190.5 instead of 197.5 F.

If the 1 in. pipe is used for the circuit, the gravity head will be 22 milinches per foot, or 220 for the circuit (Fig. 5, 200 to 195). Since this is only 1 per cent of the pump head (24,000 milinches), it may be neglected in the calculation, as was done previously. However, there are cases in which the gravity head is so large compared with the pump head, that it should be included in the calculation.

The methods just described for the design of the two elementary systems are fundamental and apply to the design of all hot water heating systems. In every system, however large and complicated, the pipe system must be such that the head forcing the water from the boiler to any one radiator is equal to the friction in that radiator's circuit when the

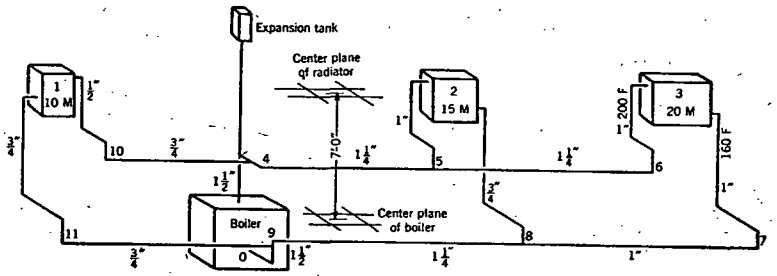


FIG. 11. TWO-PIPE GRAVITY CIRCULATION SYSTEM

radiator is receiving its proper quantity of hot water and the system is functioning at a steady rate.

Two-Pipe Gravity Circulation System

Example 3. In the system shown in Fig. 11, water leaving the boiler may flow to any one of the three radiators. If the system is designed correctly, each radiator will receive its proper share of the hot water. Since Radiator 3 has the largest load and is also farthest from the boiler, it is the least favorably located with reference to circulation, and its circuit should be designed first. If the pipes leading to it are large enough, it will be easy to secure sufficient circulation for the other two radiators.

The system is to function with a 40 deg flow-return temperature difference. The head for each radiator is 7 x 175 (Fig. 5), or 1225 milinches; the friction for each radiator circuit must, therefore, also be 1225 milinches.

Solution. In order to design a radiator circuit accurately and systematically, the circuit should be divided into sections. The division points of sections must be where the pipe sizes change or may change, and where the volume of water flowing in the pipe changes. The data relating to the several sections may be recorded as shown in Table 6.

Data recorded in Table 6 show that Circuit 3 consists of 70.4 ft of pipe and 21.5 elbow equivalents. Assuming that the average size of the pipe will be 1¼ in., the 21.5 elbow equivalents may be replaced by 21.5 x 2.6 (25/12 x 1.25 x 1.0), or 56 ft of pipe, which would make the total equivalent length of the circuit 70.4 plus 56 or 126.4 ft of pipe, and the average friction 1225/126.4, or about 10 milinches per foot. For this unit friction and a temperature difference of 40 deg (see Fig. 4), a 1 in. pipe will convey 18 Mbh, a 1¼ in. pipe, 37 Mbh, and a 1½ in. pipe, 56 Mbh. The pipe sizes for the several sections of Circuit 3 may be selected as indicated in Table 6. Having selected the pipe

sizes, the unit friction may be found from Fig. 4 and the total friction calculated and recorded as shown in Table 6. If the grand total, in the present case 1192 milinches, differs materially from the available head, 1225 milinches, one or more of the pipe sizes must be changed and the calculation repeated until the total friction is practically equal to the available head of 1225 milinches.

It is not necessary, in the design of hot water heating systems, to be extremely careful to have the friction exactly equal to the available head, because a hot water heating system has the ability to adjust itself to varying conditions of considerable magnitude. For example, in the present case, the calculated friction is 1192 milinches, or about 3 per cent less than the calculated available head; the water would, therefore, circulate a little faster than contemplated and the return temperature would be a little higher than 160 F.

TABLE 6. TABULATED DATA FOR EXAMPLE 3

Circuit No. 3. Available head 1225 milinches

SECTION	LOAD MBH	PIPE LENGTH FT	ELBOWS NO.	PIPE SIZE IN.	EQUIVALENT LENGTH, FT	UNIT FRICTION MILINCHES PER FT	TOTAL FRICTION, MILINCHES
0-4	45	3.5	2.5	1½	11.3	6.3	71
4-5	35	10.0	2.0	1¼	15.2	9.2	140
5-6	20	12.0	0.0	1¼	12.0	3.4	41
6-3	20	6.5	5.5	1	17.5	13	228
3-7	20	9.5	5.5	1	20.5	13	267
7-8	20	12.8	1.0	1	14.9	13	194
8-9	35	13.8	2.5	1¼	20.3	9.2	187
9-0	45	2.3	2.5	1½	10.1	6.3	64
Total		70.4	21.5		121.8		1192

Circuit No. 2. Available head 1225 milinches

0-4	45						71
4-5	35						140
5-2	15	6.2	6	1	18.7	7.5	140
2-8	15	9.5	7	¾	20.4	24.0	490
8-9	35						187
9-0	45						64
Total							1092

Circuit No. 1. Available head 1225 milinches

0-4	45						71
4-10	10	9.0	1.5	¾	11.3	11	124
10-1	10	5.0	5.5	½	10.7	46	434
1-11	10	9.5	5.5	¾	18.1	11	199
11-9	10	11.5	2.5	¾	15.4	11	169
9-0	45						64
Total							1067

This would immediately lower the head available for circulation and the balance with the friction would occur at some point between 1192 and 1225 milinches.

Since it is generally not necessary to make extremely refined calculations, Table 1 may often be used instead of the chart of Fig. 4 to determine pipe sizes. For example, in the present case, Table 1 shows that for a friction of 10 milinches a 1½ in. pipe would convey 28 Mbh with a temperature difference of 20 deg or 56 Mbh with a temperature difference of 40 deg. Since the 1½ in. pipe in Fig. 10 is to convey only 45 Mbh with a temperature difference of 40 deg, or only 22.5 with a temperature difference of 20 deg, it is evident from the table that the friction will be between 6 and 8 milinches. For 6 milinches the heat conveyed is 21.2 and for 8 milinches, it is 24.8; for 22.5 Mbh, the friction would be estimated to be about 6.5, which would be sufficiently accurate for the present calculation.

Having completed the design of Circuit 3, it is simple to design Circuit 2 because it has four sections in common with Circuit 3 and it is only necessary to design Sections 5-2 and 2-8, as shown in Table 6, so that the total friction of Circuit 2 will be practically equal to the total friction of Circuit 3.

Having completed the design of Circuit 2, it is necessary to design Circuit 1, as shown in Table 6, so that its total friction will be approximately equal to the total friction of the other two circuits since all three circuits have equal heads.

Two-Pipe Forced Circulation System

Example 4. In the design of a system for Fig. 12, as in the design of Example 2, there are three unknowns—head, pipe size, and flow-return temperature difference, any two of which may be assumed and the third found. In the design of a system for Fig. 9, the pressure head and the temperature difference were assumed and the pipe sizes found. In this case, the head is to be found.

Solution. In selecting the temperature difference and the pipe sizes, it should be borne in mind that the first cost of the radiation can be reduced by reducing the temperature difference and the first cost of the pipe can be reduced by increasing water velocity, but the friction is increased and, thus, the cost of pumping. The choice of temperature difference and pipe sizes which produce the greatest economy in first cost and in cost of operation can be determined after having made two or three trial designs. For the first design, 20 deg will be selected as the temperature difference and the pipe

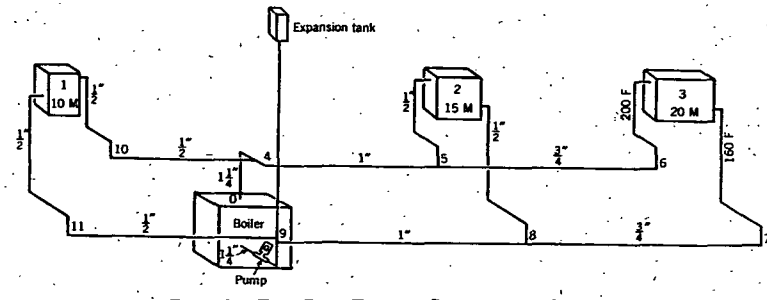


FIG. 12. TWO-PIPE FORCED CIRCULATION SYSTEM

sizes will be chosen as shown in Fig. 12. A calculation similar to that of Table 7 will show that the friction of Circuits 1, 2, and 3 will be, respectively, about 9000, 15,300, and 14,300 milinches. To increase the friction of Circuit 1 from 9000 to 15,000 milinches would require the insertion in the circuit of a section of ⅜ in. pipe, or an orifice resistor, or a regulating valve. However, this would cause unnecessary expense. The system will function well with the pipe system shown in Fig. 12. If the circulator maintains a head of 15,300 milinches, the velocity in Circuit 1 will increase until the friction of the circuit is also 15,300 milinches; *i.e.*, its friction will be increased from the calculated 9000 to the required 15,300, or 6300 milinches.

As calculated, the three radiators are to dissipate 10, 15, and 20 Mbh, respectively, with a temperature difference of 20 deg. Consequently, water must flow through these three radiators at the rates of 1, 1.5, and 2 gpm, respectively. When water flows through a ½ in. pipe at the rate of 1 gpm, the velocity in the pipe is (Fig. 4) about 13 in. per second and the unit friction is 165 milinches. The equivalent length of sections 4-10, 10-1, 1-11, and 11-9 of this circuit is 50 ft, and its total friction is 8250. In order that the friction may be increased 6300 milinches, the unit friction must be increased 126 milinches; consequently, the velocity in the ½ in. pipe (Fig. 4) must be increased from 13 to 16 in. per second. Hence, when the friction of Circuit 1 has been increased to 15,300, water will flow through Radiator 1 at the rate of 16 ÷ 13, or 1.23 gpm. This increase in volume of water will increase the load on the circulating pump in the proportion of 450 to 473 and will increase the heat dissipation of Radiator 1 slightly (about 3 per cent) but otherwise will not affect the operation of the system.

One-Pipe Gravity Circulation System

Example 5. A one-pipe system is one in which the cooled water from the heating units is returned to the supply main. A two-pipe system as shown in Figs. 11 and 12 is one in which the cooled water returns to the boiler through a separate return main.

Many large heating systems contain some one-pipe and some two-pipe sections. The piping system shown in Fig. 13 functions with a flow-return temperature difference of 40 deg.

Solution. Since the four radiators are each to deliver 15 Mbh, and since the water is to leave the boiler at 200 F and return at 160 F, Radiator 1 will receive 200 F water, Radiator 2, 190 F water, Radiator 3, 180 F water, and Radiator 4, 170 F water. Since the system is to supply 60 Mbh with a temperature difference of 40 deg and since 1 lb of water liberates 1 Btu when cooled 1 deg, it is necessary that water circulate through this system at the rate of 60,000 ÷ 40, or 1,500 lb per hour, or 25 lb per minute. Assuming one gallon of water to weigh 8.33 lb, water must circulate in the system at the rate of 3 gpm. If the temperature difference were 20 deg instead of 40, the circulation would be at the rate of 6 gpm. It is well to remember that, with a temperature difference of 20 deg, water circulating at the rate of 1 gpm will convey heat at the rate of 10 Mbh. The chart of Fig. 4 shows the rate of circulation in gpm on the upper scale and the corresponding heat conveyance on the lower scale.

The system may be divided into 5 separate systems. Each of the four radiators with its flow and return lines constitutes an elementary heating system (similar to Example 1), and the flow main with its two risers is also a complete elementary system.

TABLE 7. TABULATED DATA FOR EXAMPLE 7

CIRCUIT	LOAD MBH	PIPE LENGTH FT	ELBOWS NO.	EQUIVALENT LENGTH FT	SIZE IN.	UNIT FRICTION MILINCHES PER FT	TOTAL FRICTION, MILINCHES
0-1	3600	65	1.8	78	6	100	7,800
1-2	3150	130	0	130	5	180	23,900
2-3	2700	130	0	130	5	150	19,500
3-4	2250	130	0	130	5	100	13,000
4-5	1800	130	2	142	4	210	29,800
5-6	1350	130	0	130	4	125	16,200
6-7	900	130	0	130	3½	113	14,700
7-8	450	130	0	130	2½	190	24,700
8-16	450	Estimated Friction					20,000
16-17	3600	65	1.8	78	6	100	7,800
17-18	7200	130	0	130	8	90	11,670
						Total	188,570

If the center of the boiler is 4 ft below the center of the flow main, and if the flow riser contains 200 F water and the return riser 160 F water, the head for the main circuit is (Fig. 5) 4 x 175, or 700 milinches. The circuit consists of 110 ft of pipe and 10 elbow equivalents; its equivalent length is about 150 ft if a 2 in. pipe is used as the main. The average head will be 700 ÷ 150, or 4.7 milinches. According to Fig. 4 or Table 1 for a 4.7 milinch friction, a 2 in. pipe will convey about 70 Mbh. Since the pipe is to convey only 60 Mbh, it is slightly too large but should be used. The friction will, then, be 3.5 milinches instead of the permissible 4.7. The water will circulate with a temperature difference slightly less than 40 deg, and the three last radiators would receive water slightly warmer than indicated in Fig. 13.

As the water flows in the main and arrives at one of the four points marked *A*, the flow will be divided and a portion of the water will take the short path in the main to the point *B*, and the remainder will take the long path through the radiator to the point *B*. Since the two paths together offer less resistance to the flow than the one path alone, the unit friction will be less than 3.5 milinches between the points *A* and *B*. If the distance from *A* to *B* is 4 ft, and if, for example, the flow in the short path is 2 gpm, the head forcing the water along the two paths is 4 x 1.6, or about 6 milinches. Since the friction of the long path is much greater than the friction of the short path, only a comparatively small portion of the water would take the long path. However, the gravity head of the radiator supplies an additional head for the long path. If the center of the radiator is 4 ft above the main and if the radiator circuit is designed for a temperature difference of 30 deg, the radiator head will be about 4 x 120, or 480 milinches. In that case, the head is 6 milinches for the short path and 486 milinches for the long path. The radiator circuit consists of 11 ft of pipe and about 14 elbow equivalents. If the circuit is of 1 in. pipe, its equivalent length is about 40 ft and the unit friction should be 486 ÷ 40, or 12 milinches. For this friction and 30 deg temperature difference, a 1 in.

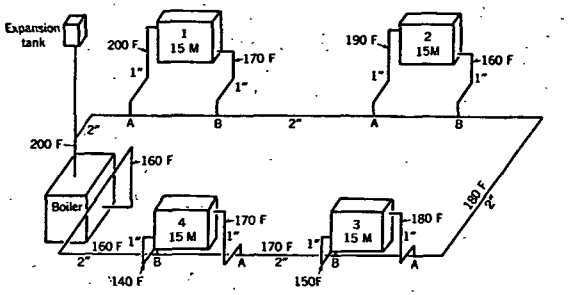


FIG. 13. ONE-PIPE GRAVITY CIRCULATION SYSTEM

pipe conveys about 15 Mbh (Fig. 4). A 1 in. pipe is, therefore, the correct size, and the water would flow through the radiator with a temperature difference of 30 deg.

Since each of the four radiators delivers one fourth of the total heat, and since the total temperature difference is to be 40 deg, the water would cool 10 deg in every radiator if all the water passed through every radiator. Since the water cools about 30 deg in flowing through the radiator only ⅓ of the water flows through the radiator and ⅔ flow through the main from point *A* to point *B*.

It follows from these calculations that, if the main is of 2 in. pipe and the radiator branches are of 1 in. pipe, water will circulate through the system at the rate of 3 gpm with a temperature difference somewhat less than 40 deg and that, at the radiator connections, water will circulate through the radiators at the rate of 1 gpm and that, consequently, between radiator branch connections the friction is less since only 2 gpm flow through the main between those points.

One-Pipe Forced Circulation System

Example 6. The system shown in Fig. 13 as a gravity circulation system may be changed to a forced circulation system by inserting a circulating pump as shown in Fig. 14. The location of the expansion tank should then be changed as indicated.

Solution. To design this system the pipe sizes and the temperature difference may be assumed; for example, 1¼ in. pipe may be selected for the main and ¾ in. pipe for the radiator branches and risers, and 20 deg as the flow-return temperature difference. Since the system is to deliver 60 Mbh with a temperature difference of 20 deg, the pump must circulate 60 ÷ 10, or 6 gpm.

For this load, the unit friction for a 1¼ in. pipe is 86 milinches. The main circuit consists of 110 ft of pipe and 10 elbow equivalents and may be placed equal to 136 ft of 1¼ in. pipe. The total friction for the main circuit is 136 x 86, or 11,696 milinches, or practically 1 ft for a flow of 6 gpm.

At the points *A*, a portion of the water will be diverted through the radiator circuit,

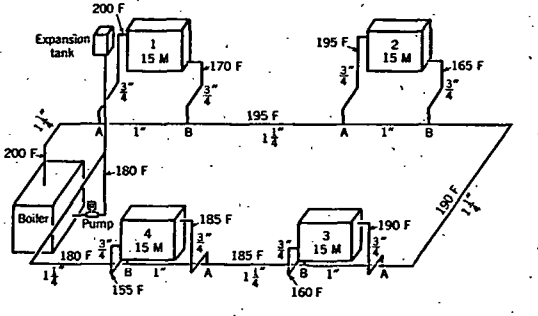


FIG. 14. ONE-PIPE FORCED CIRCULATION SYSTEM

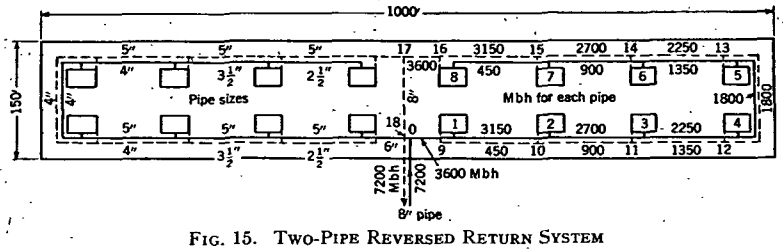


FIG. 15. TWO-PIPE REVERSED RETURN SYSTEM

and as a result less than 6 gpm will flow in the main between the points *A* and *B*, and the friction will be slightly less than 86 milinches per foot between these points. But the difference will be so small that it may be neglected and the total friction from *A* to *B* assumed to be 4 x 86, or 344 milinches. The head forcing the water through the radiator circuit will then be 344 milinches. The radiator circuit consists of 11 ft of pipe and about 14 elbow equivalents and may be placed equal to 32 ft of pipe and the head 344 ÷ 32, or about 11 milinches per foot. With this head, a ¾ in. pipe will convey about 5 Mbh (Fig. 4), or 0.5 gpm. Hence, only 5 ÷ 60, or about 8 per cent of the water, would flow through the radiator, if the radiator's gravity head is not considered. The water would, then, have to cool 60 deg in order to deliver 15 Mbh, and the average radiator temperature would be 170 if the water entered at 200. This would require a large radiator and result in an unsatisfactory installation.

To secure a larger flow of water through the radiator it is necessary to increase the friction of the short path *A-B* in the main. This may be done by inserting special resistance tees at points *A* and *B*, or by inserting an orifice resistor between points *A* and *B*, or by reducing the 1¼ in. main between the points *A* and *B* to the next smaller size, *i.e.* 1 in.

The relative quantity of water flowing through the radiator may then be found by trial calculations. Assume, first, that 1 gpm will flow through the radiator and 5 gpm through the main. The friction for 1 gpm and a ¾ in. pipe is 40 milinches per foot, or 32 x 40, or 1280 for the radiator circuit.

The main circuit consists of 4 ft of 1 in. pipe and two reducing tees. The two reducing tees may be placed equal to 0.8 elbow equivalents (Table 4), and the equivalent length of the main circuit equal to 5.7 ft.

The friction for 5 gpm and a 1 in. pipe is 240 milinches per foot, or 5.7 x 240, or 1370 milinches for the main circuit. Since this is only slightly more than the calculated friction for the radiator circuit, it is evident that the flow through the radiator will be slightly more than 1 gpm, and it is not necessary to make a second trial calculation. The quantity of water flowing through the radiator can be varied by varying the distance between the points *A* and *B*, where the radiator branches join the main.

In order to deliver 15 Mbh to the radiator with a temperature difference of 20 deg, it is necessary that 1.5 gpm flow through the radiator; since, in this case, the flow through the radiator is only 1 gpm, the temperature difference must be 30 deg.

If the water enters the radiator at 190 F, the average water temperature will be 175 F. The quantity of water circulating through the radiator may be varied considerably without an appreciable effect on the quantity of heat dissipated by the radiator. This is evident from the following calculation.

Assume that Radiator 2 has been designed so that it will dissipate 15 Mbh when its flow of water is at the rate of 1 gpm, and when its average temperature is 175 F. Assume

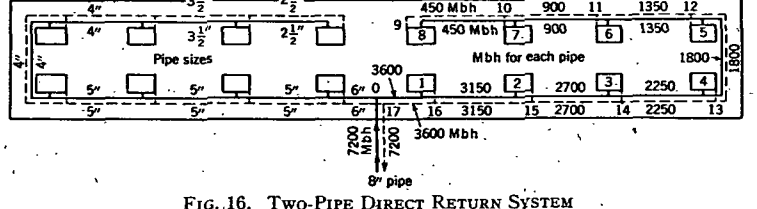


FIG. 16. TWO-PIPE DIRECT RETURN SYSTEM

that the flow of water is increased 50 per cent—from 1 gpm to 1.5 gpm. The water will then flow through the radiator in two-thirds the time and will cool two-thirds as much; *i.e.*, it will cool 20 deg instead of 30 deg, and the average radiator temperature will be 180 F instead of 175 F. If the surrounding temperature is 70 F, the temperature differences, radiator and surroundings, will be 110 and 115 deg, respectively. Consequently, the heat dissipation will be increased only about 6 per cent when the quantity of water flowing through the radiator is increased 50 per cent.

By decreasing the main from 1¼ to 1 in. between radiator branches while the flow is decreased from 6 to 5 gpm, the friction in that section of the main is increased from 344 to 1370 milinches, or 926 milinches. Hence, for the four radiator sections the increase is 3704 milinches, and the total friction for the circuit will be 11,696 plus 3704, or 15.4 in. instead of 11.7 in. as first calculated. The pump must, therefore, circulate 6 gpm against a head of 1.3 ft.

Reversed and Direct Return Systems

In a reversed return system the radiators are connected so that all circuits are practically of equal length and so that the water flowing through the radiator nearest the boiler must travel practically as far as the water flowing through the radiator farthest from the boiler, as illustrated in Fig. 15. In a direct return system the radiators are connected so that all water returns to the boiler along the most direct path after it has passed through its radiator, as illustrated in Fig. 16.

Example 7. In Fig. 15, sixteen air conditioning units, each demanding 450 Mbh, are to be supplied with water from a central plant. The system is divided into two equal parts as shown. Each part supplies eight units and has, therefore, eight circuits. The total length of each of the eight circuits is about 1170 ft.

Solution. If the total friction is to be about 15 ft, the unit friction must be about 150 milinches per foot. With this preliminary estimate, pipe sizes may be selected from Fig. 4 and recorded with corresponding calculations as shown in Table 7 for Circuit 8, from which it appears that the friction of this circuit is 188,570 milinches, or 15.7 ft.

In order that each of the eight air conditioning units may receive an equal supply of water, the friction of each of the remaining circuits must also be 15.7 ft. Since all pipe sizes have been selected as shown in Example 6, any adjustments that may be necessary must be made in the connections from the main through the air conditioning unit and back to the main. For Circuit 8 the friction through the air conditioning unit was assumed to be 20,000 milinches. For Circuit 4, for example, a tabular calculation like that for Circuit 8 shows that the friction for Section 4-12 must be 19,700 milinches in order that the total friction may be 15.7.

This is practically equal to the 20,000 milinches assumed for Circuit 8 and this shows how simple it is to secure well-balanced circuits in a reversed return system.

Example 8. The direct return, forced circulation system shown in Fig. 16 is similar to Fig. 15, except that the water passing through Unit 1 returns directly to the boiler and the total length of this circuit is about 130 ft, whereas, the total length of Circuit 8 is about 1990 ft, or about 15 times as long.

Solution. The design must begin with Circuit 8. The calculations for this circuit if tabulated as shown for Circuit 8 of Fig. 15 will show that the total friction is 318,200 milinches, or 26.5 ft, as compared with 15.7 ft for the reversed return system.

In order that each of the eight units will receive its correct share of the water, the friction of each of the other seven circuits must also be 26.5 ft. For Circuit 1, for example, the total friction for Section 0-1 and 16-17 is 15,600 milinches; hence, the friction in Section 1-16 (through the air conditioning unit) must be 302,600 milinches, or 25.15 ft to prevent Unit 1 having an advantage over Unit 8.

Comparing the reversed return system of Fig. 15 with the direct return system of Fig. 16 it appears that the head against which the pump must deliver the 720 gpm is 15.7 ft as compared with 26.5 ft for the direct return and that the installation of the reversed return would require 130 ft of 8 in. pipe not necessary for the direct return system. The friction in the lines joining the pump to the pipe system shown in Figs. 15 and 16 is not included in this calculation.

CHAPTER 25

Radiators and Convectors

Heat Emission of Radiators and Convectors, Types of Radiators, Convectors, Radiator and Convector Ratings, Effect of Operating Conditions, Heating Effect, Heating Up the Radiator and Convector, Enclosed Radiators

THE accepted terms for heating units are: (1) *radiators*, for direct surface heating units, either exposed, enclosed, or shielded, which emit a large percentage of their heat by radiation; and (2) *convectors*, for heating units having a large percentage of extended fin surface and which emit heat principally by convection. Convectors are dependent upon enclosures to provide the circulation by gravity of large volumes of air.

HEAT EMISSION OF RADIATORS AND CONVECTORS

Most heating units emit heat by *radiation* and *convection*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits roughly half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, the proportion of radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible.

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not a true index of output. (The engineering unit of output is the *Mbh* or 1000 Btu per hour.) However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet*. For steam service this is based on an emission of 240 Btu per (hour) (square foot) and for hot water service 150 Btu.

TYPES OF RADIATORS

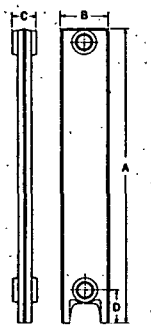
Present day radiators may be classified as tubular, wall, or window type and are generally made of cast-iron. Only the small-tube type of tubular radiators with a spacing of 1¾ in. per section are now available, the large-tube type which had a spacing of 2½ in. per section having been discontinued. Small-tube radiators occupy less space and are particularly suited for installation in recesses.

After a complete 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 R174-43 for small-tube cast-iron radiators. Table 1 shows the size and dimensions of small-tube cast-iron radiators which are being manufactured at the present time.

Wall radiators are now rated in terms of equivalent square feet, the same as small-tube radiators. Tests have shown that the heat emitted

TABLE 1. 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]	
			Minimum	Maximum			
	Sq Ft	In.	In.	In.	In.	In.	
3[d]	1.6	25	3¼	3½	1¾	2½	
4[d]	1.6	19	4 7/16	4 13/16	1¾	2½	
	1.8	22	4 7/16	4 13/16	1¾	2½	
	2.0	25	4 7/16	4 13/16	1¾	2½	
5[d]	2.1	22	5 5/8	6 5/16	1¾	2½	
	2.4	25	5 5/8	6 5/16	1¾	2½	
6[d]	1.6	14	6 13/16	8	1¾	2½	
	2.3	19	6 13/16	8	1¾	2½	
	3.0	25	6 13/16	8	1¾	2½	
	3.7	32	6 13/16	8	1¾	2½	



[a]The square foot of equivalent direct steam radiation is defined as the ability to emit 240 Btu per hour, with steam at 215 F, in air of 70 F. These ratings apply only to installed radiators exposed in a normal manner; not to radiators installed behind enclosures, grilles, etc. (See A.S.H.V.E. Code for Testing Radiators adopted January, 1927.)

[b]Length equals number of sections times 1¾ in.

[c]Over-all height and leg height, as produced by some manufacturers, are one inch (1 in.) 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 4¼ in.

[d]Or equal.

from a wall-type radiator may be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually such a large difference in the temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of the room satisfactorily.

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but now wall-type radiators are most frequently used for this service. When coils are used, the miter type assembly is to be preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2. This has been developed from avail-

TABLE 2. 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

able data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

CONVECTORS

Cast-iron radiators may be concealed in a cabinet or other enclosure for appearance. In such cases a greater percentage of heat is conveyed to the room by convection thereby resulting in a form of gravity convector. A typical recessed convector is shown in Fig. 1. The heating element consisting of a large percentage of fin surface is usually shallow in depth and placed low in the enclosure in order to produce maximum chimney effect in the enclosure. The air enters the enclosure near the floor line just below the heating element, is moderately heated in passing through

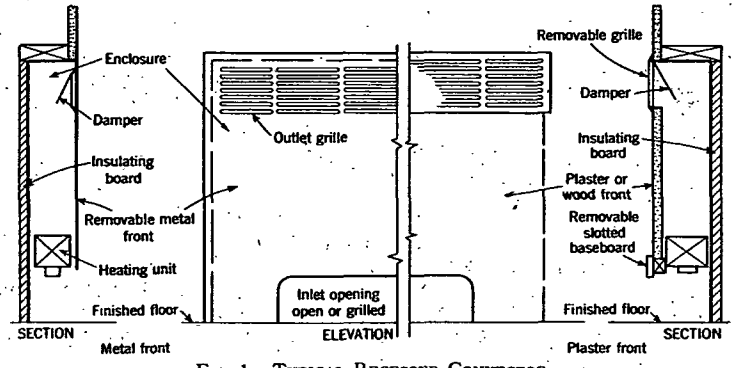


FIG. 1. TYPICAL RECESSED CONVECTOR

the core and delivered to the room through an opening near the top of enclosure. Since the air can only enter the enclosure at the floor line, the cooler air in the room, which always lies at this level, is constantly being withdrawn and replaced by the warmer air. This air movement accomplishes the desired reduction in temperature differentials and assures maximum comfort in the living zone.

Concealed heaters or convectors are generally available as completely built-in units. The enclosing cabinet should be designed with suitable air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are cataloged for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall-hung type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille in a plane parallel with the floor, although the front outlet is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector. It is desirable that the enclosure or housing for the convector fit as snugly as possible

so that the air to be heated must pass through the convector and cannot be by-passed in the enclosure.

The output of a convector, for any given length and depth, is a function of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation. When more than one heating unit is used, one mounted above the other in the same cabinet, the output of the upper unit or units will be materially less than that of the bottom unit.

RADIATOR AND CONVECTOR RATINGS

A standard method of testing radiators was adopted by the A.S.H.V.E. in 1927 [1]. This Code 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 lb per square inch absolute. 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. This output divided by 240 gives the steam rating of the radiator in square feet.

Similar test methods for convectors are the A.S.H.V.E. Codes for Testing and Rating Concealed Gravity Type Radiation [2], (Steam Code 1932 and Hot Water Code 1933). These Codes recognize a different type of test booth, and the air temperature used is that of the air entering the convector casing instead of the temperature in the center of the room. The entering air temperature for standard test conditions is 65 F. For hot water the standard test conditions call for a mean temperature of the water in the convector of 170 F.

The *Convector Manufacturers Association* has adopted the A.S.H.V.E. standard in the formulation of its ratings and has compiled a tentative standard of heating effect allowances for various enclosure heights to be included in the ratings by its members.

All published ratings bearing the title *C.M.C. Ratings* (*Convector Manufacturers Certified Ratings*) indicate that the convectors have been tested in accordance with the A.S.H.V.E. Code by an impartial and disinterested laboratory and that the ratings have been approved by the Standardization Committee of the *Convector Manufacturers Association*.

Effect of Operating Conditions

The heat output of a radiator is proportional to the 1.3 power of the temperature difference between the air in the room at the 60 in. level and the heating medium in the radiator. The heat output of a convector is proportional to the 1.5 power of the temperature difference between the air entering the convector and the heating medium, steam or hot water, within the convector [3]. For hot water the arithmetical average between entering and leaving water temperatures is used. These laws may be expressed as correction factors to change from output under standard

TABLE 3. 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	Factors for Convectors — Inlet Air 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	3.14	2.83	2.57	2.35	2.15	1.98	1.84
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44	2.57	2.35	2.15	1.98	1.84	1.71	1.59
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28	2.15	1.98	1.84	1.71	1.59	1.49	1.40
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15	1.84	1.71	1.59	1.49	1.40	1.32	1.24
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05	1.59	1.49	1.40	1.32	1.24	1.17	1.11
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96	1.40	1.32	1.24	1.17	1.11	1.05	1.00
Lb per Sq In.																
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85	1.17	1.11	1.05	1.00	0.95	0.91	0.87
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76	1.00	0.95	0.91	0.87	0.83	0.79	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66	0.83	0.79	0.76	0.73	0.70	0.68	0.65
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58	0.70	0.68	0.65	0.63	0.60	0.58	0.56
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49	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, divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table.

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 capacities at the basic conditions by the proper factor from the above table.

rating-test conditions to output under other operating conditions. Such factors are given in Table 3.

When it is desired to change the output under any test conditions to the corresponding output under standard Code test conditions, the reciprocal form of correction factor may be derived. The equations for steam units are:

For radiators:

$$C_s = \left(\frac{215 - 70}{t_s - t_r}\right)^{1.3} \qquad (1)$$

For convectors:

$$C_s = \left(\frac{215 - 65}{t_s - t_i}\right)^{1.5} \qquad (2)$$

The output under standard conditions will be:

$$H_s = C_s H_t \qquad (3)$$

where

C_s = correction factor.
t_s = steam temperature during test, Fahrenheit degrees.
t_r = room temperature during test, Fahrenheit degrees.
t_i = inlet air temperature during test, Fahrenheit degrees.
H_s = heat emission rating under standard conditions, Btu per hour.
H_t = heat output under test conditions, Btu per hour.

The relation between the size of the radiator or convector and the size of the test room will affect the results obtained in a capacity-rating test.[4] The height and location of the radiator and the insulation of the test room are other important factors that are not specifically regulated by the Code.

For a radiator, 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 per cent or more in the total heat output of the radiator [5,6,7].

Radiator enclosures and convector casings affect the heat distribution within the room as well as the total amount of heat supplied by the steam or hot water [8].

Heating Effect

For several years the term *heating effect* has been used to designate the relation between the *useful output* of a radiator, in the comfort zone of a room, and the total input as measured by steam condensation or water temperatures [9,10]. The application of such a heating effect factor is a recognition that some radiators and convectors use less steam than others for producing equal comfort heating results in the room.

No standard method for evaluating the heating effect of radiators and convectors and correlating it with comfort has yet been accepted. One method, with test data [11] on radiators and convectors, and making use of the eupatheoscope for evaluating the environment produced, has been suggested by the University of Illinois. The principle underlying the eupatheoscope involves the measurement of the heat loss from a sizable

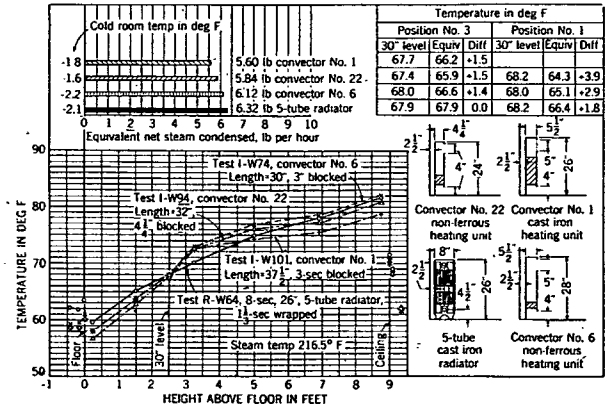


FIG. 2. TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR RADIATOR AND CONVECTORS WITH COMMON 30 IN. LEVEL TEMPERATURE

body by radiation and convection, when the surface is maintained at some constant temperature. Through the use of this instrument and its calibration curve, non-uniform environments may be referred to uniform environments in which the air and all surrounding surfaces are at the same temperature. The temperatures of the uniform environments are referred to as equivalent temperatures.

The Kata thermometer [12], the thermo-integrator [13,14], and the globe [15] thermometer are other instruments which have been used to measure the influence of air temperature, air movement and radiation in an environment.

Data given in Fig. 2 show that while the air temperature at the 30-in. level is the same for the three convectors and the one large-tube cast-iron radiator, in position No. 3 in the test room, the equivalent temperature is 1.5 F lower than the air temperature in the case of the three convectors, and the same as the air temperature in the case of the radiator. The difference between the minimum and the maximum amount of heat

required to maintain the common air temperature at the 30-in. level is of the order of 13 per cent.

In Fig. 3 are shown the results of tests made with the same three convectors and the one large-tube cast-iron radiator, so adjusted in size that each gave approximately the same equivalent temperature in the No. 3 position in the test room. The difference between the minimum and the maximum amount of heat required to maintain the common equivalent temperature is of the order of 7 per cent.

The following statements applying to the use of radiators are based on experience and test results:

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.

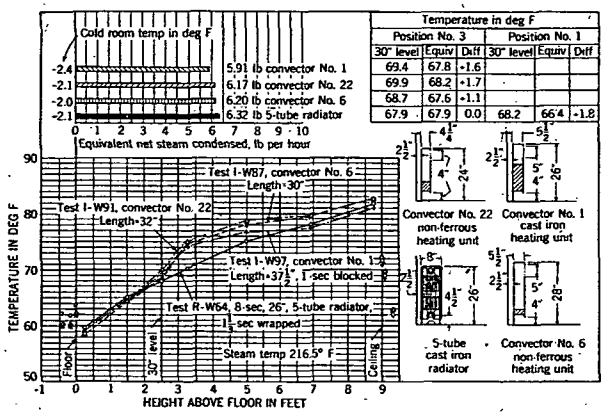


FIG. 3. TEMPERATURE GRADIENTS AND EQUIVALENT TEMPERATURES FOR RADIATOR AND CONVECTORS WITH COMMON EQUIVALENT TEMPERATURE

2. Smaller floor-to-ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.

3. The larger portion of the floor-to-ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.

4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room. The comfort-indicating temperature should be taken below the breathing line level.

5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath windows.

HEATING UP THE RADIATOR AND CONVECTOR

The maximum condensation occurs in a heating unit when the steam is first turned on. Tests [16] on an old-style column-type cast-iron radiator indicated that in the first 10 min the condensation rate reached a peak of 0.95 lb per square foot of radiator per hour and 10 to 15 min later lowered

to a rate of 0.24 lb. In practice the rate of steam supply to the heating unit, while heating up, is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard the supply of steam. Vacuum types of air venting valves may be used to reduce the length of the venting periods.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heating effect. Investigations [10] indicate that in the design of the enclosure three things should be considered:

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.

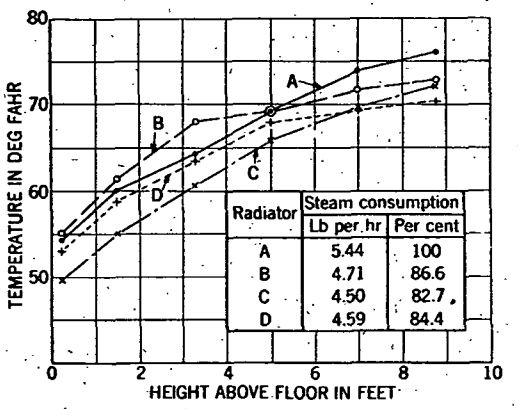


FIG. 4. STEAM CONSUMPTION OF EXPOSED AND CONCEALED RADIATORS

2. The lessened steam consumption may not materially change the radiator heating performance.

3. The enclosed radiator may inadequately heat the space.

A comparison between a bare or exposed radiator (*A*) and the same radiator with a well-designed enclosure (*B*), with a poorly-designed enclosure (*C*), and with a cloth cover (*D*) will illustrate the relative heating effects. In Fig. 4 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly-designed enclosures. Curve (*D*) shows that the effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

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 [17] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from

such pans, when the radiator is steam hot and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

REFERENCES

1—A.S.H.V.E. Code for Testing Radiators (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 18).

2—A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam), (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367); (Hot Water), (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 237). (See also A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 38, and Vol. 42, 1936, p. 29).

3—A.S.H.V.E. RESEARCH REPORT No. 998—Factors Affecting the Heat Output of Convectors, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 443).

4—Factors Influencing the Heat Output of Radiators, by A. C. Davis, W. M. Sawdon and David Dropkin (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1942, p. 185) and Cornell University, *Engineering Experiment Station Bulletin* No. 29, April, 1942.

5—Heat Emission from Radiators, by K. F. Rubert (Cornell University, *Engineering Experiment Station Bulletin* No. 24, 1937).

6—Comparative Tests of Radiator Finishes, by W. H. Severns (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 41).

7—Heat Loss from Direct Radiation, by J. R. Allen (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 11).

8—Heat Output of Concealed Radiators, by E. A. Allcut (University of Toronto, *School of Engineering, Research Bulletin* No. 140, 1933).

9—The Heating Effect of Radiators, by Charles Brabbée (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 33).

10—University of Illinois, *Engineering Experiment Station Bulletins* Nos. 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 77).

11—A.S.H.V.E. RESEARCH REPORT No. 962—The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303).

12—The Kata Thermometer—Its Value and Defects, by W. J. McConnell and C. P. Yagloglou. (Reprint No. 953 from *U. S. Public Health Service Report*, pp. 2293-2337, September 5, 1924).

13—The Thermo-Integrator—A New Instrument for the Observation of Thermal Interchanges, by C.-E. A. Winslow and Leonard Greenburg (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 149).

14—The Calibration of the Thermo-Integrator, by C.-E. A. Winslow, A. P. Gagge, Leonard Greenburg, I. M. Moriyama and E. J. Rodee. (*The American Journal of Hygiene*, Vol. 22, No. 1, July, 1935, pp. 137-156).

15—The Globe Thermometer in Studies of Heating and Ventilation, by T. Bedford and C. G. Warner. (*The Journal of Hygiene*, Vol. 34, No. 4).

16—A.S.H.V.E. RESEARCH REPORT No. 1067—The Cooling and Heating Rates of a Room with Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 389).

17—University of Illinois, *Engineering Experiment Station Bulletin* No. 230, p. 20.

CHAPTER 26

Unit Heaters, Unit Ventilators, Unit Humidifiers

Definitions; Unit Heaters: Classification, Application, Outlet Velocities, Ratings for Various Types, Temperature, Location, Automatic Control, Piping Connections; Boiler Capacity for Steam Unit Heaters; Unit Ventilators: Ratings, Applications, Location, Exhaust Vents; Window Ventilators; Unit Humidifiers

DESCRIPTIONS of heating, cooling, ventilating, humidifying, and dehumidifying systems are given in other chapters. This chapter deals with unit heaters, unit ventilators, and unit humidifiers. Cooling units, unit air conditioners, and attic fans are described in Chapter 36.

Definitions

The generally accepted meaning of the word *unit* in the terms unit heaters, unit ventilators and unit humidifiers is that of a factory made, encased assembly of the functional elements indicated by its name. Such units can be shipped complete or in sections so that the only field work necessary is the assembling of the sections, providing proper supports and connecting the unit to sources of heat (or fuel), power and water supply, and, if necessary, to vent pipes for combustion gases.

The term *unit heater* describes an assembly of elements whose principal function is heating. The essential elements of a unit heater are a fan and motor, a heater, a housing, and outlet vanes or diffusers.

The term *unit ventilator* describes an assembly whose principal function is to ventilate. It may serve to circulate air within the space, or to introduce air from without the space, or may accomplish both purposes. The essential elements of a unit ventilator are a fan and motor, a heater, a set of dampers, a housing and outlet vanes or diffusers.

The term *unit humidifier* describes an assembly of elements whose principal function 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 HEATERS

Classification

The various types of unit heaters which are at present available can usually be classified according to one of the three following methods:

1. *By type of heater.* Under this classification there are three types of heating elements to be considered: (*a*) *the steam or hot water type*, (*b*) *the electric type*, and (*c*) *the direct fired type* which may be *gas*, *oil*, or *coal fired*.

2. *By type of fan.* Under this classification there are two types of fans to be considered: (*a*) the *propeller type* [1] which may be equipped with a horizontal or vertical shaft, and (*b*) the *centrifugal type* which may be designed for *horizontal* or *vertical blow*.

3. *By arrangement of elements.* Under this classification there are two types of heaters 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 heater.

Unit heaters are available in any combination of the three preceding general classifications. For example, the steam or hot water type may be secured with either the propeller or centrifugal type of fans, and in either the draw-through or blow-through type.

Unit heaters also vary in other minor respects. For example, steam and return inlets and outlets may be located on the top and bottom respectively or on the same side of the unit. Some units are supported by the piping and some have independent supports. The heating surface of steam or water type units is generally made up of a non-ferrous tube-and-fin assembly; or it may be fabricated of steel, or cast in steel.

Application of Unit Heaters

Unit heaters are used principally for commercial and industrial applications such as garages, factories, laboratories, etc. They may also be used for heating finished rooms, if properly applied and concealed and if some consideration is given to the problem of noise.

Unit heaters may also be adapted 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 unit heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. See discussion of condensation in Chapter 6.

There are three major factors to consider in the application of unit heaters, namely: (1) location of unit, (2) air distribution, and (3) heating medium.

There are a variety of applications which are favorable to the use of *electric unit heaters*. For supplemental heat in residence bath rooms, for the heating of ticket booths, watchmen's offices, factory offices, locker rooms and other isolated rooms scattered over large areas, their use is peculiarly adaptable. They are particularly useful in isolated and unattended pumping stations or pits where they may be thermostatically controlled to prevent freezing temperatures.

Gas fired unit heaters find application in industrial plants, offices, stores, garages, in fact in almost every location where steam type units are used. The installation cost of gas fired units is usually less than that of a type requiring that a new boiler be installed.

Oil fired unit heaters are used in industrial plants, garages and commercial buildings.

Coal fired unit heaters are of finned, welded steel, or cast-iron construction and equipped with centrifugal blowers. They are usually stoker-fired to insure proper firing of fuel. They are used principally in large industrial plants such as foundries or assembly plants, and provide a very economical source of heat, both from the installation and the operating standpoint.

Outlet Velocities

Outlet velocities of unit heaters vary from about 400 to 2500 fpm depending upon the type of unit and the distance to which the air is to be

projected. Noise and drafts must be considered in the choice of air velocities, since both increase with increase of air velocity.

Velocities and decibel ratings for the various types of unit heaters illustrated in Figs. 1, 2, 3 and 4 are given in Table 1.

In the selection of unit heaters it is important not to exceed the throw for which the unit was designed.

These are dependent to a marked degree on the temperature of air leaving the heater as well as upon velocities. See discussion under heading of Inlet, Outlet, and Space Temperatures with Unit Heaters.

Air Outlets

In order to direct the air to points desired and to diffuse the air to avoid drafts, it is common practice to equip unit heaters with directional outlets, adjustable louvers or fixed types of diffusers.

TABLE 1. OUTLET VELOCITIES, DISTANCE OF BLOW AND DECIBEL RATINGS FOR VARIOUS TYPES OF UNIT HEATERS

TYPE OF UNIT HEATER	OUTLET VELOCITIES FPM	DISTANCES OF BLOW—FT	DECIBEL RATING
Centrifugal Fan	1500–2500	20–200	34–90
Horizontal Propeller Fan	400–1000	30–100	26–84
Vertical Propeller Fan	1200–2200	70	50–82

RATINGS OF UNIT HEATERS

It is standard practice to rate 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. This applies to all types of unit heaters, the steam or hot water type, the electric type and the direct fired type. There are, however, other factors which must be taken into account, especially when an attempt is made to compare one type of heater with another. These are the temperature of the heating element and the velocity of air through it. Consideration is given to these factors in the discussion of ratings for each type of unit heater in the following paragraphs.

Ratings for Steam Type

The rating of steam type unit heaters has been standardized by a code [2] in which the following items are the basis of rating: dry saturated steam at 2 psig pressure (29.92 in. Hg barometric pressure) at the heater coil; air at 60 F 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 be calculated approximately from any given rating by the use of factors in Tables 2 and 3. Table 2 is used for the blow-through type and Table 3 for the draw-through type of unit.

Ratings for Hot Water Type

A standard for the rating of hot water type unit heaters has also been established by code [3] in which the following items are the basis of rating:

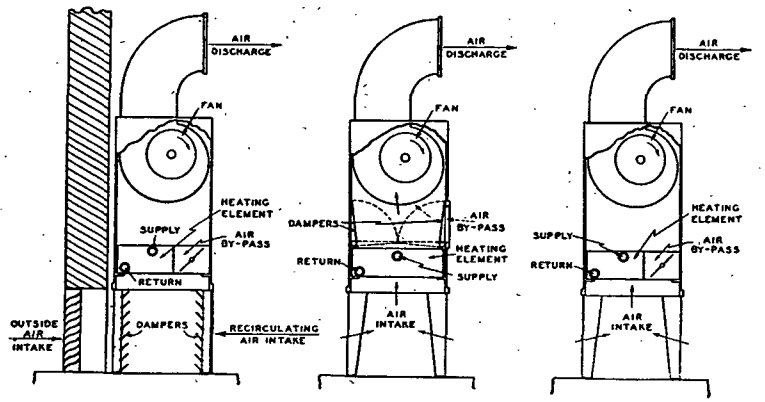


FIG. 1. CENTRIFUGAL FAN TYPE UNIT HEATER—FLOOR MOUNTED

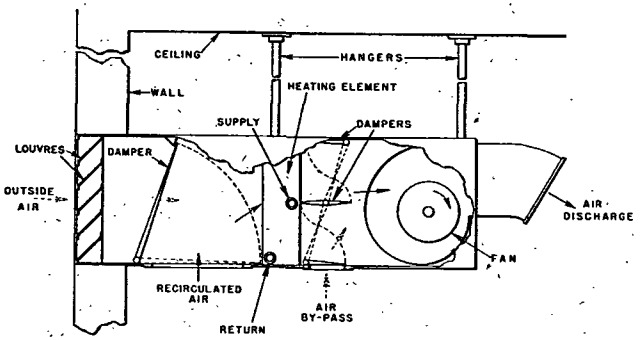


FIG. 2. CENTRIFUGAL FAN TYPE UNIT HEATER—SUSPENDED

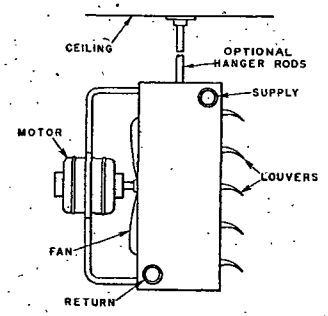


FIG. 3. PROPELLER FAN TYPE UNIT HEATER—HORIZONTAL BLOW

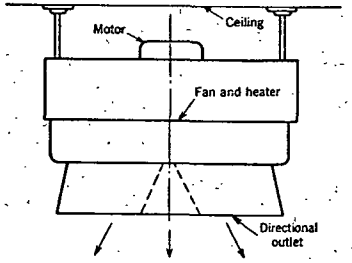


FIG. 4. PROPELLER FAN TYPE UNIT HEATER—VERTICAL BLOW

entering water at 200 F; entering air at 60 F (29.92 in. Hg barometric pressure); and heater operating free of external resistance to air flow. This code also prescribes a method of translating the output in Btu and the temperature rise as obtained under test conditions to standard conditions of air and water temperature.

Ratings for Electric Type

Electric type unit heaters are usually limited in size to a maximum of about 9 kw capacity. They consist of resistance type heating elements combined with fan and motor, together with a suitable casing. Electric unit heaters are made in the built-in-wall model, suspension model, and free-standing or portable model.

Electric unit heaters are rated on the energy input to the heater, expressed in terms of kilowatts, Btu or EDR. Quite often all three ratings are given in parallel columns in the catalogs.

Ratings for Gas Fired Type

Gas fired type unit heaters are built with either the propeller or centrifugal type fans. They are available in a wide range of sizes from 45,000 to 1,650,000 output capacity and in suspended or floor models.

Gas fired unit heaters are usually rated in terms of both input and output according to the approval requirements of the *American Standards Association.*

Ratings for Oil Fired Type

The oil fired type of unit heater is usually equipped with the centrifugal type of fan only and can be secured in sizes ranging from 125,000 to 1,650,000 Btu per hour output capacity in standard units. It is furnished in either the floor mounted or in smaller sizes in the suspended type.

Ratings for Coal Fired Type

The stoker fired type of unit heater can be secured in ranges of from 300,000 to 6,000,000 (or more) Btu per hour output capacity. Ratings are based upon the delivered output at the heater outlet.

Effect of Resistance Upon Capacity

Unit heaters are customarily rated as free delivery type units. If outside air intakes, air filters, or ducts on the discharge side are used with the unit, a reduction in air and heating capacity will result because of this added resistance. The percentage of 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. In general, however, propeller fan type units will experience a larger reduction in capacity than housed centrifugal fan units for a given added resistance and a given heater will have a larger reduction in capacity as the fan speed is lowered. The heat output to be expected under other than free delivery conditions should be secured from the manufacturer.

Inlet, Outlet and Space Temperatures with Unit Heaters

In the selection of unit heaters for any particular design, consideration should be given to the temperature of air entering the heaters as well as the temperature to be maintained in the working zone of the space. In

TABLE 2. CONSTANTS FOR DETERMINING THE CAPACITY OF BLOW-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 psig and Entering Air Temperature of 60 F)

STEAM PRESSURE PSI	TEMPERATURE OF ENTERING AIR											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1.446	1.369	1.273	1.191	1.110	1.034	0.956	0.881	0.809	0.739	0.671
2	1.585	1.495	1.405	1.320	1.237	1.155	1.078	1.000	0.926	0.853	0.782	0.713
5	1.640	1.550	1.456	1.370	1.289	1.206	1.127	1.050	0.974	0.901	0.829	0.760
10	1.730	1.639	1.545	1.460	1.375	1.290	1.211	1.131	1.056	0.982	0.908	0.838
15	1.799	1.708	1.614	1.525	1.441	1.335	1.275	1.194	1.117	1.043	0.970	0.897
20	1.861	1.769	1.675	1.584	1.498	1.416	1.333	1.251	1.174	1.097	1.024	0.952
30	1.966	1.871	1.775	1.684	1.597	1.509	1.429	1.346	1.266	1.190	1.115	1.042
40	2.058	1.959	1.862	1.771	1.683	1.596	1.511	1.430	1.349	1.270	1.194	1.119
50	2.134	2.035	1.936	1.845	1.755	1.666	1.582	1.498	1.416	1.338	1.262	1.187
60	2.196	2.094	1.997	1.902	1.811	1.725	1.640	1.555	1.472	1.393	1.314	1.239
70	2.256	2.157	2.057	1.961	1.872	1.782	1.696	1.610	1.527	1.447	1.368	1.293
75	2.283	2.183	2.085	1.990	1.896	1.808	1.721	1.635	1.552	1.472	1.392	1.316
80	2.312	2.211	2.112	2.015	1.925	1.836	1.748	1.660	1.577	1.497	1.418	1.342
90	2.361	2.258	2.159	2.063	1.968	1.880	1.792	1.705	1.621	1.541	1.461	1.383
100	2.409	2.307	2.204	2.108	2.015	1.927	1.836	1.749	1.663	1.581	1.502	1.424

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering air and 2 psi. When increasing steam pressure it is important to determine whether the heater is suitable for the increased pressure application and whether the resulting increased outlet temperature is satisfactory.

TABLE 3. CONSTANTS FOR DETERMINING THE CAPACITY OF DRAW-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2 psig and Entering Air Temperature of 60 F)

STEAM PRESSURE PSI	TEMPERATURE OF ENTERING AIR											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.483	1.405	1.329	1.253	1.178	1.105	1.032	0.962	0.892	0.822	0.754	0.688
2	1.520	1.442	1.363	1.290	1.215	1.141	1.069	1.000	0.930	0.861	0.792	0.728
5	1.565	1.485	1.410	1.334	1.260	1.187	1.114	1.045	0.975	0.906	0.838	0.771
10	1.637	1.558	1.480	1.403	1.328	1.253	1.182	1.112	1.042	0.973	0.903	0.838
15	1.688	1.610	1.533	1.458	1.382	1.310	1.239	1.168	1.099	1.028	0.960	0.895
20	1.728	1.649	1.572	1.498	1.421	1.350	1.278	1.208	1.138	1.070	1.002	0.936
30	1.803	1.725	1.648	1.572	1.497	1.423	1.352	1.281	1.212	1.145	1.078	1.010
40	1.864	1.787	1.710	1.637	1.563	1.491	1.420	1.350	1.282	1.215	1.148	1.081
50	1.927	1.850	1.773	1.700	1.628	1.554	1.483	1.416	1.347	1.278	1.211	1.145
60	1.973	1.897	1.820	1.748	1.673	1.601	1.531	1.463	1.394	1.325	1.260	1.194
70	2.018	1.943	1.869	1.795	1.722	1.651	1.582	1.512	1.443	1.377	1.310	1.243
75	2.043	1.970	1.895	1.822	1.750	1.680	1.609	1.540	1.471	1.402	1.333	1.268
80	2.064	1.988	1.914	1.841	1.770	1.698	1.629	1.560	1.491	1.422	1.354	1.288
90	2.102	2.028	1.951	1.878	1.804	1.732	1.661	1.590	1.523	1.457	1.387	1.321
100	2.150	2.071	1.994	1.919	1.845	1.770	1.700	1.630	1.560	1.492	1.425	1.359

Note: To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering air and 2 psi. When increasing steam pressure it is important to determine whether the heater is suitable for the increased pressure application and whether the resulting increased outlet temperature is satisfactory.

general, the temperature differences per foot of elevation, when using unit heaters are less than corresponding variations when using direct radiation[4]. High velocity units will maintain slightly lower temperature differences than low discharge velocity units. Correspondingly, units with lower discharge air temperature will maintain lower temperature differences than units with higher discharge temperatures. Directional control of the discharged air from a unit heater can be an important factor, added to qualities of reasonably good outlet velocity and outlet temperature, in effecting satisfactory distribution of heat and restraining floor-to-ceiling temperature difference.

When some *outside air* is introduced, the temperature of the mixture of outside and recirculating air must be calculated and used as the entering air temperature at the heater. Unit heaters connected in this manner perform the function of unit ventilators. For a discussion of this function see the section of this chapter entitled *Unit Ventilators*.

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

Thermostatic control of unit heaters may be accomplished by starting and stopping the fan or by controlling the flow of heat to the heating element. If the fan is controlled, it is advisable to provide a thermostat on the return from the heater so as to prevent the fan from starting until the heating element is heated throughout.

Unit heaters may be used in the summer time as a means of circulating the air to give some measure of comfort due to air motion. In such cases the heating element should be shut off from the source of heat by closing the steam valve to the heating element, and also the return line thermostat should be provided with a by-pass switch, which upon being closed will permit the fan to be operated independently of the heating element.

Location of Unit Heaters

Unit heaters should not be located in a corner or placed close to a wall, since the full effect of the unit will not be available under such conditions if the circulation of air is impeded. The best arrangement is to locate units so that they discharge air nearly parallel to exterior walls, and in a direction which will produce a rotational circulation around the room. This is preferable to directing the discharge against the outside walls.

Various types and makes of unit heaters are illustrated in the *Catalog Data Section* of this edition. As hot blasts of air in working zones are usually objectionable, heaters mounted on the floor should have their discharge outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close above the head line as possible, yet not into the working zone.

In connection with the use of vertical type unit heaters, care must be exercised in the selection of the heater. It has been found that the higher the unit is placed above the floor, the lower must be the outlet temperature of the air leaving the heater in order that the heated air may be forced into the occupied zone.

Determining Unit Heater Requirements

The formulas given in the section on Unit Ventilators may be used to determine unit heater capacity requirements.

PIPING CONNECTIONS FOR STEAM 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 themselves to function as intended. The basic piping principles for steam systems are discussed in Chapter 23.

Rapid condensation of steam, especially during heating-up periods, is

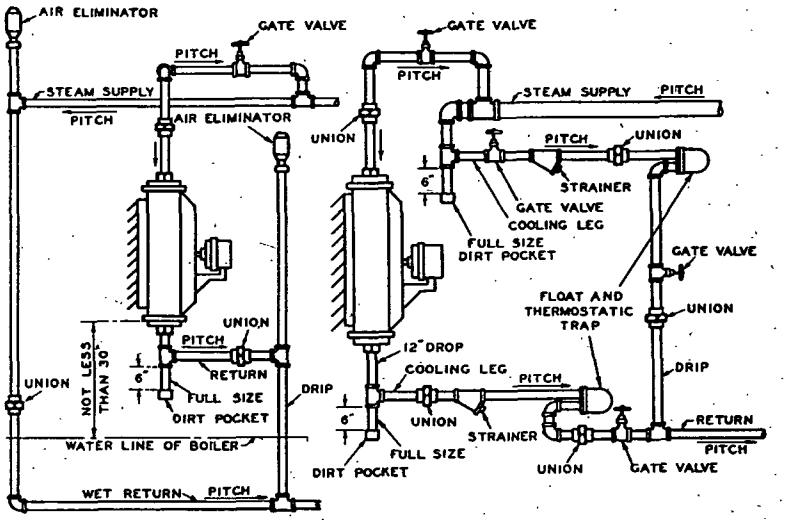


FIG. 5. UNIT HEATER CONNECTION TO ONE-PIPE GRAVITY STEAM SYSTEM

FIG. 6. UNIT HEATER CONNECTION FOR VACUUM OR VAPOR SYSTEM DISCHARGING CONDENSATION INTO DRY RETURN

characteristic of this type of equipment. The return piping must be planned to keep the heating coil free of rapid condensation, while 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 essential where a unit heater fan is operated under start-and-stop control and where all or part of the air is taken from the outside. In such installations the condensation rate may vary rapidly and the necessity for ample pipe capacity is particularly important.

A method of connecting a unit heater to a one-pipe gravity system is illustrated in Fig. 5. In cases where the return main is located above the boiler water line, an artificial water line must be created by providing an equalizing loop to prevent steam passing into the return and thus into other units.

A piping arrangement where both the air and condensate pass through a common return to a boiler, with vent trap or condensate pump and

receiver, is shown in Fig. 6. The traps must pass air and condensate rapidly to keep the return piping partially full of water.

Since unit heaters are often constructed with sufficient strength, the use of high pressure steam in them is a common practice. As shown in Fig. 7 the condensate and air reach the return overhead through traps, and check valves are located in the return piping. It is, however, preferable to locate the high pressure return below the heater.

For two-pipe closed gravity return systems, the return from each unit should be fitted with a heavy duty or blast trap, and an automatic air valve should be connected into the return header of each unit heater. Provisions must be made to compensate for the pressure drop by elevating the unit heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual

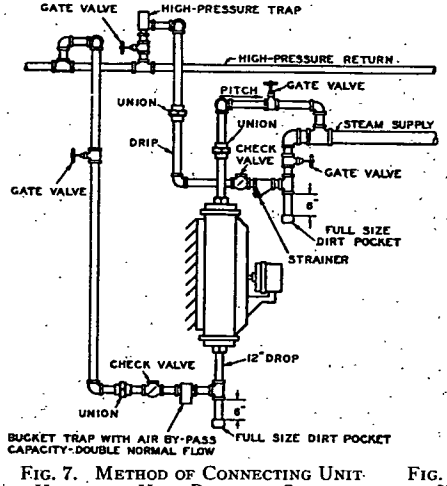


FIG. 7. METHOD OF CONNECTING UNIT HEATER TO HIGH PRESSURE SYSTEM

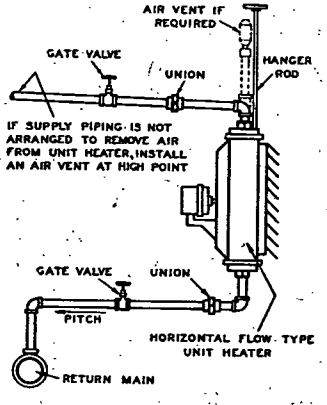


FIG. 8. METHOD OF CONNECTING UNIT HEATER TO HOT WATER SYSTEM

air valves on the heaters, or it may be carried into the returns as in vacuum systems and the entire return system be free-vented to the atmosphere, provided all units, drip points, and radiation are properly trapped to prevent steam entering the returns.

On vacuum or open vent systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, with its elimination at some other point in the system.

Fig. 8 represents the connections to a hot water heating system. The

air vent is not required if main is above heater and air can be eliminated through the piping system.

BOILER CAPACITY FOR STEAM UNIT HEATERS

The capacity of the boiler should be based on the rated capacity of the unit heaters at the lowest entering air temperature and highest fan speed that will occur, plus an allowance for pipe line losses. It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attention to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller unit heaters instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their heating surfaces are designed for those pressures, and when proper provision is made for returning the condensate. If units admit air that may be at a temperature below freezing, a steam pressure of not less than 5 lb should be maintained on the heating element, or a corresponding differential in pressure between the supply and returns should be maintained by means of a vacuum.

UNIT VENTILATORS

Unit ventilators are similar in principle to unit heaters, except that they are provided with an arrangement of dampers for introducing outdoor and recirculated air in varying quantities. In general, they are obtainable only in the steam type, but unit heaters of all types may be adapted to the unit ventilator principle by the addition of outdoor air and recirculating dampers. Unit ventilators are used to supply air with a discharge temperature at or slightly higher than room temperature. Also they are provided with an arrangement for introducing outdoor and recirculated air in varying quantities. If the unit is only used for circulating air, then radiators or some other equipment must be provided for heating the room. Unit ventilators are intended primarily for schools, offices, and semi-commercial establishments. Unit ventilators may be of the radiator controlled or damper controlled type and of the blow-through or draw-through type as illustrated in Fig. 9. A roof ventilator used for exhausting air is sometimes termed a unit ventilator. For information on roof ventilators, see Chapter 9.

Ratings

Unit ventilators of the steam type are customarily cataloged with two ratings, one the input and the other the output capacity. The first is the heat input to the unit which is determined by measuring the temperature and quantity of condensate and the pressure and quality of the steam. The second is the heat output of the unit which is determined by measuring the quantity of air delivered and the temperature of the air to and from the unit. Table 4 shows the air handling capacities by the two methods of rating[5] and also the approximate heating data. In accordance with the A.S.H.V.E. Standard Code for Testing and Rating Steam

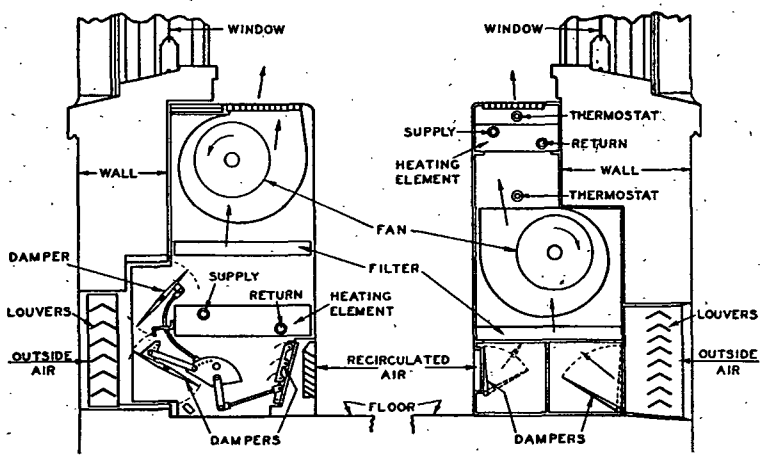
Unit Ventilators the information to be supplied regarding ratings and the basis of rating is the following:

Rating Factors to Be Specified. The rating of the unit ventilator shall specify the following:

a. Final temperature at different entering air temperatures.
b. Total EDR at different entering air temperatures.
c. Air delivered by the unit in cubic feet per minute at the *standard basis of rating* with the fans operated at rated speed, with all air being blown through the heating unit and with the standard louver and grille on the outlet.

The Standard Basis of Rating shall be as follows:

a. Dry saturated steam at a temperature at the unit corresponding to an absolute pressure of 16.7 psi (218.5 F).



DRAW-THROUGH TYPE BLOW-THROUGH TYPE

FIG. 9. TYPICAL UNIT VENTILATORS SHOWING TWO OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

b. Entering air temperature of zero Fahrenheit degrees.
c. Volume delivered in cubic feet per minute converted to standard air at 70 F.

Rating Tables for unit ventilators shall contain the following data in addition to the standard rating, for entering air temperatures from −30 F to +60 F:

a. Inlet temperature, Fahrenheit degrees.
b. Final temperature, Fahrenheit degrees.
c. Total EDR at the specified entering temperature.
d. Surplus or heating EDR at the specified entering temperature.

Surplus or Heating Equivalent Direct Radiation for the purposes of this code shall be construed to mean difference between the total EDR at a specified inlet temperature and the EDR required to heat the air from that temperature to 70 F.

If no direct heating surface (radiation) is installed to take care of the normal heat transfer losses, and the unit ventilator is to be used for both heating and ventilation, then the combined requirements must be taken care of by the unit ventilator.

TABLE 4. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

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 Deg
Anemometer Rating	Condensate Rating			
750	500	214	56	95
1000	750	320	84	95
1260	1000	427	112	95
1560	1250	534	141	95

Heat Required for Ventilating Only

When all of the air handled by the unit is taken from the outside, the total heat to be supplied is obtained by means of Equations 1, 2, 3, and 4.

$$H = 0.24\ W\ (t_y - t) \quad (1)$$
$$H_v = 0.24\ W\ (t - t_o) \quad (2)$$
$$H_t = 0.24\ W\ (t_y - t_o) = H + H_v \quad (3)$$
$$W = d\ 60\ Q \quad (4)$$

From Equations 2, 3, and 4:

$$H_t = H + 0.24\ d\ 60\ Q\ (t - t_o). \quad (5)$$

where

d = density of air, pounds per cubic foot.
H = heat loss of room, 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. $= H + H_v$.
Q = volume of air handled by the ventilating equipment, cubic feet per minute.
t = temperature to be maintained in the room, Fahrenheit degrees.
t_o = outside temperature, Fahrenheit degrees.
t_y = temperature of the air leaving the unit, Fahrenheit degrees.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure.

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 outside at zero, 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. Substituting in Equation 5:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000\ (70 - 0) = 99{,}600 \text{ Btu per hour}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F.}$$

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\ W\ (t_y - t_o) \quad (6)$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to provide exactly for the ventilating requirements, the direct radiation would be selected on the usual basis.

However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_y for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

Heat Required for Ventilating and Recirculating

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24 W_o (t_y - t_o) + 0.24 W_i (t_y - t) \quad (7)$$

$$W_o = d_o 60 Q_o \quad (8)$$

$$W_i = d_i 60 Q_i \quad (9)$$

$$t_y = \frac{H}{0.24 (W_o + W_i)} + t \quad (10)$$

$$H_t = H + 0.24 d_o 60 Q_o (t - t_o) \quad (11)$$

where

W_o = weight of air, pounds per hour taken from out-of-doors.
W_i = weight of air, pounds per hour taken from the room.
d_o = density of air, pounds per cubic foot at temperature t_o.
d_i = density of air, pounds per cubic foot at temperature t.
Q_o = volume of air taken in from the outside, cubic feet per minute.
Q_i = volume of air taken in from the room, cubic feet per minute.

Equations 7, 8, 9, 10 and 11 may be used in the same manner as is illustrated previously for Equations 3, 4 and 5. It may be noted in Equation 11, representing the total heat requirements, that as the quantity Q_o is diminished the heat requirements for the unit diminish very materially.

In *Example 1*, if one third of its air volume is taken from the outside and two thirds from the room, the total heat requirement would be $24{,}000 + \frac{99{,}600 - 24{,}000}{3} = 59{,}200$ Btu per hour. Units designed and operated on this principle show an average heat requirement and, therefore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

Heat Required for Recirculating Only

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24 W (t_y - t) \quad (12)$$

In *Example 1*, if the quantity of air taken in from the outside is reduced to zero, or all of the air handled by the unit is recirculated, the total heat requirement H_t reduced from 99,600 to 24,000 Btu per hour, or to about one-fourth.

Applications of Unit Ventilators

Items to be considered in the application of unit ventilators are: (1) combination with other means of heating, (2) location of units, and (3) method of venting or exhausting.

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at or slightly above room temperature, and enough radiation is installed in the room to take care of the normal heat transfer losses. Where the unit ventilator selected has a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct radiation installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

The *combined* system employs a unit ventilator with sufficient capacity for both ventilation and the normal heat transfer losses. In such a case no direct radiation is required. The necessary operation of the fan when the room is being heated also gives assurance that some ventilation is being provided, especially if automatic dampers are used in the air intake and in the recirculating intake. These dampers are arranged to provide a certain quantity of outside air, depending upon the weather conditions. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating, but if the electric energy fails there will be available only the heating effect of the units acting as convectors.

Location of Unit Ventilator

The location of the unit ventilator in a room is important. Wherever possible it should be placed against an outside wall. It is difficult to obtain proper air distribution if the unit is erected either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom.

Air Exhaust Vents and Flues

The size and location of the air exhaust vent[6] outlet is important. In many cases the sizes for public buildings are regulated by law. See table of state codes and standards in Chapter 51.

In cases where no codes govern, the location and size of vents is left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system. Many states, however, have regulations that will not permit velocities as high as 800 fpm. If a vent opening at or near the floor is near a desk or place where a person is seated, a velocity of 800 fpm in the vent opening will produce an objectionable draft. In such a case the velocity in the vent opening should not exceed 400 to 450 fpm, although duct velocities are maintained at 600 to 800 fpm if codes permit.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

WINDOW VENTILATORS

A window ventilator illustrated in Fig. 10 consists of filter and switch controlled motor driven fans enclosed in a cabinet to be mounted on the window sill. Such units accomplish ventilation, air cleaning, and air circulation, but have no means of heating the air. The direction of air discharge is manually adjustable for seasonal operation.

UNIT HUMIDIFIERS

Unit humidifiers which may be procured in the market at the present time, 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

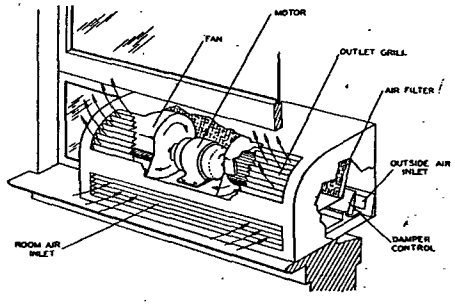


FIG. 10. TYPICAL WINDOW VENTILATOR

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.

To the simple type of nozzle unit may be added the following accessories: a water heater whose function is to increase the vapor pressure, an air heater to heat the air either before or after humidification takes place, a screen or air filter over which the water is sprayed and through which the air is drawn for intimate contact with the water, a fan to create an air stream and to deliver air to the space to be humidified, and a housing to enclose all the elements. This type of unit is procurable in practically all of the variations mentioned. It has the disadvantage of clogging the nozzles and must be serviced continually.

Fig. 11 illustrates a nozzle unit having a humidifying capacity sufficient for a residence or small building. These units usually include air filters and in some cases provide ventilation air by means of an outside air duct connection to the unit. The units are available for either floor or ceiling mounting and are usually placed in a central location in the basement with short supply and return duct connections from the first floor. Room air is brought into the unit through the return duct connection, is passed over a tempering coil heated by steam or hot water, and is then humidified by passing through some type of spray humidifier. Surplus moisture is removed by an eliminator and the humidified air is delivered to the room through a duct connection. Since a large percentage of the tempering

coil capacity is transformed into latent heat during the humidifying process, the unit does not generally eliminate any existing steam radiation but, does tend to improve comfort conditions by supplying heating during the off-period of furnace operation[7].

In the *rotary type* of humidifier the spray is created by rotating vanes or discs which throw the water by centrifugal force and in so doing break it up into a fine mist. 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 liable 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. There are many variations of this type of humidifier.

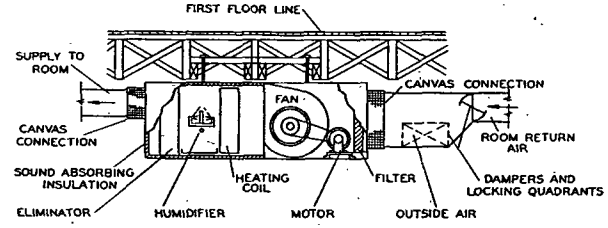


FIG. 11. TYPICAL UNIT HUMIDIFIER OF THE SPRAY TYPE WITH STEAM COIL TO PREHEAT THE AIR FOR RESIDENCES

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.

For a discussion of various methods of humidification see Chapter 37.

REFERENCES

1—See *National Fan Manufacturers Association* standard definitions in Chapter 32.

2—Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165), prepared by a Joint Code Committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Industrial Unit Heater Association* and adopted 1930.

3—Standard Code for Testing Hot Water Unit Heaters prepared by Engineering Committee of *Industrial Unit Heater Association*. Adopted by *Industrial Unit Heater Association* August 1942 and published September 1942.

4—A.S.H.V.E. RESEARCH REPORT No. 958—Temperature Gradient Observations in a Large Heated Space, by G. L. Larson, D. W. Nelson and O. C. Cromer (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 243). A.S.H.V.E. RESEARCH REPORT No. 1011—Tests of Three Heating Systems in an Industrial Type of Building by G. L. Larson, D. W. Nelson and John James (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 185).

5—A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25).

6—A.S.H.V.E. RESEARCH REPORT No. 936—Investigation of Air Outlets in Class Room Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 463). A.S.H.V.E. RESEARCH REPORT No. 1017—Air Supply to Classrooms in Relation to Vent Flue Openings, by F. C. Houghten, Carl Gutberlet and M. F. Lichtenfels (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 279).

7—Estimating the Humidification Requirements of Residences, by W. H. Severns (Papers Presented at the First Annual Conference on Air Conditioning, University of Illinois, *Engineering Experiment Station Circular, No. 26*, October, 1936).

CHAPTER 27

Pipe, Fittings, Welding

Pipe Material, Types of Pipe Used, Dimensions of Pipe Commercially Available, Expansion and Flexibility of Pipe, Pipe Threads and Hangers, Types of Fittings, Welding as Applied to Erection of Piping, Valves, Corrosion of Piping

IMPORTANT considerations in the selection and installation of pipe and fittings for heating, ventilating, and air conditioning work are dealt with in this chapter.

PIPE MATERIALS

Use of corrosion-resistant materials for pipe, including special alloy steels and irons, wrought-iron, copper and brass, has increased considerably during the past few years. The recent development of copper, brass, and bronze fittings which can be assembled by soldering or sweating permits the use of thin-wall pipe and thereby has reduced the initial cost of such installation. The following brief discussion indicates the variety of pipe materials and the types of pipe available.

Wrought-Steel Pipe. Because of its low price, 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.

Seamless steel pipe is frequently used for high pressure work or where pipe is desired for close coiling, cold bending, or other forming operation. Its advantages are its somewhat 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 is claimed to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher first cost is said to be justified on the basis of longer life expectancy. 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 the difference.

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, or 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. Steel pipe bearing a small alloy of copper or other alloying element and iron pipe bearing a small alloy of copper and molybdenum have been claimed to possess more resistance to corrosion than plain

TABLE 1. DIMENSIONS OF SCHEDULES 30 AND 40 AND STANDARD WEIGHT PIPE[a]

Size	Diameter, In.		Thickness, In.	Weight per Ft, Lb		Threads per In.	Circumference, In.		Transverse Area, Sq In.			Length of Pipe, Ft per Sq Ft		Length of Pipe, Ft Containing 1 Cu Ft	Weight of Water, Lb per Ft
	External	Internal		Plain Ends	Threads and Couplings		External	Internal	External	Internal	Metal	External Surface	Internal Surface		
⅛	0.405	0.269	0.068	0.244	0.245	27	1.272	0.845	0.129	0.057	0.072	9.431	14.199	2533.775	0.025
¼	0.540	0.364	0.088	0.424	0.425	18	1.696	1.144	0.229	0.104	0.125	7.073	10.493	1383.789	0.045
⅜	0.675	0.493	0.091	0.567	0.568	18	2.121	1.549	0.358	0.191	0.167	5.658	7.748	754.360	0.083
½	0.840	0.622	0.109	0.850	0.852	14	2.639	1.954	0.554	0.304	0.250	4.547	6.141	473.906	0.132
¾	1.050	0.824	0.113	1.130	1.134	14	3.299	2.589	0.866	0.533	0.333	3.637	4.635	270.034	0.231
1	1.315	1.049	0.133	1.678	1.684	11½	4.131	3.296	1.358	0.864	0.494	2.904	3.641	166.618	0.375
1¼	1.660	1.380	0.140	2.272	2.281	11½	5.215	4.335	2.164	1.495	0.669	2.301	2.768	96.275	0.65
1½	1.900	1.610	0.145	2.717	2.731	11½	5.969	5.058	2.835	2.036	0.799	2.010	2.372	70.733	0.88
2	2.375	2.067	0.154	3.652	3.678	11½	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	1.45
2½	2.875	2.469	0.203	5.793	5.819	8	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077	2.07
3	3.500	3.068	0.216	7.575	7.616	8	10.996	9.638	9.621	7.393	2.228	1.091	1.245	19.479	3.20
3½	4.000	3.548	0.226	9.109	9.202	8	12.566	11.146	12.566	9.886	2.680	0.954	1.076	14.565	4.29
4	4.500	4.026	0.237	10.790	10.889	8	14.137	12.648	15.904	12.730	3.174	0.848	0.948	11.312	5.50
5	5.563	5.047	0.258	14.617	14.810	8	17.477	15.856	24.306	20.006	4.300	0.686	0.756	7.198	8.67
6	6.625	6.065	0.280	18.974	19.185	8	20.813	19.054	34.472	28.891	5.581	0.576	0.629	4.984	12.51
8c	8.625	8.071	0.277	24.696	25.000	8	27.096	25.356	58.426	51.161	7.265	0.443	0.473	2.815	22.18
8	8.625	7.981	0.322	28.554	28.809	8	27.096	25.073	58.426	50.027	8.399	0.443	0.478	2.878	21.70
10c	10.750	10.136	0.307	34.240	35.000	8	33.772	31.843	90.763	80.691	10.072	0.355	0.376	1.785	34.95
10	10.750	10.020	0.365	40.483	41.132	8	33.772	31.479	90.763	78.855	11.908	0.355	0.381	1.826	34.20
12c	12.750	12.090	0.330	43.773	45.000	8	40.055	37.982	127.676	114.800	12.876	0.299	0.315	1.254	49.70
12	12.750	12.000	0.375	49.562	50.706	8	40.055	37.699	127.676	113.097	14.579	0.299	0.318	1.273	49.00

[a]Standard-weight wrought-iron pipe has approximately the same wall thicknesses and weights as contained herein for steel pipe. For exact dimensions, see *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10.

[b]Thicknesses shown in bold face type are identical with thicknesses for Schedule 40 pipe of *A.S.A.* B36.10.

[c]Same as Schedule 30, *A.S.A.* B36.10.

steel pipe and they are advertised and sold under various trade names.

Copper Pipe and Fittings. Owing to its inherent resistance to corrosion, copper and 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 recent introduction of fittings which permit erection by soldering or sweating allows the use of pipe with thinner walls than are possible with threaded connections, thereby reducing the cost of installations.

The initial cost of brass and copper pipe installations generally runs higher than the corresponding job with steel pipe and screwed connections in spite of the use of thin wall pipe, but the corrosive nature of the fluid conveyed or the inaccessibility of some of the piping may warrant use of a more expensive material than plain steel. The advantages of corrosion-

resisting pipe and fittings should be weighed against the correspondingly higher initial cost.

COMMERCIAL PIPE DIMENSIONS

The two weights of steel and wrought-iron pipe commonly used are known as *standard weight* and *extra strong*, which correspond to Schedules 40 and 80 respectively of the *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10. The same external diameter is used for both weights of each nominal size for manufacturing reasons as well

TABLE 2. STANDARD WEIGHTS AND DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE[a]

Size	Outside Diameter, In.	No. of Threads per In.	Standard-Weight Pipe				Extra-Strong Pipe				Double Extra-Strong Pipe[b]	
			Schedule 30		Schedule 40		Schedule 60		Schedule 80			
			Wall Thickness, In.	Weight per Ft, Lb T & C	Wall Thickness, In.	Weight per Ft, Lb T & C	Wall Thickness, In.	Weight per Ft, Lb Plain Ends	Wall Thickness, In.	Weight per Ft, Lb Plain Ends	Wall Thickness In.	Weight per Ft, Lb Plain Ends
⅛	0.405	27	.....	.....	0.068	0.25	.....	.....	0.095	0.31	.....	.....
¼	0.540	18	.....	.....	0.088	0.43	.....	.....	0.119	0.54	.....	.....
⅜	0.675	18	.....	.....	0.091	0.57	.....	.....	0.126	0.74	.....	.....
½	0.840	14	.....	.....	0.109	0.85	.....	.....	0.147	1.09	0.294	1.71
¾	1.050	14	.....	.....	0.113	1.13	.....	.....	0.154	1.47	0.308	2.44
1	1.315	11½	.....	.....	0.133	1.68	.....	.....	0.179	2.17	0.358	3.66
1¼	1.660	11½	.....	.....	0.140	2.28	.....	.....	0.191	3.00	0.382	5.21
1½	1.900	11½	.....	.....	0.145	2.73	.....	.....	0.200	3.63	0.400	6.41
2	2.375	11½	.....	.....	0.154	3.68	.....	.....	0.218	5.02	0.436	9.03
2½	2.875	8	.....	.....	0.203	5.82	.....	.....	0.276	7.66	0.552	13.70
3	3.500	8	.....	.....	0.216	7.62	.....	.....	0.300	10.25	0.600	18.58
3½	4.000	8	.....	.....	0.226	9.20	.....	.....	0.318	12.51	0.636	22.85
4	4.500	8	.....	.....	0.237	10.89	.....	.....	0.337	14.98	0.674	27.54
5	5.563	8	.....	.....	0.258	14.81	.....	.....	0.375	20.78	0.750	38.55
6	6.625	8	.....	.....	0.280	19.19	.....	.....	0.432	28.57	0.864	53.16
8	8.625	8	0.277	25.00	0.322	28.81	.....	.....	0.500	43.39	0.875	72.42
10[c]	10.750	8	0.307	35.00	0.365	41.13	0.500	54.74	.....	.....	.....	.....
12[d]	12.750	8	0.330	45.00	0.375	50.71	0.500[d]	65.41	.....	.....	.....	.....

From Standard Specifications for Welded and Seamless Steel Pipe of the *American Society for Testing Materials, A.S.T.M.* Designation A120.

[a]Sizes larger than those shown in the table are measured by their outside diameter, such as 14 in. outside diameter, etc. These larger sizes will be furnished with plain ends, unless otherwise specified. The weights will correspond to the manufacturers' published standards although it is possible to calculate the theoretical weights for any given size and wall thickness on the basis of 1 cu in. of steel weighing 0.2833 lb.

[b]The American Standard for Wrought-Iron and Wrought-Steel Pipe *A.S.A.* B36.10-1939 has assigned no schedule number to *Double Extra-Strong* pipe.

[c]A 10 in. Standard Weight pipe is also available with 0.279 in. wall thickness, but this wall is not covered by a Schedule Number.

[d]Owing to a departure from the *Standard-Weight* and *Extra-Strong* wall thicknesses for the 12 in. nominal size, Schedules 40 and 60, Table 2 of the *A.S.A.* B36.10-1939, Standard for Wrought-Iron and Wrought-Steel Pipe, the regular *Standard* and *Extra-Strong* wall thicknesses (0.375 in. and 0.500 in.) have been substituted.

as to afford interchangeability in threading, and other elements associated with fabrication and erection. Hence the difference in wall thickness is accompanied by a corresponding change in inside diameter. In sizes up to 14 in., pipe is designated by its nominal size which corresponds roughly to the inside diameter of Schedule 40 pipe. In sizes 14 in. and upward, pipe is designated by its outside diameter (O. D.), and the wall thickness is specified.

While the demands for pipe for the heating and ventilating industry are reasonably well served by Schedule 40 (standard weight) pipe, the erection of pipe by welding sometimes warrants using lighter wall thicknesses. The considerations governing pipe wall thickness and its relation to joint design are covered in the *American Standard Code for Pressure Piping, A.S.A.* B31.1-1942, see Section 122. Standard schedules of pipe thick-

nesses are contained in the *American Standard for Wrought-Iron and Wrought-Steel Pipe, A.S.A.* B36.10, which includes standard-weight and extra-strong thicknesses in Schedules 40 and 80, respectively, and eight other schedules of varying wall thickness to provide for different service conditions. Dimensions and other useful data for Schedules 30 and 40 pipe are given in Table 1. Table 2 from *A.S.T.M.* Specifications A53 and

TABLE 3. STANDARD DIMENSIONS AND WEIGHTS, AND TOLERANCES IN DIAMETER AND WALL THICKNESS FOR COPPER WATER TUBES[a]

(All tolerances in this table are plus and minus except as otherwise indicated)

Standard Water Tube Size, In.	Actual Outside Diameter In.	Average Outside Diameter Tolerance, In.		Wall Thickness, In.						Theoretical Weight, Lb per Ft		
				Type K		Type L		Type M				
		Annealed	Drawn Temper	Nominal	Tolerance	Nominal	Tolerance	Nominal	Tolerance	Type K	Type L	Type M
⅛	0.250	0.002	0.001	0.032	0.003	0.025	0.0025	0.025	0.0025	0.085	0.068	0.068
¼	0.375	0.002	0.001	0.032	0.004	0.030	0.0035	0.025	0.0025	0.134	0.126	0.107
⅜	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.145
½	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.204
⅝	0.750	0.0025	0.001	0.049	0.004	0.042	0.0035	0.030	0.0025	0.418	0.362	0.263
¾	0.875	0.003	0.001	0.065	0.0045	0.045	0.004	0.032	0.003	0.641	0.455	0.328
1	1.125	0.0035	0.0015	0.065	0.0045	0.050	0.004	0.035	0.0035	0.839	0.655	0.465
1¼	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0.884	0.682
1½	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.940
2	2.125	0.005	0.002	0.083	0.007	0.070	0.006	0.058	0.006	2.06	1.75	1.46
2½	2.625	0.005	0.002	0.095	0.007	0.080	0.006	0.065	0.006	2.93	2.48	2.03
3	3.125	0.005	0.002	0.109	0.007	0.090	0.007	0.072	0.006	4.00	3.33	2.68
3½	3.625	0.005	0.002	0.120	0.008	0.100	0.007	0.083	0.007	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.010	0.110	0.009	0.095	0.009	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.010	0.125	0.010	0.109	0.009	9.67	7.61	6.66
6	6.125	0.005	0.002	0.192	0.012	0.140	0.010	0.122	0.010	13.9	10.2	8.92
8	8.125	0.006	+0.002 −0.004	0.271	0.016	0.200	0.014	0.170	0.014	25.9	19.3	16.5
10	10.125	0.008	+0.002 −0.006	0.338	0.018	0.250	0.016	0.212	0.015	40.3	30.1	25.6
12	12.125	0.008	+0.002 −0.006	0.405	0.020	0.280	0.018	0.254	0.016	57.8	40.4	36.7

[a]From Standard Specifications for Copper Water Tube of the *American Society for Testing Materials, A.S.T.M.* Designation B88-41.

NOTE 1:—For copper gas and oil burner tubes, the tolerances shown above for various wall thicknesses (type K) apply irrespective of diameter.

NOTE 2:—For tubes other than round no standard tolerances are established. These tolerances do not apply to condenser and heat exchanger tubes.

A120 combines the schedule thicknesses of *A.S.A.* B36.10 and the old series designations.

Standard-weight pipe is generally furnished with threaded ends in random lengths of 16 to 22 ft, although when ordered with plain ends, 5 per cent may be in lengths of 12 to 16 ft. Five per cent 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 per cent may be in lengths of 6 to 12 ft.

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 tubing 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*. There are three standard wall-thickness schedules of copper water tubing classified in accordance with their principal uses as follows:

Type *K*—Designed for underground services and general plumbing service.
Type *L*—Designed for general plumbing purposes.
Type *M*—Designed for use with soldered fittings only.

In general, Type *K* is used where corrosion conditions are severe, and Types *L* and *M* where such conditions may be considered normal as, for instance, in heating work. Types *K* and *L* are available in both hard and soft tempers; Type *M* is available only in hard temper. 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. New or

TABLE 4. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]

(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 ft from −20 F up				Saturated Steam		Elongation in Inches per 100 ft from −20 F up			
Vacuum Inches of Hg.	Pressure Psig	Temperature Fahrenheit Degrees	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe	Pressure Psig	Temperature Fahrenheit Degrees	Cast-Iron Pipe	Steel Pipe	Wrought-Iron Pipe	Copper Pipe
		−20	0	0	0	0	2.5	220	1.634	1.852	1.936	2.720
		0	0.127	0.145	0.152	0.204	10.3	240	1.780	2.020	2.110	2.960
		20	0.255	0.293	0.306	0.442	20.7	260	1.931	2.183	2.279	3.189
		40	0.390	0.430	0.465	0.655	34.5	280	2.085	2.350	2.465	3.422
29.39		60	0.518	0.593	0.620	0.888	52.3	300	2.233	2.519	2.630	3.665
28.89		80	0.649	0.725	0.780	1.100	74.9	320	2.395	2.690	2.800	3.900
27.99		100	0.787	0.898	0.939	1.338	103.3	340	2.543	2.862	2.988	4.145
26.48		120	0.926	1.055	1.110	1.570	138.3	360	2.700	3.029	3.175	4.380
24.04		140	1.051	1.209	1.265	1.794	180.9	380	2.859	3.211	3.350	4.628
20.27		160	1.200	1.368	1.427	2.008	232.4	400	3.008	3.375	3.521	4.870
14.63		180	1.345	1.528	1.597	2.255	293.7	420	3.182	3.566	3.720	5.118
6.45		200	1.495	1.691	1.778	2.500	366.1	440	3.345	3.740	3.900	5.358

[a]From *Piping Handbook*, by Walker and Crocker. This table gives the expansion from −20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

exposed work generally employs copper pipe of a hard temper. All three classes are extensively used with soldered fittings.

Standard dimensions, weights, and diameter and wall-thickness tolerances for these classes of copper tubing are given in Table 3. Copper pipe is also available with dimensions of steel pipe.

Refrigeration lines used in connection with air conditioning equipment also employ copper tubing extensively. For refrigeration use where tubing absolutely free from scale and dirt is required, bright annealed copper tubing that has been deoxidized is used. This tubing is available in a variety of sizes and wall thicknesses.

EXPANSION AND FLEXIBILITY

The increase in temperature of a pipe from room temperature to an operating steam or water temperature 100 deg or more above room temperature results in an increase in length of the pipe for which provision must be made. The amount of linear expansion (or contraction in the case of refrigeration lines) per unit length of material per degree change in

temperature is termed the *coefficient of linear expansion*, or commonly, the *coefficient of expansion*. This coefficient varies with the material.

The linear expansion of cast-iron, steel, wrought-iron, and copper pipe, the materials most frequently used in heating and ventilating work, can be determined from Table 4.

The three methods by which the elongation due to thermal expansion may be taken care of are: (1) Expansion joints; (2) Swivel joints; (3) Inherent flexibility of the pipe itself utilized through pipe bends, right-angle turns, or offsets in the line.

Expansion joints of the slip-sleeve, diaphragm, or corrugated types made of copper, rubber, or other gasket material are all used for taking up expansion, but generally only for low pressures or where the inherent flexibility of the pipe cannot readily be used as in underground steam or hot water distribution lines.

Swivel joints are used to some extent in low-pressure steam and hot-

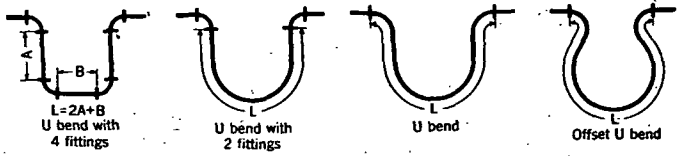


FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

water heating systems, and in hot-water supply lines. Since swivel joints permit the expansive movement of the pipe by turning of threaded joints, which may ultimately result in a leak, it is preferable to provide sufficient flexibility without resorting to swiveling in the threads.

Probably the most economical method of providing for expansion of piping in a long run is to take advantage of the directional changes which must necessarily occur in the piping and proportion the offsets so that sufficient flexibility is secured. Ninety-degree bends with long, straight tangents in either a horizontal or a vertical plane are an excellent means for securing adequate flexibility with larger sizes of pipe. When flexibility cannot be obtained in this manner, it is necessary to make use of some type of expansion bend. The exact calculation of the size of expansion bends required to take up a given amount of thermal expansion is relatively complicated[1]. The following approximate method, however, has been found to give reasonably good results and is deemed to be sufficiently accurate for most heating work.

Fig. 1 shows several types of expansion bends commonly used for taking up thermal expansion. The amount of pipe, L, required in each of these bends may be computed from Equation 1.

$$L = 6.16\sqrt{D\,\Delta} \qquad (1)$$

where

L = length of pipe, feet.
D = outside diameter of the pipe used, inches.
Δ = the amount of expansion to be taken up, inches.

This formula, based on the use of mild-steel pipe with wall thicknesses not heavier than extra-strong, assumes a maximum safe value of fiber

stress of 16,000 psi. When square type bends are used, the width of the bend should not exceed about twice the height, since for a given total length of pipe in the bend, the height of the bend becomes progressively less with increase in width until the height approaches zero and no flexibility exists. Actually, wide bends utilize to best advantage the inherent flexibility of the line, but such bends cannot be proportioned on the basis of Equation 1. For such applications, more accurate methods [1] should be employed. It is further assumed that the corners are made with screwed or flanged elbows or with arcs of circles having radii five to six times the pipe diameter. Use of welding elbows with radii of 1½ times the pipe diameter will decrease the end thrusts somewhat but will raise the fiber stress correspondingly.

All risers must be anchored and safeguarded so that the difference in length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

Proper anchoring of piping is especially necessary with light-weight radiators, to allow for freedom of expansion in order that no pipe strain will distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters, they usually emit disturbing sounds.

HANGERS AND SUPPORTS

Heating system piping requires careful and substantial support. Where changes in temperature of the line are not large, such simple methods of support may be utilized as hanging the line by means of rods or perforated strip from the building structure, or supporting it by brackets or on piers.

When fluids are conveyed at temperatures of 150 F or above, however, hangers or supporting equipment must be fabricated and assembled to permit free expansion or contraction of the piping. This can be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe giving contact only at the bottom, or trolley hangers. In all cases, allowance should be made for rod clearance to permit swinging without setting up severe bending action in the rods.

For pipes of small size, perforated metal strip is often used. For horizontal mains, the rod or strip usually is attached to the joists or steel work of the floor above. For long runs of vertical pipe subject to considerable thermal expansion, either the hangers should be designed to prevent excessive load on the bottom support due to expansion, or the bottom support should be designed to withstand the entire load.

THREADING PRACTICE

In all threaded pipe for heating and ventilating installations the American Standard taper pipe thread, *A.S.A.* B2.1-1942 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 [2] 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 *A.S.A.* 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 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.

TYPES OF FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of

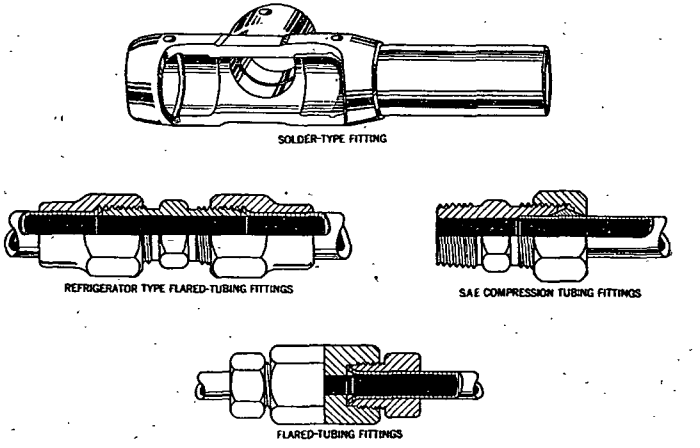


FIG. 2: COPPER OR BRASS TUBING FITTINGS

large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this, malleable-iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure. Malleable iron fittings, like brass fittings, are cast with a round instead of a flat band or bead, or with no bead at all. Fittings are designated as male or female, depending on whether the threads are on the outside or inside, respectively. Screwed galvanized fittings are made according to the 150 lb American Standard.

As in the case of pipe, several weights of fittings are manufactured. Recognized American Standards for the various weights are as follows:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure, *A.S.A.* B16b2, B16a, and B16b respectively.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure, *A.S.A.* B16c.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure, *A.S.A.* B16d.

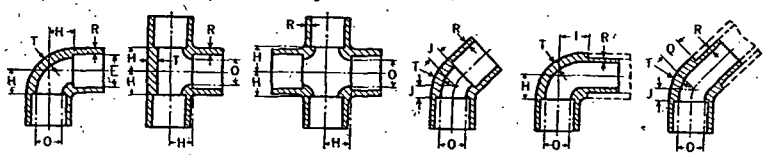
Steel flanged fittings for 150 and 300 lb maximum steam service pressure, *A.S.A.* B16e.

The allowable cold water working pressures for these standards vary from 43 lb for the 25 lb standard to 500 lb for the 300 lb steel standard.

War standard ratings in effect for the duration of the emergency now permit higher ratings for certain sizes of the 125 lb cast-iron flanged fitting standard, and for 300 lb steel flanges and flanged fittings than are shown in the regular American Standards mentioned previously.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings of steel or wrought-iron; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close

TABLE 5. AMERICAN STANDARD DIMENSIONS OF ELBOWS, TEES, CROSSES, AND 45 DEG ELBOWS, SOLDERED-JOINT FITTINGS, *A.S.A.* A40.3-1941



Nominal Size[a]	Cast Brass[b]							Wrought Metal	Bore of Fittings
	Laying Length, Tee, Ell, and Cross	Laying Length, Ell With External Shoulder	Laying Length, 45 Deg Ell	Laying Length, 45 Deg Ell External Shoulder	Inside Diameter of Fittings,[c] Min.	Metal Thickness[d]		Metal Thickness[e] Min.[f]	
	H	I	J	Q	O	T	R	T and R	E, Min.
¼	¼	⅜	3/16	¼	0.31	0.08	0.048	0.030	0.378
⅜	5/16	7/16	3/16	5/16	0.43	0.08	0.048	0.035	0.503
½	7/16	9/16	3/16	5/16	0.54	0.09	0.054	0.040	0.628
¾	9/16	11/16	¼	⅜	0.78	0.10	0.060	0.045	0.878
1	¾	⅞	5/16	7/16	1.02	0.11	0.066	0.050	1.1285
1¼	⅞	1	7/16	9/16	1.26	0.12	0.072	0.055	1.3785
1½	1	1⅛	½	⅝	1.50	0.13	0.078	0.060	1.629
2	1¼	1⅜	9/16	¾	1.98	0.15	0.090	0.070	2.129
2½	1½	1⅝	⅝	⅞	2.46	0.17	0.102	0.080	2.629
3	1¾	1⅞	¾	1	2.94	0.19	0.114	0.090	3.129
3½	2	2⅛	⅞	1⅛	3.42	0.20	0.120	0.100	3.629
4	2¼	2⅜	15/16	1¼	3.90	0.22	0.132	0.110	4.129
5	3⅛	-----	1 1/16	-----	4.87	0.28	0.168	0.125	5.129
6	3⅝	-----	1⅝	-----	5.84	0.34	0.204	0.140	6.129

All dimensions given in inches.

[a]This size is the nominal bore of the tube.

[b]These dimensions may be used for wrought-metal fittings as well as for cast-brass fittings at manufacturer's option.

[c]This dimension is the same as the inside diameter Class L tubing (American Standard Specifications for Copper Water Tube, *A.S.A.* H23.1-1939 (*A.S.T.M.* B88).

[d]Patterns shall be designed to produce body thicknesses given in the table. Metal thickness at no point shall be less than 90 per cent of the thicknesses given in the table.

[e]This dimension has the same thickness as Type L tubing.

[f]These dimensions are minimum, but in every case the thickness of wrought fittings should be at least as heavy as the tubing with which it is to be used.

NOTE 1:—Wrought fittings, as well as cast fittings, must be provided with a shoulder or stop at the bottom end of socket.

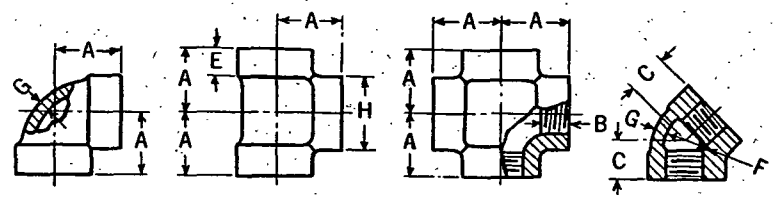
NOTE 2:—Street fittings with male ends are for use in connection with other fittings illustrated.

or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped eccentrically to permit free drain-

age of the water of condensation in steam lines or free escape of air in water lines.

Fittings for copper tubing are available in the soldered, flared, or compression types. Illustrations of each of these types are shown in Fig. 2. Fittings for copper pipe of *IPS* dimensions are available in screwed or soldered types of connection. Table 5 from *A.S.A.* Standard A40.3-1941

TABLE 6. AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45-DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR CLASS 125 CAST-IRON SCREWED FITTINGS, *A.S.A.* B16a-1939



Nominal Pipe Size	A	C	B	E	F		G	H
	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		Metal Thickness,[a] Min.	Outside Diameter of Band, Min.
					Min.	Max.		
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08[b]	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50[b]	5.97	1.88	1.88	12.750	12.850	0.800	15.47

All dimensions given in inches.

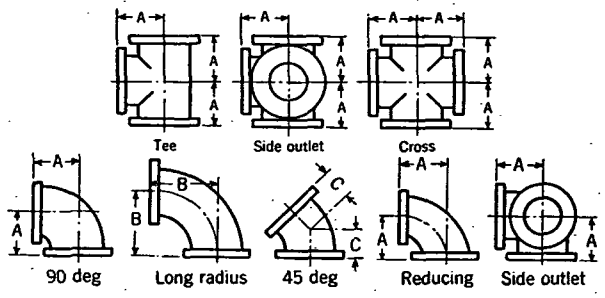
[a]Patterns shall be designed to produce castings of metal thickness given in the table. Metal thickness at no point shall be less than 90 per cent of the thickness given in the table.

[b]Applies to elbows and tees only.

contains dimensions for soldered joint elbows, tees, crosses, and 45 deg elbows.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in both large and small sizes. An American Standard, *A.S.A.* A40.2-1936 has been prepared to standardize dimensions for brass fittings for flared copper water tubes. Flared tube fittings are widely used in refrigerating work where *S.A.E.* dimensions and a 45-deg flare render most fittings interchangeable, although

TABLE 7. AMERICAN STANDARD DIMENSIONS OF TEES, CROSSES[a] (STRAIGHT SIZES), AND ELBOWS FOR CLASS 125 CAST-IRON FLANGED FITTINGS, *A.S.A.* B16a-1939



Nominal Pipe Size[b-c]	A Center to Face Tees, Crosses[c-d] and Elbows	AA Face to Face Tees and Crosses[c-d]	B Center to Face Long Radius Elbows[e-f-g]	C Center to Face 45 Deg Elbows[g]	Diameter of Flange	Thickness of Flange, Min.	Metal[h] Thickness of Body
1	3½	7	5	1¾	4¼	7/16	5/16
1¼	3¾	7½	5½	2	4⅝	½	5/16
1½	4	8	6	2¼	5	9/16	5/16
2	4½	9	6½	2½	6	⅝	5/16
2½	5	10	7	3	7	11/16	5/16
3	5½	11	7¾	3	7½	¾	⅜
3½	6	12	8½	3½	8½	13/16	7/16
4	6½	13	9	4	9	15/16	½
5	7½	15	10¼	4½	10	15/16	½
6	8	16	11½	5	11	1	9/16
8	9	18	14	5½	13½	1⅛	⅝
10	11	22	16½	6½	16	1⅜	¾
12	12	24	19	7½	19	1¼	13/16
14 O.D.	14	28	21½	7½	21	1⅜	⅞
16 O.D.	15	30	24	8	23½	1⅞	1
18 O.D.	16½	33	26½	8½	25	1⅞	1 1/16
20 O.D.	18	36	29	9½	27½	1 11/16	1⅛
24 O.D.	22	44	34	11	32	1⅞	1¼
30 O.D.	25	50	41½	15	38¾	2⅛	1 7/16
36 O.D.	28	56	49	18	46	2⅜	1⅝
42 O.D.	31	62	56½	21	53	2⅝	1 13/16
48 O.D.	34	68	64	24	59½	2¾	2

All dimensions given in inches.

[a] Crosses both straight and reducing sizes 18 in. and larger shall be reinforced to compensate for the inherent weakness in the casting design.

[b] Size of all fittings listed indicates nominal inside diameter of port.

[c] Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

[d] Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

[e] Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

[f] Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45-deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90-deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

[g] Side outlet elbows shall have all openings on intersecting center-lines.

[h] Body thickness at no point shall be less than 87½ per cent of the dimensions given in the table.

for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits. Brass fittings with *S.A.E.* dimensions are not interchangeable with the American Standard fittings for water tubes.

Ammonia pipe fittings made of cast-iron were formerly used extensively in handling refrigerants in large installations. Replacement of ammonia by other refrigerants operating at lower pressures has seriously curtailed the market for these fittings. For this reason formulation of an American Standard for these fittings was abandoned by the *A.S.A.* in 1936.

FLANGE FACINGS AND GASKETS

A number of different flange facings in common use are plain face, raised face, tongue and groove, and male and female. Cast-iron fittings for 125 psi and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a 1/16-in. raised face. The standard facing for steel flanged fittings for 150 and 300 psi is a 1/16-in. raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 6, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Table 7.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain raised face flanges, rubber or rubber inserted gaskets are commonly employed. Asbestos composition gaskets are probably the most widely used, particularly where the temperature exceeds 250 F. Jacketed asbestos and metallic gaskets may be used for any pressure and temperature conditions, but preferably only with a narrow recessed facing.

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 a greatly reduced cost of maintenance and repair, of less weight resulting from the use of a lighter-weight pipe, and of increased economy in pipe insulation, hangers, and supports rather than on the basis of any economy that might be effected in actual erection by welding on low to medium pressure heating jobs.

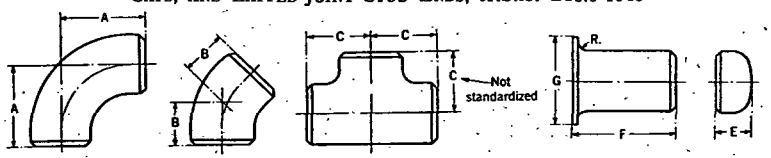
Fusion welding, commonly used in erection of piping, is defined as the process of joining metal parts in the molten, or molten and vapor states, without the application of mechanical pressure or blows. Fusion welding embraces gas welding and electric arc welding, both of which are commonly used to produce acceptable welds. Welding processes and procedure are described in various publications[3].

Welding application requires the same basic knowledge of design as do the other types of assembly, but, in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. 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.

For piping which is to be operated at pressures in excess of 15 psi, rules for fusion welding of pipe joints and the qualification of welders and

TABLE 8. AMERICAN STANDARD DIMENSIONS FOR BUTT-WELDING ELBOWS, TEES, CAPS, AND LAPPED-JOINT STUB ENDS, *A.S.A.* B16.9-1940



Nominal Pipe Size	Outside Diameter	Center-to-End			Caps Eb-c	Lapped-Joint Stub Ends		
		90-Deg Elbows A	45-Deg Elbows B	Of Run Tee C		Length Fb	Radius of Fillet R	Diam. of Lap Gd
1	1.315	1½	⅞	1½	1½	4	1/8	2
1¼	1.660	1⅞	1	1⅞	1½	4	3/16	2½
1½	1.900	2¼	1⅛	2¼	1½	4	¼	2⅞
2	2.375	3	1⅜	2½	1½	6	5/16	3⅝
2½	2.875	3¾	1¾	3	1½	6	5/16	4⅛
3	3.500	4½	2	3⅜	2	6	⅜	5
3½	4.000	5¼	2¼	3¾	2½	6	⅜	5½
4	4.500	6	2½	4⅛	2½	6	7/16	6 3/16
5	5.563	7½	3⅛	4⅞	3	8	7/16	7 5/16
6	6.625	9	3¾	5⅝	3½	8	½	8½
8	8 625	12	5	7	4	8	½	10 5/8
10	10.750	15	6¼	8½	5	10	½	12¾
12	12.750	18	7½	10	6	10	½	15

All dimensions given in inches.

[a]The dimensions of welding tees cover those which have side outlets from one size less than half the size of the run-way opening of the tees to full size.

[b]Dimensions *E* and *F* are applicable only to these fittings in schedules up to and including Schedule 80, *A.S.A.* Standard B36.10-1939.

[c]The shape of these caps shall be ellipsoidal and shall conform to the requirements of the *A.S.M.E.* Boiler Construction Code.

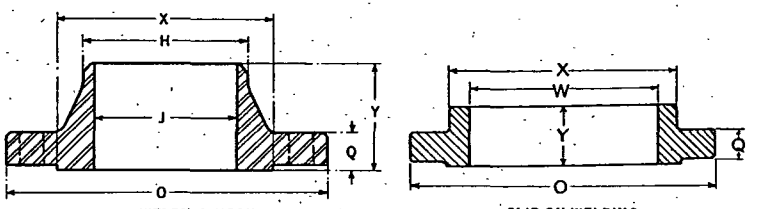
[d]This dimension is for standard machined facings in accordance with American Standard for Steel Pipe Flanges and Flanged Fittings (*A.S.A.* B16e-1939). The back face of the lap shall be machined to conform to the surface of the flange on which it seats. Where ring joint facings are to be applied, use dimension *K* as given in *A.S.A.* B16e-1939.

welding procedures as contained in the *American Standard Code for Pressure Piping* should be observed. U. S. Navy specifications for welding and qualification of operators are available [4]. For pressures 15 psi or below, the foregoing rules or those set forth in the *Standard Manual on Pipe Welding* of the *Heating, Piping and Air Conditioning Contractors National Association* should be followed to insure a satisfactory welded joint.

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* to unify heretofore divergent dimensions for the same type welding fittings as produced by different manufacturers. Standard dimensions for steel butt-welding elbows, tees, caps, and lapped-joint stub ends are given in Table 8. Dimensions for eccentric and concentric reducers, and 180-deg return bends are not shown in Table 8 but

are included in the American Standard. Larger sizes also are available in some types of fittings. The welding bevel which is a straight 37½-deg V for wall thickness ¾ in. and below, and a U-bevel for thicknesses heavier than ¾ in., conforms to the recommended practice of *A.S.A.*

TABLE 9. AMERICAN STANDARD DIMENSIONS OF STEEL WELDING NECK AND SLIP-ON WELDING FLANGES FOR STEAM SERVICE PRESSURE RATING OF 150 PSI (GAGE) AT A TEMPERATURE OF 500 F, AND 100 PSI (GAGE) AT 750 F, *A.S.A.* B16e-1939



Nominal Pipe Size	Diameter of Flange	Thickness of Flg.[a] Min.	Diameter of Hub	Hub Diam. Beginning of Chamfer[b-c]	Length Thru Hub[a]	Inside Diam. of Pipe Schedule 40[c]	Bore of Slip-on Flanges Min.	Diam. of Bolt Circle	No. of Bolts	Size of Bolts
	O	Q	X	H	Y	J	W			
½	3½	7/16	1 3/16	0.84	1⅞	0.62*	0.88	2⅜	4	½
¾	3⅞	½	1½	1.05	2 1/16	0.82*	1.09	2¾	4	½
1	4¼	9/16	1 15/16	1.32	2 3/16	1.05*	1.38	3⅛	4	½
1¼	4⅝	⅝	2 5/16	1.66	2¼	1.38*	1.72	3½	4	½
1½	5	11/16	2 9/16	1.90	2 7/16	1.61*	1.97	3⅞	4	½
2	6	¾	3 1/16	2.38	2½	2.07*	2.44	4¾	4	⅝
2½	7	⅞	3 9/16	2.88	2¾	2.47*	2.94	5½	4	⅝
3	7½	15/16	4¼	3.50	2¾	3.07*	3.56	6	4	⅝
3½	8½	15/16	4 13/16	4.00	2 13/16	3.55*	4.06	7	8	⅝
4	9	15/16	5 5/16	4.50	3	4.03*	4.56	7½	8	⅝
5	10	15/16	6 7/16	5.56	3½	5.05*	5.66	8½	8	¾
6	11	1	7 9/16	6.63	3½	6.07*	6.72	9½	8	¾
8	13½	1⅛	9 11/16	8.63	4	7.98*	8.72	11¾	8	¾
10	16	1 3/16	12	10.75	4	10.02*	10.88	14¼	12	⅞
12	19	1¼	14 3/16	12.75	4½	----------	12.88	17	12	⅞
14 O.D.	21	1⅜	15¾	14.00	5	To Be	14.19	18¾	12	1
16 O.D.	23½	1 7/16	18	16.00	5	Specified	16.19	21¼	16	1
18 O.D.	25	1 9/16	19⅞	18.00	5½	by	18.19	22¾	16	1⅛
20 O.D.	27½	1 11/16	22	20.00	5 11/16	Purchaser	20.19	25	20	1⅛
24 O.D.	32	1⅞	26⅛	24.00	6		24.19	29½	20	1¼

All dimensions given in inches.

[a]A raised face of 1/16 in. is included in *thickness of flange minimum* and in *length through hub.*

[b]The outside surface of the welding end of the hub shall be straight or tapered at not more than 6 deg.

[c]Dimensions *H* and *J* correspond to the outside and inside diameters of pipe as given in *A.S.A.* B36.10-1939, Schedule 40.

*These diameters are identical with the diameters of what was formerly designated as *Standard Weight Pipe* of the corresponding sizes.

Standard B16e-1939, American Standard for Steel Pipe Flanges and Flanged Fittings. The latter also contains dimensions for steel welding neck flanges for pressures up to 2500 psi, and slip-on welding flanges for 150 and 300 psi. Table 9 gives these dimensions for welding-neck and slip-on welding flanges suitable for 150 psi gage pressure.

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

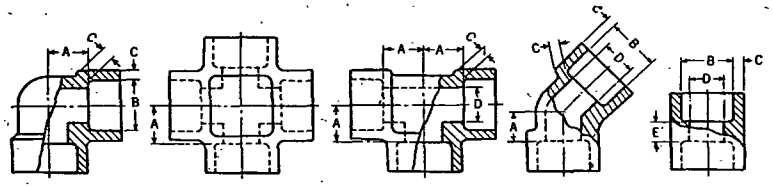
which range commercial fittings are available. This type of fitting has gained rapid acceptance owing to its ease of installation, low cost, and ability to make a pressure tight joint without weakening the pipe as is the case with threading. Dimensions for socket-welding fittings as offered by most manufacturers of the product are given in Table 10.

VALVES

Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger

TABLE 10. PROPOSED AMERICAN STANDARD DIMENSIONS OF SOCKET-WELDING ELBOWS, TEES, CROSSES, 45-DEG ELBOWS, AND COUPLINGS



Nominal Pipe Size	Minimum Depth of Socket	Center to Bottom of Socket: 90-Deg Ells, Tees, Crosses, Sched. 40 & 80	Center to Bottom of Socket: 90-Deg Ells, Tees, Crosses, Sched. 160	Center to Bottom of Socket: 45-Deg Ells, Sched. 40 & 80	Center to Bottom of Socket: 45-Deg Ells, Sched. 160	Couplings Distance Between Bottom Sockets	Bore Diameter of Socket, Minimum	Minimum Socket Wall Thickness: Sched. 40	Minimum Socket Wall Thickness: Sched. 80	Minimum Socket Wall Thickness: Sched. 160	Bore Diameter of Fittings: Sched. 40	Bore Diameter of Fittings: Sched. 80	Bore Diameter of Fittings: Sched. 160
		A		A		E	B	C[a]			D		
¼	3/8	11/16	.....	5/8	.....	½	0.555	0.156	0.156	.....	0.364	0.302	.....
3/8	3/8	11/16	.....	5/8	.....	½	0.690	0.156	0.158	.....	0.493	0.423	.....
½	7/16	7/8	¾	¾	½	5/8	0.855	0.156	0.184	0.234	0.622	0.546	0.466
¾	½	1	1 1/16	¾	⅝	¾	1.065	0.156	0.193	0.273	0.824	0.742	0.614
1	½	1 1/16	1 1/16	15/16	15/16	⅞	1.330	0.166	0.224	0.313	1.049	0.957	0.815
1¼	½	1 5/16	1 5/16	1 5/16	1 5/16	⅞	1.675	0.175	0.239	0.313	1.380	1.278	1.160
1½	½	1 5/8	1 5/8	1 5/8	1 5/8	⅞	1.915	0.181	0.250	0.351	1.610	1.500	1.338
2	½	1 15/16	1 15/16	1	1	1	2.406	0.193	0.273	0.429	2.067	1.939	1.689
2½	½	2 5/16	2 5/16	1 5/16	1 5/16	1 1/16	2.906	0.254	0.345	0.469	2.469	2.323	2.125
3	½	2¼	2½	1¼	1 5/8	1 1/16	3.535	0.270	0.375	0.546	3.068	2.900	2.626

All dimensions are given in inches.
[a]Dimension *C* is 1¼ times the nominal pipe thickness, minimum, but not less than 1/16 in.
Reducing sizes have same center to bottom of socket dimension as the largest size of reducing fitting.

sizes either cast-iron, cast-steel or some of the steel alloys are employed. Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

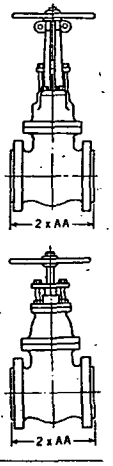
Gate valves are the most frequently used of all valves since in their open position the resistance to flow is a minimum, but they should not be used where it is desired to throttle the flow; globe valves should be used for this purpose. These valves may be secured with either a rising or a non-

rising stem, although in the smaller sizes the rising stem is more commonly used. The rising stem valve is desirable because the positions of the handle and stem indicate whether the valve is open or closed, although space limitations may prevent its use. The globe valve is less expensive to manufacture than the gate valve, but its peculiar construction offers a high resistance to flow and may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

An American Standard, *A.S.A.* B16.10-1939, has been prepared giving the face-to-face dimensions of ferrous flanged and welding-end valves. The following types are covered: wedge gate, double disc gate, globe and angle, and swing check. One purpose of establishing these dimensions is

TABLE 11. AMERICAN STANDARD CONTACT SURFACE TO CONTACT SURFACE DIMENSIONS OF CAST-IRON AND STEEL FLANGED WEDGE GATE VALVES, *A.S.A.* B16.10-1939

Nominal Pipe Size	Contact Surface-to-Contact Surface Dimensions, (2 × AA): Cast-Iron[a] 125	Cast-Iron[a] 175[b,c]	Cast-Iron[a] 250[b]	Steel 150[b]	Steel 300[b]
1	.....	.....	.....	.....	.....
1¼	.....	.....	.....	.....	.....
1½	.....	.....	.....	.....	7½
2	7	7¼	8½	7	8½
2½	7½	8	9½	7½	9½
3	8	9¼	11⅛	8	11⅛
3½	8½	10	11⅞	8½	11⅞
4	9	10½	12	9	12
5	10	11½	15	10	15
6	10½	13	15⅞	10½	15⅞
8	11½	14¼	16½	11½	16½
10	13	16¾	18	13	18
12	14	17½	19¾	14	19¾
14 O.D.	15	.....	22½	15	30
16 O.D.	16	.....	24	16	33
18 O.D.	17	.....	26	17	36
20 O.D.	18	.....	28	18	39
24 O.D.	20	.....	31	20	45



All dimensions given in inches.
[a]These dimensions are the same for Cast-Iron Double Disc Flanged Gate Valves.
[b]These are pressure designations which refer to the primary service ratings in pounds per square inch of the connecting end flanges.
[c]The connecting end flanges of 175 lb valves are the same as those on 250 lb valves.
NOTE 1:—Where dimensions are not given, the sizes either are not made or there is insufficient demand to warrant the expense of unification.
NOTE 2:—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.

to insure that gate valves of a given rating and flange dimension of either the wedge or double disc design will be interchangeable in a pipe line. Contact surface to contact surface dimensions of cast-iron and steel flanged-wedge-gate valves are given in Table 11. End-to-end dimensions for steel butt-welding valves in sizes up to 8 in., inclusive, are the same as those given in Table 11 for steel valves.

Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check in which a dead weight disc moves vertically from its seat.

Valves commonly used for controlling steam or water supply to radiators constitute a special class since they are manufactured to meet

heating system requirements. These valves are generally of the angle type and are usually made of brass. Graduations on the heads or lever handles are often supplied to indicate the relative opening of the valve.

Automatic control of steam supply to individual radiators can be effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are capable of adjustment to permit variation in room temperature to suit individual taste.

Ordinary steam valves may be used for hot water service by drilling a 1/16-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made for use in hot water heating systems are of simpler design, one type consisting of a simple butterfly valve, and another of a quick opening type in which a part in the valve mechanism matches up with an opening in the valve body.

In one-pipe steam-heating systems, automatic air valves are required at the radiators. Two common types of air valves available are the vacuum type and the straight-pressure type. Vacuum valves permit the expulsion of air from the radiators when the steam pressure rises and, in addition, act as checks to prevent the return of air into the radiator when a vacuum is formed by the condensation of steam after the supply pressure has dropped. Ordinary air valves permit the expulsion of air from the radiator when steam is supplied under pressure, but when a vacuum tends to be formed the air is drawn back into the radiator.

CORROSION

Corrosion is sometimes encountered in heating work on the outside of buried pipes or the inside of steam heating systems; it is seldom experienced in hot water heating systems unless the water is frequently renewed. Piping buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford some protection.

Internal corrosion [5] in steam heating systems occurs principally in the condensate return pipes and is nearly always caused by oxygen or carbon dioxide, or both, in solution in the condensate. Oxygen may enter the heating system with the steam, owing to its presence in the boiler-feed water, or it may enter as air through small leaks, particularly in systems which operate at sub-atmospheric pressures. When a steam heating system is operated intermittently, air rushes in during each shutdown period and oxygen is absorbed by the condensate which clings to the interior surfaces of the pipes and radiators. The rate of corrosion depends upon the amounts of oxygen and carbon dioxide present in solution, upon the operating temperature, and upon the length of time that the pipe surfaces are in contact with gas-laden condensate.

Another possible cause of corrosion is a flow of electric current sometimes resulting from faulty electrical circuits which should be corrected. Electrolytic corrosion also may occur because of the presence of two dissimilar metals, such as brass and iron, but the condensate in practically all steam heating systems is such a weak electrolyte that this cause of corrosion is very infrequent.

If trouble is experienced from corrosion, oxygen should be eliminated from the feed water by proper deaeration with commercial apparatus.

The elimination of the oxygen due to air leakage is more difficult because of the multitude of small leaks which exist around valve stems and in pipe joints. In vacuum systems, however, an attempt should be made to minimize such leakage.

Carbon dioxide in varying amounts is contained in steam produced from the majority of water supplies. It is formed from the breaking down of carbonates and bicarbonates which are present in nearly all natural waters. It can be partly removed by chemical treatment and deaeration, but there is no simple method whereby it can be entirely eliminated.

These gases cause corrosion only when in solution in the condensate; when they are mixed with dry steam their corrosive effect is negligible. The amount of gas in solution depends upon the partial pressure of that gas in the atmosphere above the surface of the solution, in accordance with the well known physical law of Henry and Dalton [6]. The correct application of this law, however, requires equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

There is a distinction between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam at relatively high rates, particularly at the times of the cycle when the steam consumption is at its heaviest. In such equipment the gases tend to accumulate in the steam space and to become dissolved in the condensate in high concentrations, thus greatly increasing the possibilities of corrosion. The condensate will more nearly approach in composition the composition of the steam than will the normal condensate from apparatus such as room radiators, and will, therefore, normally include in solution more contaminants. It is possible that careful venting of such equipment would reduce the amount of contaminants dissolved in the condensate, thus giving less corrosion. There is evidence that the partial pressures of the gases and the possibility of corrosion are much lower in heating systems than in high usage equipment. Hence, corrosion observed in the condensate discharge lines from high usage equipment does not necessarily indicate that equally serious corrosion is taking place in the heating system.

The seriousness of corrosive conditions is best determined by actual measurement rather than by inference from isolated instances of pipe failures. The *National District Heating Association* has perfected a corrosion tester for measuring the inherent corrosiveness of existing conditions. This corrosion tester consists of a frame supporting three coils of wire which are carefully weighed. After the tester has been inserted in the pipe line for a definite length of time, the loss of weight of the coils, referred to an established scale, indicates the relative corrosivensss of the condensate. Accompanying such corrosion measurements, a careful chemical analysis should be made of the condensate, and the findings will serve as a basis for an intelligent study of the problem [7].

There are some indications that after a condensate containing carbon dioxide has dissolved some iron and thereby raised its *pH* value, its corrosive action is greatly reduced and the solution will remain comparatively inactive until admission of oxygen permits the precipitation of the dissolved iron as ferric oxide. The *pH* value of the condensate may be buffered to a fairly high value by the solution of iron and not correspond to the *pH* value to be expected in the unbuffered solution containing the same amount of carbon dioxide.

Although inhibitors of various types have had considerable trial and experimentation and successes have been reported, they require further

study. Among these inhibitors are oil, sodium silicate, sodium hydroxide, tannin, and various other organic compounds, some of which release ammonia gas. The possible toxic effects, particularly if the steam is used for contact cooking of food, should not be overlooked.

In view of the fact that corrosion is most frequently found in the return lines from special equipment, which constitute a relatively small part of the total piping in a building, a simple solution of the corrosion problem may be to use non-corroding materials in those certain portions of the piping system, since the higher cost will usually be an unappreciable portion of the total. Brass and copper are undoubtedly less subject to this type of corrosion than the ferrous metals, and considerable attention is now being given to corrosion-resistant linings for ferrous pipe. Cast-iron pipe, sometimes alloyed with other metals, also deserves consideration.

Eighteen ferrous and non-ferrous metals and alloys were tested in a large air conditioning installation [8]. Observations were made in the wash water of the dehumidifier and in the air stream beyond the eliminator plates. The corrosion rates of all metals and alloys utilizing a dichromated-treated wash water were extremely low. Localized attack in the form of pitting was found to occur on steel in crevices or under solid accumulations. Just beyond the dehumidifier eliminator plates corrosive conditions were observed to be particularly severe and in such locations non-ferrous metals and alloys and stainless steels were most resistant. Alloy steels were found to be superior to mild steel.

REFERENCES

1—See (1) Piping Handbook, by Walker and Crocker (McGraw-Hill Co.); (2) A Manual for The Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S. Smith, E. T. Cope, (The Detroit Edison Company).

2—See *API Specification* 5L *for Line Pipe, American Petroleum Institute.*

3—Welding Handbook (*American Welding Society*, 1942). Standard Manual on Pipe Welding, (*Heating, Piping and Air Conditioning Contractors National Association*).

4—General Specifications for Inspection of Material, Appendix VII, Welding (U. S. Navy Department, Jan. 3, 1939).

5—New Light on Heating System Corrosion, by J. H. Walker (*Heating and Ventilating*, May, 1933). A.S.H.V.E. Research Report No. 983—Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 253). A.S.H.V.E. Research Report No. 1037—Corrosion Studies in Steam Heating Systems, by R. R. Seeber, F. A. Rohrman and G. E. Smedberg, (A.S.H.V.E. Transactions, Vol. 42, 1936, p. 263). A.S.H.V.E. Research Report No. 1071—Corrosion Studies in Steam Heating Systems, by R. R. Seeber and Margaret R. Holley (A.S.H.V.E. Transactions, Vol. 43, 1937, p. 461). Corrosion in Steam Heating Systems, by L. F. Collins and E. L. Henderson, (*Heating, Piping and Air Conditioning*, September, 1939 to May, 1940).

6—Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford (A.S.H.V.E. Transactions, Vol. 38, 1932, p. 121).

7—A Method of Measuring Corrosiveness, by J. H. Walker, (*Proceedings, American Society for Testing Materials*, 1940).

8—A.S.H.V.E. Research Report No. 1203—Corrosion Tests in a Water-Recirculating Air Conditioning System, by W. Z. Friend (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 233).

CHAPTER 28

Pipe Insulation

Heat Losses from Bare and Insulated Pipes, Low Temperature Pipe Insulation, Insulation of Pipes to Prevent Freezing, Economical Thickness of Pipe Insulation, Underground Pipe Insulation

THE heat loss from uninsulated pipes may be of considerable magnitude if the temperature of the surrounding medium differs appreciably from that of the fluid conveyed. Losses are increased by rapid motion of the surrounding air or by contact of the pipe with bodies of high conductivity. Careful consideration must, therefore, be given to this factor in a properly designed system and adequate insulation provided, if necessary.

HEAT LOSSES FROM BARE PIPES

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 1. Heat losses from horizontal copper tubes and pipes with tarnished surfaces, are given in Table 2 [1].

Heat losses from bare pipe of materials having lower emissivities may be calculated from data appearing in Chapter 5.

The area in square feet per linear foot of pipe is given in Table 3 for various standard pipe sizes, and Table 4 for copper tubing, while 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.

Very often, when pipes are insulated, flanges and fittings are left bare so as to allow for easy access to the fittings in case of repairs. The fact that a pair of 8-in. standard flanges having an area of 2.41 sq ft would lose, at 100 lb steam pressure, an amount of heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

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 1.

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 lb pressure and is exposed to an average air temperature of 70 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 − 70 = 169.4 F. By interpolation of Table 1 between temperature differences of 157.1 and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4 F is found to be 1.624 Btu per (hour) (linear foot) (Fahrenheit degree). The total annual heat loss from the entire line = 1.624 × 169.4 × 165 (linear feet) × 4000 (hours) = 181,600 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 over-all efficiency of 55 per cent, 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 × 181.6 (thousand Mb) = $146.00.

PIPE INSULATIONS

Pipe insulations are of several general forms and are made of various types of material. The most common form is the rigid sectional covering either split longitudinally into halves or cut through on one side and scored on the other, to facilitate assembling on pipes. Preformed materials are supplied in segments for assembly on large pipes. The sectional coverings are generally supplied with a pasted on canvas

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE STEEL PIPES

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				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	299.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.455	0.495	0.546	0.584	0.612	0.706	0.760
¾	0.555	0.605	0.666	0.715	0.748	0.866	0.933
1	0.684	0.743	0.819	0.877	0.919	1.065	1.147
1¼	0.847	0.919	1.014	1.086	1.138	1.324	1.425
1½	0.958	1.041	1.148	1.230	1.288	1.492	1.633
2	1.180	1.281	1.412	1.512	1.578	1.840	1.987
2½	1.400	1.532	1.683	1.796	1.883	2.190	2.363
3	1.680	1.825	2.010	2.153	2.260	2.630	2.840
3½	1.900	2.064	2.221	2.433	2.552	2.974	3.215
4	2.118	2.302	2.534	2.717	2.850	3.320	3.590
5	2.580	2.804	3.084	3.303	3.470	4.050	4.385
6	3.036	3.294	3.626	3.886	4.074	4.765	5.160
8	3.880	4.215	4.638	4.960	5.210	6.100	6.610
10	4.760	5.180	5.680	6.090	6.410	7.490	8.115
12	5.590	6.070	6.670	7.145	7.500	8.800	9.530

TABLE 2. HEAT LOSS FROM HORIZONTAL TARNISHED COPPER PIPE

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 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
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

TABLE 3. EXTERNAL SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	10	2.817
1½	0.498	4	1.178	12	3.338

jacket. Blanket insulations are sometimes used for wrapping large pipes, particularly where removal for frequent servicing of the pipe is necessary. Fittings and bends are commonly covered with portions of standard preformed insulation or, when irregular in contour, with plastic materials known as insulating cements.

Insulation is secured to pipes with staples which are used to bridge the joint between half sections, and with metal pipe covering bands or rings of wire which secure individual sections and effect a junction between abutting sections. A number of surface finishes are used over pipe insulation depending upon the service encountered and appearance desired. Canvas jackets are most common although asbestos paper or asbestos finishing cements are sometimes employed. Insulation outdoors should be waterproof and is generally protected with an asphalt felt for piping and asphaltic cements for fittings. Insulation on lines carrying cold water, brine, or other cold fluids is carefully finished to obtain adequate sealing against the penetration of water vapor.

The selection of pipe insulation for a particular service condition must be made with full consideration of a number of properties in addition to thermal conductivity. Factors which may be of more importance than the thermal conductivity are: ease of application, fire resistance, heat stability, weathering stability, resistance to damage by physical abuse, and others which may apply to a particular installation. A complete evaluation of pipe insulation cannot be included here. Insulation manufacturers should be consulted in regard to the selection of insulation which is to meet specific requirements.

HEAT LOSSES FROM INSULATED PIPES

The conductivities of various materials used for insulating steam and hot water systems are given in Table 6. They are given as functions of the mean temperatures or the arithmetic mean of the inner and outer surface temperatures of the insulations. It should be emphasized that they are the average values obtained from a number of tests made on

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)
½	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	...	.....

TABLE 5. AREAS 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.

each type of material, also, that in the use of *conductivity* all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

The heat losses through 1, 1½, and 2-in. thick 85 per cent magnesia type of insulation for temperature differences between the pipe and the surrounding atmosphere up to 280 F are shown in Figs. 1, 2, and 3. Standard thicknesses of 85 per cent magnesia pipe covering are not exactly 1 in. However, the loss through any given thickness of insulation can be obtained by interpolation. Also, the losses through any of the insulations given in Table 6 can be obtained by multiplying the losses obtained from Figs. 1, 2, or 3 by the factors given in Table 7.

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

TABLE 6. THERMAL CONDUCTIVITY (*k*) OF VARIOUS TYPE PIPE INSULATIONS FOR MEDIUM AND HIGH TEMPERATURE PIPE[a]

Expressed in Btu per (hour) (square foot) (Fahrenheit degree temperature difference per inch)

TYPES OF INSULATING MATERIALS	DENSITY LB/CU FT	TEMP. RANGE OF ACCEPTED USE	MEAN TEMPERATURE, F DEG				
			100	200	300	400	500
85% Magnesia—Type	13–15	Up to 600 F	0.41	0.45	0.48	0.52	....
Corrugated Asbestos—Type							
4 Ply per 1 in.	11–13	Up to 300 F	0.57	0.68	0.80	....	....
6 Ply per 1 in.	15–17	Up to 300 F	0.51	0.59	0.69	....	....
8 Ply per 1 in.	18–20	Up to 300 F	0.49	0.57	0.65	....	....
Laminated Asbestos—Type (35–40 laminations per 1 in.)	30–35	Up to 700 F	0.39	0.44	0.49	0.54	....
Mineral Wool—Type	10–15	Up to 800 F	0.40	0.45	0.50	0.55	....
Diatomaceous Silica—Type	25–30	Up to 1900 F	0.63	0.66	0.69	0.72	0.75
Brown Asbestos Fiber—Type	13–15	Up to 1200 F	0.34	0.39	0.44	0.49	0.54

[a]Average values from various laboratories for insulating materials of various manufacturers.

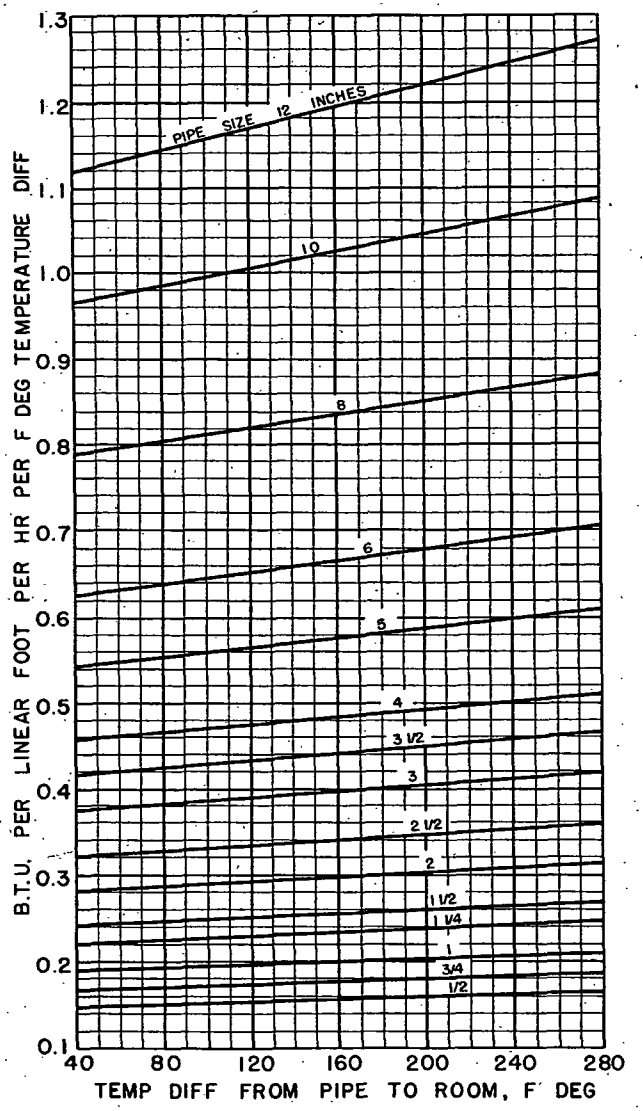


FIG. 1. HEAT LOSS THROUGH 1 IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

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

three or four 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

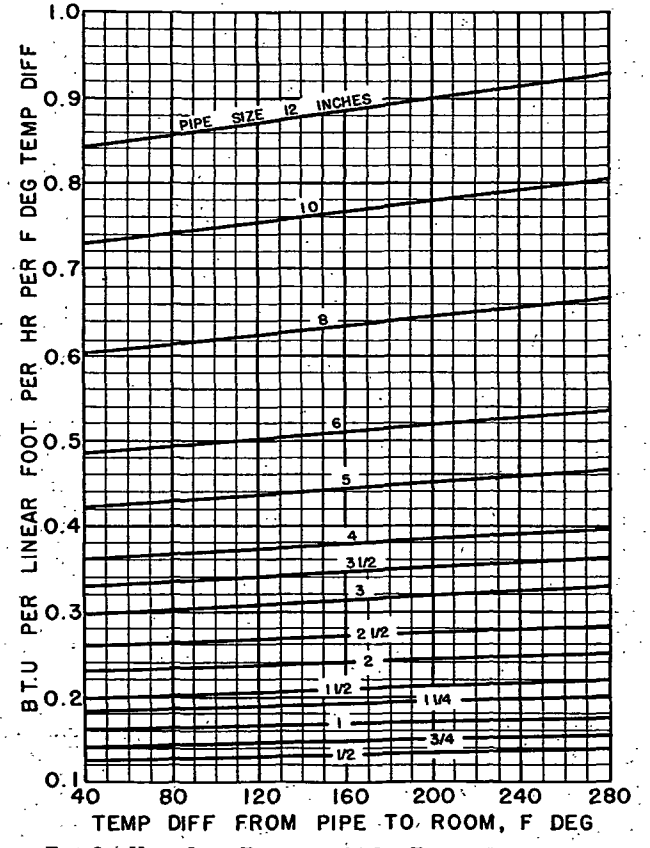


FIG. 2. HEAT LOSS THROUGH 1½ IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

transferred per square foot of outer surface of the insulation is given by the equation:

$$q_o = \frac{k\,(t_1 - t_2)}{r_2 \log_e \frac{r_2}{r_1}} \qquad (1)$$

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 degrees.
t_2 = temperature of outer surface of insulation, Fahrenheit degrees.

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. 4.

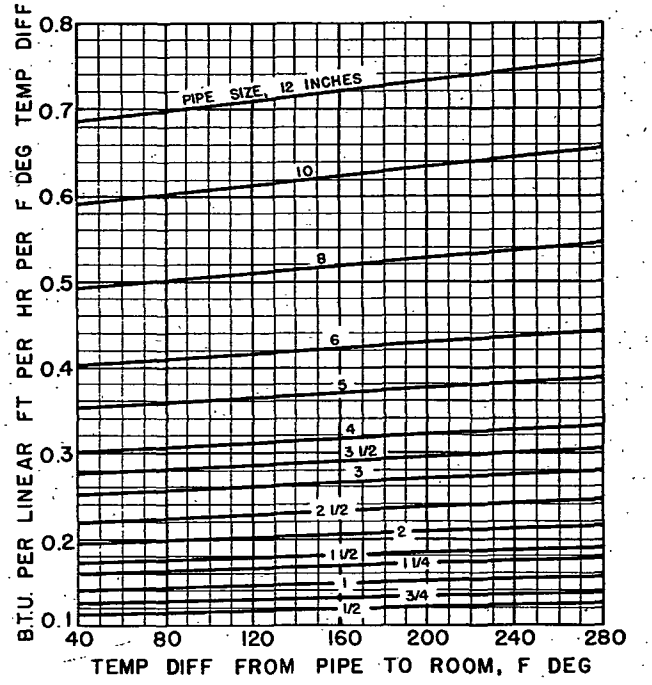


FIG. 3. HEAT LOSS THROUGH 2 IN. THICK 85 PER CENT MAGNESIA TYPE COVERING

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).

The heat loss through two or more thicknesses of insulation applied to a pipe can be calculated by means of the equation:

$$q_o = \frac{t_1 - t_2}{\frac{r_2 \log_e \frac{r_2}{r_1}}{k_1} + \frac{r_3 \log_e \frac{r_3}{r_2}}{k_2} + \cdots} \qquad (2)$$

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 diatomaceous silica, and a 2-in. thickness of 85 per cent 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 silica and 85 per cent 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 formula. First assume an outer surface temperature of 140 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 silica and (570 + 140) ÷ 2 or 355 F for the 85 per cent magnesia insulation. The

TABLE 7. PIPE COVERING FACTORS

TYPES OF INSULATING MATERIALS	TEMPERATURE DIFFERENCE, PIPE TO AIR, F DEG				
	100	200	300	400	500
Corrugated Asbestos—Type					
4 Ply per 1 in.	1.30	1.36	1.42	.....	.....
6 Ply per 1 in.	1.19	1.23	1.27	.....	.....
8 Ply per 1 in.	1.15	1.19	1.23	.....	.....
Laminated Asbestos—Type	0.96	0.98	1.00	1.02	1.04
Mineral Wool—Type	0.98	1.00	1.02	1.05	1.07
Diatomaceous Silica—Type	1.37	1.36	1.35	1.35	1.34
Brown Asbestos Fiber—Type	0.86	0.88	0.91	0.93	0.96

conductivities of these two materials at mean temperatures of 885 and 355 F interpolated from Table 8 are 0.865 and 0.5 Btu respectively.

These values are substituted in Equation 2 and a trial calculation made. For a nominal 6-in. steel pipe $r_1 = 3.312$, $r_2 = 6.312$ and $r_3 = 8.312$ then,

$$q_o = \frac{1200 - 140}{\frac{8.312 \log_e \frac{6.312}{3.312}}{0.865} + \frac{8.312 \log_e \frac{8.312}{6.312}}{0.5}} = \frac{1060}{6.2 + 4.58} = 98.3 \text{ Btu.}$$

The temperature drop from the outer surface of the insulation to the surrounding air for a heat loss of 98.3 Btu is found from Fig. 4 to be 57 F for a 16-in. O.D. cylindrical surface, or 57 + 80 F room temperature = 137 F surface temperature. Since a surface temperature of 140 F was assumed, it is evident that a temperature closer to 137 F, or, for instance, 138 F should be used for recalculation:

$$q_o = \frac{1200 - 138}{6.2 + 4.58} = 98.4 \text{ Btu.}$$

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 silica is 98.4 × 6.2 = 610 F or the temperature between the two insulating materials is (1200 − 610) = 590 F. Since a temperature of 570 F between the two materials was assumed, it is obvious that a temperature closer to 590, or for instance 586 F may be selected. The mean temperatures of the two insulations corresponding to the new assumptions are (1200 + 586) ÷ 2 = 893 and (586 + 138) ÷ 2 = 362 and the interpolated conductivities corresponding to the new mean temperatures are 0.87 and 0.505

for the diatomaceous silica and 85 per cent magnesia respectively. By substituting in Equation 2:

$$q_o = \frac{1200 - 138}{\frac{5.36}{0.87} + \frac{2.29}{0.505}} = \frac{1062}{6.16 + 4.53} = 99.3 \text{ Btu}$$

Again referring to Fig. 4, it is seen that the temperature drop from the outer surface of the insulation to the surrounding air for a heat loss of 99.3 Btu = 38 F which corresponds to the surface temperature of 138 F last assumed. The temperature drop through the diatomaceous silica is 99.3 × 6.16 = 612 F, corresponding to a temperature of 588 F between the two materials which checks very closely with the temperature of

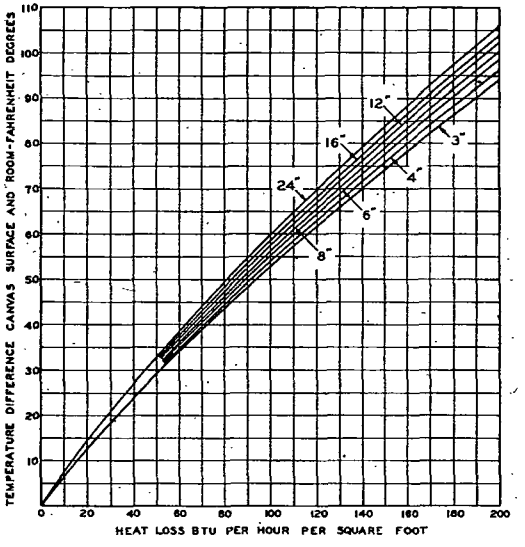


FIG. 4. HEAT LOSS FROM CANVAS-COVERED CYLINDRICAL SURFACES OF VARIOUS DIAMETERS

585 F last assumed. The heat loss is therefore 99.3 × 8.312 ÷ 3.312 or 249 Btu per square foot of pipe surface. Since the surface area per linear foot of 6-in. pipe is 1.734 sq ft (Table 5), the heat loss per linear foot of pipe will be 249 × 1.734 = 432 Btu per hour.

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 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 per cent in the case of 1-in. thick insulation, to about 5 per cent 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.

Fig. 4 shows the loss of heat from canvas-covered, cylindrical surfaces of various outside diameters when the surface to air temperature difference is low. The data are from tests made at Mellon Institute.

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

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			Brine Thickness			Heavy Brine Thickness		
	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface	Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Sq Ft Pipe Surface
½	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087	0.394
¾	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

result in heat transfer much greater than that shown in Tables 1 and 2 for transfer from bare pipe to air.

The saving due to use of insulation on piping is illustrated in *Example 4*.

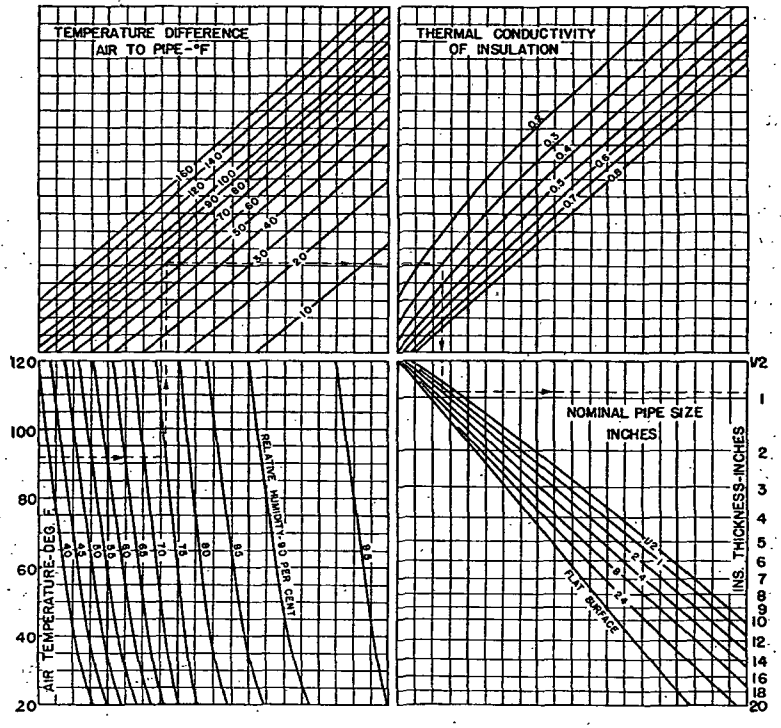
Example 4. If the steam line given in *Examples 1* and *2* is covered with 1 in. thick 85 per cent magnesia, determine the resulting total annual loss through the insulation. Also compute the monetary value of the annual saving and the percentage of saving over the heat loss from the bare pipe.

Solution. By referring to Fig. 1, the coefficient for 1 in. magnesia on a 2-in. pipe is found to be 0.300 Btu per (hour) (linear foot of pipe) (degree temperature difference) at a temperature difference of 169.4 F. The total hourly loss per linear foot of pipe will then be 0.300 × 169.4 = 50.8 Btu. The total annual loss through the insulation = 50.8 × 165 (linear feet) × 4000 (hours) = 33,500 Mb. The annual bare pipe loss as determined in the solution of *Example 1* was found to be 181,600 Mb. The saving due to insulation is then 181,600 − 33,500 = 148,100 Mb per year.

From the solution of *Example 2*, it was found that the heat supplied to the system cost $0.804 per thousand Mb. Therefore, the monetary value of the saving = 0.804 (dollars) × 148.1 (thousand Mb) = $119.07, or 81.5 per cent of the cost when using uninsulated pipe.

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



[a]Solve problems as indicated by dotted line, entering chart at lower left-hand scale.

FIG. 5. THICKNESS OF PIPE INSULATION TO PREVENT CONDENSATION ON OUTER SURFACE[a]

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 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 applicators and

users. Unless time proven methods are known, specifications of insulation manufacturers should be obtained and followed carefully.

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 approximate required thickness of insulation to prevent condensation on pipes and flat metallic surfaces may be obtained from Fig. 5 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.

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			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	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

Heat gains for pipes insulated with a material having an installed conductivity of 0.30 Btu per (square foot) (hour) (Fahrenheit degree per inch) 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 heat losses which will result and 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, and 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 of the damage and service interruptions which 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 F above, and the surrounding air temperature 50 F 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 which 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 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity required will be one-half that shown in the last column 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 neglected. 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.

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 line or to place an electric resistance heater along 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.

ECONOMICAL THICKNESS OF PIPE INSULATION

The thicknesses of insulation which ordinarily are used for various temperature conditions are given in Table 10. Where a thorough analysis of economic thickness is desired this may be accomplished through the use of the chart, Fig. 6.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly.

UNDERGROUND PIPE INSULATION

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes (See Chapter 29). Detailed data

TABLE 10. THICKNESSES OF PIPE INSULATION ORDINARILY USED INDOORS[a]

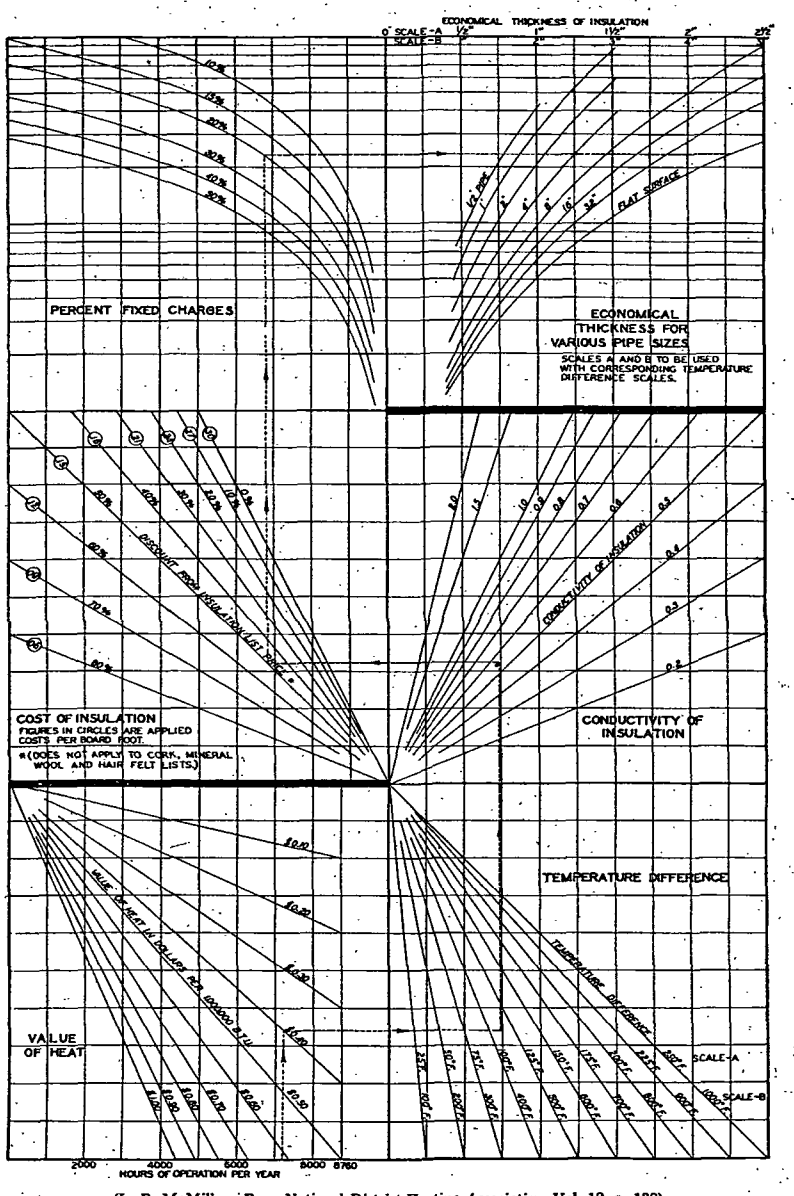
Steam Pressure Psig or Condition	Steam Temperature Fahrenheit Degrees	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 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.

on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[2].

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 firmly secure the covering with corrosion resistant wire, 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.

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[3] previously developed, together with



(L. B. McMillan, *Proc. National District Heating Association*, Vol. 18, p. 138).

FIG. 6. CHART FOR DETERMINING ECONOMICAL THICKNESS OF PIPE INSULATION

TABLE 11. 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	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

other experimental data which have been presented, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 11 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 lines 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 line estimates.

REFERENCES

1—Heat Loss from Copper Piping, by R. H. Heilman (*Heating, Piping and Air Conditioning*, September, 1933, p. 458).

2—Handbook of the *National District Heating Association*, Second Edition, 1932.

3—Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 335).

CHAPTER 29

District Heating

Steam Distribution Piping, Selection of Pipe Sizes, Conduits for Piping, Pipe Tunnels, Overhead Distribution, Inside Piping, Fluid Meters and Metering, Steam Requirements, Rates, Utilization, Automatic Temperature Control

THOSE phases of district heating which frequently fall within the province of the heating engineer are outlined here with data and information for solving incidental problems in connection with institutions and factories. Some data are included to cover the piping peculiar to heating systems which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry.

STEAM DISTRIBUTION PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the 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 reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per (hour) (square foot of equivalent heating surface) is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Miscellaneous steam requirements such as laundry, cooking, or process should be individually calculated.

The steam requirements for water heating should be taken into account, but in most types of buildings this load will be relatively small compared with the heating load and will seldom occur at the time of the heating peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend upon (1) boiler pressure, (2) whether exhaust or live steam, (3) pressure requirements of apparatus to be served. If steam has been passed through electrical generating units, the pressure will be considerably lower than if live steam, direct from the boilers, is used.

The advantages of low pressure distribution (2 to 30 psi) are (1) smaller heat loss per square foot of pipe surface, (2) less trouble with traps and valves, (3) simpler problems in pressure reduction at the buildings, and

(4) general reduction in maintenance costs. With distribution pressures not exceeding 40 psi there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than 50 psi a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators.

The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than

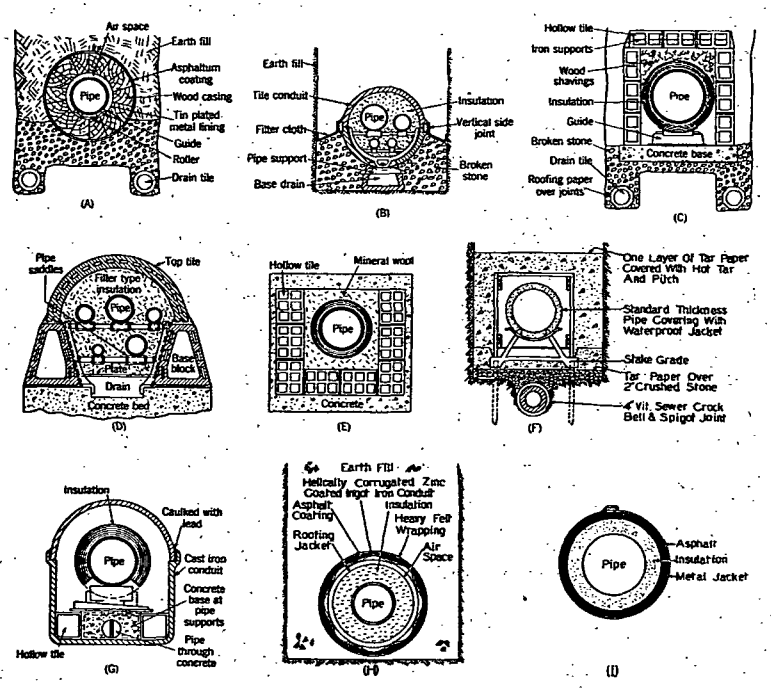


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

building heating, (3) wider flexibility as to allowance for maximum pressure drop.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 psi, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 psi is usually demanded although 30 psi is sufficient if the flat work ironer is equipped with a large number of rolls and if a slower rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 psi. Where pressures are not as high as desired higher pressures can be obtained by a steam compressor.

PIPE SIZES

The lengths of pipe, steam quantities, and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of Babcock's pressure drop formula:

$$P = 0.0000000367 \left(1 + \frac{3.6}{D}\right) \frac{W^2 L}{dD^5}$$

$$W = 5220 \sqrt{\frac{PdD^5}{\left(1 + \frac{3.6}{D}\right) L}}$$

where

P = loss in pressure in pounds.
D = inside diameter of pipe in inches.
L = length of pipe in feet.
d = weight of 1 cu ft of steam.
W = pounds of steam per hour.

Numerical values of the various factors are given in Table 1, Chapter 23.

CONDUITS FOR PIPING

Conduits for steam pipes buried underground should be reasonably waterproof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the insulation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural or cast steel set in concrete. In general, cast steel is preferable to structural steel.

In laying out underground conduits the following points should be borne in mind:

1. The depth of the buried conduit should be kept at a minimum. Excavation costs are a large factor in the total cost.

2. An expansion joint, offset, or bend should be placed between each two anchors. Advantage should be taken of the flexibility of piping to absorb expansion wherever possible. Information on provisions for expansion will be found in Chapter 27.

3. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic test pressure should be one and one-half times the maximum service pressure and it should be held for a period of at least two hours without evidence of leakage.

There are many types of conduits, some of which are manufactured products and some of which are built in the field. Some of the more common forms are illustrated in Fig. 1.

The conduit (A) is of a wood casing construction which has been widely used in the past. The wood casing is segmented, lined with tin, and bound with wire. The outside of the conduit is coated with asphaltum. It is not suitable for high temperatures or poorly drained soils.

In Fig. 1 (B), (C), (D), (H) and (I) are patented forms of conduits. The insulation is sometimes a loose filler packed into the conduit. Conduits (H) and (I) are prefabricated. Both of these conduits are enclosed in metal jackets.

At (C) and (E) are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled par-

tially or wholly with a loose insulating material. The addition of this loose insulating material to the sectional insulation is, of course, optional and is justified only where high pressure steam is used.

(E) and (F) are conduits used by two district heating companies, and have the advantage of being constructed of common materials.

Conduit (G) is of cast-iron construction, assembled with lead joints and is water-tight, if properly laid. It is obviously expensive and is justified only in exceptional cases.

Since it is difficult to make a concrete or masonry conduit absolutely water-tight, provision should be made for some seepage. The pipe should

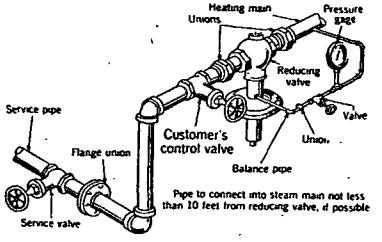


FIG. 2. CONNECTIONS FOR REDUCING VALVE WITHOUT BYPASS

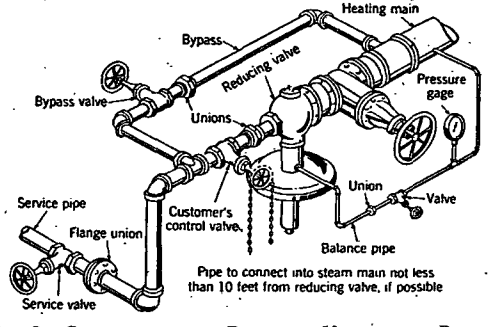


FIG. 3. CONNECTIONS FOR REDUCING VALVE WITH BYPASS

be protected by a waterproof jacket over the insulation and the seepage drained from the inside of the conduit.

Underdrainage of the conduit is generally provided for by a tile drain laid in crushed stone or gravel underneath the conduit. The tile underdrain should be carried to the sewer or some other drainage point.

Manholes are required at intervals for access to valves, traps, and some types of expansion joints.

Where steam and return piping are installed in the same conduit, the return piping usually follows the same grade as the steam piping. In general, the condensation is pumped back under pressure.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or

roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt-saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

Because of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes omitted along heating lines unless they are required to accommodate miscellaneous other services or provide underground passage between buildings.

OVERHEAD DISTRIBUTION

In some industrial and institutional applications, the distribution piping may be installed, entirely or in part, above ground. This method

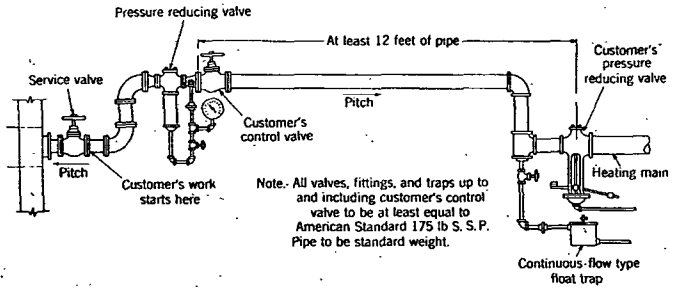


FIG. 4. STEAM SUPPLY CONNECTION WHEN USING TWO REDUCING VALVES

of construction has the advantage of requiring no excavation and being easily maintained.

INSIDE PIPING

Figs. 2 and 3 show typical service connections used for low pressure steam service.

Fig. 2 shows installation of a reducing valve without a bypass, which is usually omitted in the case of smaller size valves.

Fig. 3 illustrates the use of a reducing valve, with a bypass which is generally provided for larger installations. This latter construction permits the operation of the line in case of failure in the reducing valve. In the smaller sizes, the reducing valve can be removed, a filler installed, and the house valve used to throttle the flow of steam until repairs are made.

Fig. 4 shows a typical installation used for high pressure steam service[1]. The first reducing valve effects the initial pressure reduction. The second reducing valve reduces the steam pressure to that required.

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer. There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very cold weather without endangering plumbing. It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or restaurants on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the

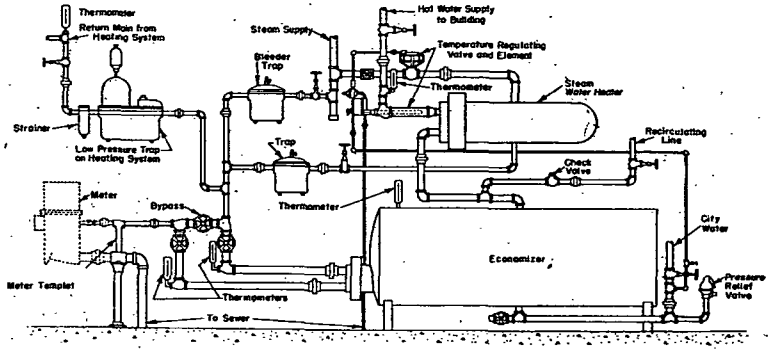


FIG. 5. METHOD OF INSTALLING A WATER HEATER AND ECONOMIZER IN A GRAVITY HEATING SYSTEM

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.

2. Residual heat in the condensate should be salvaged.

This heat may be salvaged by means of a cooling coil, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building.

The condensation from the heating system, after leaving the trap, passes through the economizer. The supply to the hot water 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 connection should be tied in *between* the economizer and the water heater proper, not at the economizer inlet, because the recirculated hot water is itself at a high temperature.

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensate if storage capacity is provided for the preheated water. Frequently a type of economizer is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

The maximum in economical operation and satisfactory heating can only be obtained by the use of some automatic temperature control system.

The perfection of fluid meters has contributed as much to the advance-

ment of district heating as any other one thing. Meters are classified into two groups: *Quantity Meters* and *Rate of Flow Meters*.

Quantity Meters

The one type of quantity meter used is the condensation meter, which may be of the *tilting bucket* or *revolving drum* type.

The condensation meter is a popular type for use on small and medium sized installations, where all the condensate 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 low pressure distribution has made it standard equipment with many heating companies.

Condensation meters should not be operated under pressure; they are made for either gravity or vacuum installations. Where bucket traps are used, a vented receiver is essential ahead of the meter. Where continuous flow traps are used, a vented receiver is not necessary, but is

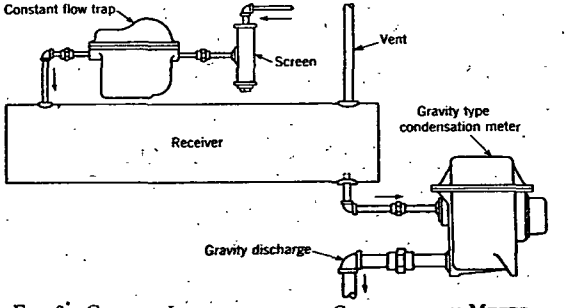


FIG. 6. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

desirable. Fig. 6 illustrates a gravity condensation meter installation using a vented receiver.

Rate of Flow or Flow Meters

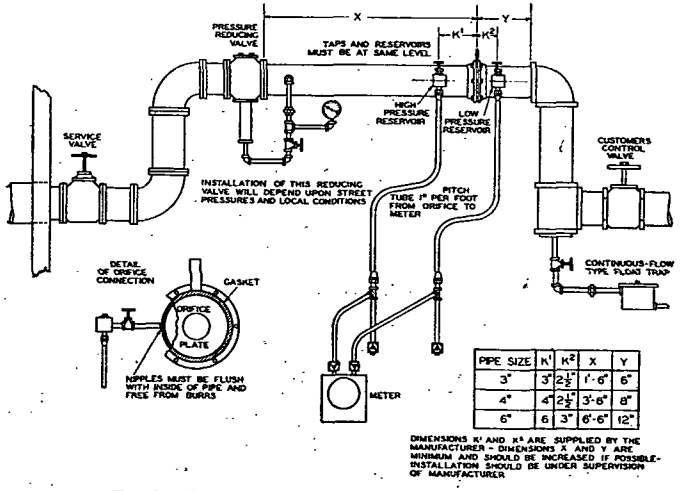
Flow meters used for district heating work are of three types: *Area Meters*, *Head Meters* and *Velocity Meters*.

Area meters are those, in the operation of which, a variation in the cross-section of stream under constant head is used as an indication of the rate of flow. A tapered plug is suspended in an orifice and moves axially with the flow, which is vertically upward. The weight of the plug provides a definite pressure differential and the plug floats at such a height as will provide enough orifice area to pass the flow at the pressure difference. The position of the plug is transmitted by means of a lever and pencil and records the flow on a graduated strip chart.

Head meters are those in which the stream of fluid creates a difference of pressure, or differential head. This head is created by an orifice, Venturi tube, flow nozzle, or Pitot tube and will depend upon the velocity and density of the fluid.

The secondary element must contain a differential pressure gage, which will translate the pressure difference into rate of flow or total flow. This mechanism may be either mechanical or electrical. The electric flow meter has the advantage of being able to locate the instruments at some distance from the primary element.

Fig. 7 is a typical example of an orifice-type meter installation. A few general points to be considered in installing a meter of this type are: (1) It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available. (2) Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company. (3) The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above. (4) Meter piping should be kept free from leaks. (5) Sludge should not be permitted to collect in the meter body. (6) The meter body and meter piping should be kept above freezing temperatures. (7) It is best not to connect a meter body to more than one service. (8) Special instructions are furnished for metering a turbulent or pulsating flow.



PIPE SIZE	K¹	K²	X	Y
3"	3"	2½"	1'-6"	6"
4"	4"	2½"	3'-6"	8"
6"	6	3"	6'-6"	12"

FIG. 7. ORIFICE METER STEAM SUPPLY CONNECTION

Velocity meters are those in which the primary element is some device that is kept in continual rotation by the linear motion of the stream. The secondary element is, essentially, a revolution counter. The primary and secondary elements are combined into one unit.

For steam metering, the shunt meter is an example of the velocity type. This unit is connected directly in 2, 3 and 4 in. pipe lines. Larger size mains are metered by installing a 2 in. meter in a bypass with a restricting orifice in the main line.

Selection of Meter

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations: (1) Its use in a new or an old installation. (2) Method to be used in charging for the service. (3) Location of the meter. (4) Large or small quantity to be measured. (5) Temporary or permanent installation. (6) Cleanliness of the fluid to be measured. (7) Temperature of the fluid to be measured. (8) Accuracy expected. (9) Nature of flow: turbulent, pulsating, or steady.

(10) Cost. *a.* purchase price, *b.* installation cost, *c.* calibration cost, *d.* maintenance cost. (11) Servicing facilities of the manufacturer. (12) Pressure at which fluid is to be metered. (13) Type of record desired as to indicating, recording or totalizing. (14) Stocking of repair parts. (15) Use of open jets where steam is to be metered. (16) Metering to be done by one meter or by a combination of meters. (17) Use as a check meter. (18) Its facilities for determining or recording information other than flow.

STEAM REQUIREMENTS

Methods of estimating steam requirements for heating various types of buildings are given in Chapter 20.

Table 7 in Chapter 20 represents information obtained from all sections of the United States, and the group of buildings from which the information was taken represents a cross-section of all types of heating systems.

Steam requirements for water heating can be satisfactorily estimated by using a consumption of 0.0025 lb per (day) (cubic foot of heated space) for office buildings, without restaurants, and 0.0065 lb per (day) (cubic foot of heated space) for apartment buildings.

Complete information on water heating requirements is given in Chapter 50.

Additional data on steam requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association.*

RATES

Fundamentally, district heating rates are based upon the same principles as those recognized in the electric light and power industry, the main object being a reasonable return on the investment. However, there are other requirements to be met; the rate for each class of service should be based upon the cost to the utility company of the service supplied and upon the value of the service to the consumer, and it must be between these two limits. District heating rates should be designed to produce a sufficient return on the investment regardless of weather conditions, although existing rate schedules do not conform to this principle. Lastly, the rate schedule must be reasonably simple and understandable.

Glossary of Rate Terms

Load Factor. The ratio, in per cent, of the average hourly load to the maximum hourly load. This is usually based on a one year period but may be applied to any specified period.

Demand Factor. The relation between the connected radiator surface or required radiator surface and the demand of the particular installation. It varies from 0.25 to 0.3 lb per (hour) (square foot of surface).

Diversity Factor. The ratio of the sum of the individual demands of a number of buildings to the actual composite demand of the group.

Types of Rates

1. Flat Rates.
   - *a.* Radiator surface charge. *Obsolescent.*
2. Meter Rates.
   - *a.* Straight-line.
   - *b.* Step. *Obsolescent.*
   - *c.* Block.
     - (*a*) Class rates.

Straight-Line Meter Rate. The price charged per unit is constant, and the consumer pays in direct proportion to his consumption without regard to the difference in costs of supplying the individual customers.

Block Meter Rate. The pounds of steam consumed by a customer are divided into blocks of thousands of pounds each, and lower rates are charged for each successive block consumed. This type of charge predominates in steam heating rate schedules for it has the advantage of proportioning the bill according to the consumption and the cost of service. It has the disadvantage of not discriminating between customers having a high load factor (relatively low demand) and those having a low load factor (relatively high demand). The utility company must maintain sufficient capacity to serve the high demand customers and the cost of the increased plant investment is divided equally among the users, so the high demand customers are benefited at the expense of the others.

3. Demand Rates.
   a. Flat demand.
   b. Wright.
   c. Hopkinson.
   d. Doherty (or Three charge).

Demand Rates. These refer to any method of charge based on a measured maximum load during a specified period of time.

The *flat demand rate* is usually expressed in dollars per thousand pounds of demand per month or per annum. It is based on the size of a customer's installation, and is seldom used except where a meter is not practicable.

The *Wright demand rate* is similar in calculation to the block rate except that it is expressed in terms of hours' use of the maximum demand. It is seldom used but forms the basis for other forms of rates.

The *Hopkinson demand rate* is divided into two elements:
(*a*) A charge based upon the demand, either estimated or measured.
(*b*) A charge based upon the amount of steam consumed.

This rate may be modified by dividing the quantities of steam demanded and consumed into blocks charged for at different rates.

The *Doherty rate* is divided into three elements:
(*a*) A charge based upon demand.
(*b*) A charge based upon steam consumed.
(*c*) A customer charge.

In the *Hopkinson rate*, the last two elements are combined into one element.

Demand rates are comparatively new and are not yet widely used; though they are equitable and competitive they are difficult for the average layman to understand. They are of benefit to utility companies and to consumers because the investment and operating costs can be divided to suit the particular circumstances into demand, customer, and consumption groups through the use of some modification of the Hopkinson rate. Demand rates are an advantage to the customer in that the use of such a rate reduces the rate per thousand pounds to the long-hour user.

Fuel Price Surcharge. It is usually desirable to establish a rate upon a specified basic cost of fuel to the utility company. Where there are wide variations in the price of fuel, it is also desirable to add a definite charge per thousand pounds of steam sold for each increment of increase in the price of fuel. This surcharge automatically compensates for the variations without necessitating frequent changing of the whole rate structure.

Some utility companies include a labor surcharge as well as a coal surcharge.

UTILIZATION

Considerable savings can be made by the proper and intelligent operation of heating systems. It should be borne in mind that a heating system is designed to heat a building to 70 F inside when the outside temperature is at its lowest point for that particular locality. There is a tendency to overheat the building at any time the outside temperature is above the design temperature unless some method of regulation is used, either automatic or manual.

The general rules for economical operation[2] are as follows:

1. *Reduce the heat losses from the building to a minimum.*
   a. Weatherstrip all windows, and caulk all window frames.
   b. Provide revolving or vestibule doors on all entrances. Separate shipping and receiving rooms from the remainder of the building by partitions so that the large doors will not ventilate the entire building.
   c. Eliminate all unnecessary ventilation. Ventilating equipment is usually sized to meet extreme requirements. In a theater or auditorium, do not supply enough ventilation for an audience of 2000 when there are only 200 present.

2. *Limit the hours of heating to those in which the required temperature is necessary.*
   a. Determine the hours that heating is required and see that steam is shut off for the maximum time when not required, such as nights, Sundays, and holidays.
   b. Shut steam off entirely in unoccupied sections of the building, taking care to avoid freezing plumbing.
   c. Install separate lines for those parts of the building that require long-hour or 24-hour heating. This is much cheaper than heating the entire building.
   d. Control the heat supplied to water storage tanks located on or above the roof. Such tanks require heat to prevent freezing when the outdoor temperature is below 32 F.

3. *Regulate the amount of heat so as to prevent overheating and to maintain uniform temperatures during the hours of occupancy.*
   a. Determine the temperature required for the occupancy of a building. Do not heat a storage garage or a furniture warehouse to the temperature required in a hospital ward.
   b. Shut off steam during the day whenever possible. An automatic control will do this, but it can be done by hand, with good results.
   c. Provide some good means of temperature control.

4. *See that the heat input is properly balanced throughout the building.*
   a. See that the entire heating system responds rapidly when steam is turned on. Locate and eliminate the cause of any sluggish circulation. Balance the radiation, provide adequate air elimination, and correct any trapped run-outs to provide quick system drainage.
   b. Place the radiation near the outside walls under the windows or where the exposure occurs, if possible.
   c. Do not obstruct radiators or prevent the free circulation of air around them; to do so seriously reduces the heating capacity of a radiator.

5. *Keep all heating equipment in first class condition.*
   a. Keep the system in good repair. This applies to all traps, valves, vents, steam and return piping, vacuum pumps, and temperature control apparatus.
   b. In a vacuum system, maintain the degree of vacuum recommended by the control manufacturer. If this is not possible, locate and eliminate all leaks.
   c. Insulate all steam pipes not used as heating surface.

6. *Arrange the heating system to obtain from it the highest possible efficiency.*
   a. Locate all valves and controls so as to be convenient and accessible. It is only human nature to delay or avoid doing that which is unnecessarily inconvenient.
   b. Investigate every complaint of "No Heat;" find the cause and correct it. Do not overheat an entire building to correct a local condition.
   c. Extract the heat in the condensate for heating water or for some other useful purpose.

7. *Make a study of the heating system and heating requirements.*
   a. Provide thermometers and recording pressure gages so that the heating system may be operated with full knowledge of what is being accomplished.
   b. Keep daily consumption records and check against the theoretical requirements.
   c. Study the system and understand its functions and its operations.

AUTOMATIC TEMPERATURE CONTROL

As stated in Chapter 34, Automatic Control, properly applied to heating, ventilating and air conditioning systems, makes possible the maintenance of desired conditions with maximum operating economy. The use of adequate temperature control provides more healthful, comfortable, and efficient working conditions in buildings.

There are three general means of obtaining centralized control of heat output of radiators.

1. *Controlling the rate of steam flow* into the radiators. This is accomplished by equipping the radiator inlets with orifices and controlling the flow of steam through them into the radiator by controlling the difference in pressure between the supply and return.

2. *Controlling the temperature of steam* in the radiators by varying its pressure. This involves the use of high vacuums to obtain low steam temperatures. This must be supplemented by some other type of control for low heat output.

3. *Controlling the length of time steam flows* into the radiators by admitting steam to a heating system intermittently and varying the length of the *on* and *off* periods. Two types of controls are used. (1) A clock control providing *on* and *off* settings of various lengths, which can be changed in accordance with outside temperatures. In most cases these changes are made automatically by means of a thermostatic bulb, placed outdoors. (2) A control, having an outdoor bulb and a bulb attached to the radiator, which varies the length and frequency of the *on* intervals in such a way that the radiator temperature is varied according to the outside temperature. In some cases heat supply is controlled by combinations of the three methods described.

Before installing any type of modern temperature control equipment, it is necessary to see that the heating system is put in good operating condition. In general, the heating system in a building is not given the attention that other mechanical equipment is given because it will continue to function, after a fashion, even though changes in piping, location of radiation, settlement of piping, and the normal wear and tear or other changes have taken place. Because of this depreciation of the system, operation becomes more and more costly and parts of the building have to be greatly overheated in order to prevent underheating in other parts. Vents, traps, vacuum pumps, and valves should be given a careful inspection and replaced or repaired if required. The piping should be of adequate size and graded properly. The return piping should be inspected, and any pockets or lifts removed and properly vented. These inspections and repairs are not costly and may prevent a much greater outlay in future years. In most cities district heating companies will be willing to make a survey of heating systems and offer recommendations in regard to operation and changes in piping layout.

The selection of control equipment depends upon the type and size of building and the degree of saving which may be obtainable.

REFERENCES

1—Code for Pressure Piping, B 31-1, 1942, *American Standards Association*, Paragraph 408, p. 115.
2—Principles of Economical Heating, *National Association of Building Owners and Managers.*

CHAPTER 30

Electric Heating

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Boilers, Electric Hot Water Heating, Heating Domestic Water Supply, Reversed Cycle Refrigeration, Auxiliary Electric Heating, Control, Calculating Capacities, Radiant Drying, Induction and Electrostatic Heating, Power Problems

ELECTRIC heating is steadily assuming a more important place in heating, ventilating and air conditioning installations because it is flexible, clean, safe, convenient and easy to control. It has many basic principles in common with fuel heating, but there are also important differences. When heat is delivered by wire, no combustion process is necessary, either at a central plant or at the individual room units. The output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and it follows that the total load on an electric heating system is the total wattage of connected electric heaters, regardless of weather conditions. The main obstacle to the more general adoption of electric heating for buildings is the cost of the electricity itself.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first*, the conversion of energy into heat; *second*, the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, as 100 per cent of the energy applied to the resistor is always transformed into heat. In electric heating practice no concern need be given to efficiencies of heat production, but rather to efficiencies of heat utilization. The problem is to distribute the electrically produced heat units in such manner as to obtain conditions of maximum comfort with the minimum consumption of electricity.

DEFINITIONS

Definitions of general terms used in fuel heating are given in Chapter 1. Terms which apply particularly to electric heating are as follows:

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

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

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

RESISTORS AND HEATING ELEMENTS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor which is heated by an alternating current of electricity passing through it. One of the more common

resistors is nickel-chromium wire or ribbon which, in order to avoid oxidation, contains practically no iron.

Commercial electric heating elements are made in many types. Some have resistors exposed to the air being heated. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. This type of element is used extensively for operation at high temperatures when radiant heat is desired, also at low temperatures for convection and fan circulation heating, especially in large installations.

Some elements have metallic resistors embedded in a refractory insulating material, encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Elements are made in many forms, such as strips, rings, plates and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in some types of convection air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. Cloth fabrics woven from flexible resistor wires and asbestos thread are used for many low temperature purposes such as heating pads and aviators' clothing.

Special incandescent lamps are used as heating elements in certain applications where radiant heat is desired. These use carbon or tungsten filaments as resistors, and are designed to produce maximum energy in the infra-red portion of the spectrum.

ELECTRIC HEATERS

Electric heaters may be divided into four groups: conduction, radiant, convection, and induction.

Conduction electric heaters, which deliver most of their heat by actual contact with the object to be heated, are used in such applications as aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and water heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly upon the body. They are not satisfactory for general air heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. For a discussion of electrically heated panels as applied to radiant heating, see Chapter 31.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors. They generally have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from the floor line. The flexibility possible with electric heating elements should discourage the use of secondary mediums for heat transfer. Water and steam add nothing to the efficiency of an electric heater and entail expensive construction and maintenance.

Induction Heaters are described in the section on Electronic Heating by Induction and Electronic Means.

UNIT HEATERS

Electric unit heaters include a built-in fan unit which circulates room air over the heating elements. Heaters of this type are manufactured in many designs and sizes, and can be located in the same manner as steam unit heaters.

Electric unit heaters are used in industrial plants, sub-stations, power houses, pumping stations, etc., where the power rate for electric heating is found to be favorable. In many large plants, such as flour mills, grain elevators, etc., in which there are a number of small offices, locker rooms, etc., scattered over wide areas, electric unit heaters are frequently economical in such locations. In small unattended stations, where freezing temperatures cannot be permitted, thermostatically-controlled electric unit heaters are frequently used to maintain a temperature above freezing. The best location for the heaters depends upon local circumstances as they can be mounted on the ceiling to direct the air downward,

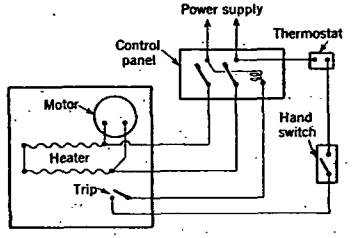


FIG. 1. WIRING DIAGRAM FOR UNIT HEATER

on the side wall about 7 ft from the floor, or near the floor line. Variations in design are necessary for different locations, as with steam unit heaters, see Chapter 26.

The arrangement of the wiring circuits is very important for electric unit heaters. In principle they are all the same and include as essential elements an automatic control panel, a thermostat, and a master hand switch. All heaters should be designed with a safety thermal trip wired in series with the magnet coil of the control panel and with the hand switch and thermostat. A typical wiring diagram is shown in Fig. 1. This applies to a single phase power supply, but for 3-phase the only difference is to have a 3-pole panel and a heater arrangement for 3-phase connection.

Portable unit heaters are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction.

CENTRAL FAN HEATING

Electric heating elements can be used for the prime source of heat in a central fan electric heating system or in the heating phase of a complete air conditioning system. They can be used in the same manner as steam heating units for tempering, preheating or reheating the air at the main supply fan location and as booster heaters at the delivery terminals of the duct system. In the humidification phase of air conditioning electric heating elements can be used to provide moisture by the evaporation

of water, or for controlling air washer dew-point temperatures when mounted as preheating units on the intake side of the air washer. (See Chapter 43.)

In coordinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature source of heat for any given pressure and a change in air volume flowing over steam coils does not greatly affect the temperatures of the delivered air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, having a constant input of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered. This occurs because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant unless some compensating action is performed by control. Automatic variation of the electrical heat input synchronized properly with the air flow can be successfully accomplished by various special methods of control. By-pass dampers as used in steam units will not control electric heat.

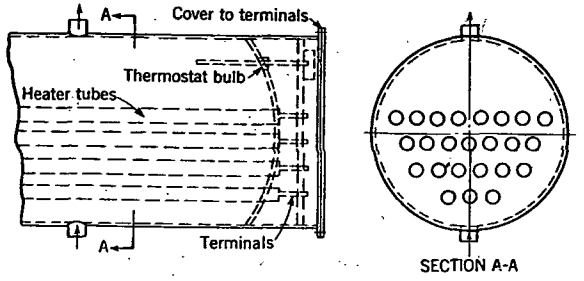


FIG. 2. RESISTANCE TYPE BOILER FOR STEAM OR HOT WATER

Electric heaters are useful in balancing the heat distribution in central fan heating systems. Even in those instances where steam is the principal heat source, the temperature of individual rooms can be controlled locally by separate electric booster heaters. These heaters can be installed in branch ducts or behind the air outlet grilles in each room. With this arrangement, the central heating unit distributes air at an average temperature, controlled from a thermostat centrally located, such as in the main return duct. The electric booster heaters may be controlled by thermostats mounted in each individual room which permit the occupant to maintain any desired temperature independent of the rest of the building.

ELECTRIC BOILERS

Steam or hot water generating boilers using electric energy are entirely automatic and are well adapted to intermittent operation. Small electric boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used either with direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood of the heating elements burning out, they should be of substantial construction, with a low heat density per unit of surface area and provision should be made for cleaning off deposits of scale which restrict the heat flow. A typical resistance type of steam or hot water boiler is shown in Fig. 2.

Large electric boilers are usually of the type employing water as the resistor, using immersed electrodes. With this type only alternating current can be used, as direct current would cause electrolytic deterioration. Such a type of electrode boiler is shown in Fig. 3.

Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and also for sterilizers, jacketed vessels and pressing machines which need a ready supply of steam. It sometimes is economical to shut down the main plant fuel burning boilers when the heating season ends, and to supply steam for summer needs with small electric steam boilers located close to the operation. In general, electric steam heating is confined to auxiliary or other limited applications. If the heating system is designed to use electricity exclusively, steam generating or distributing equipment is superfluous.

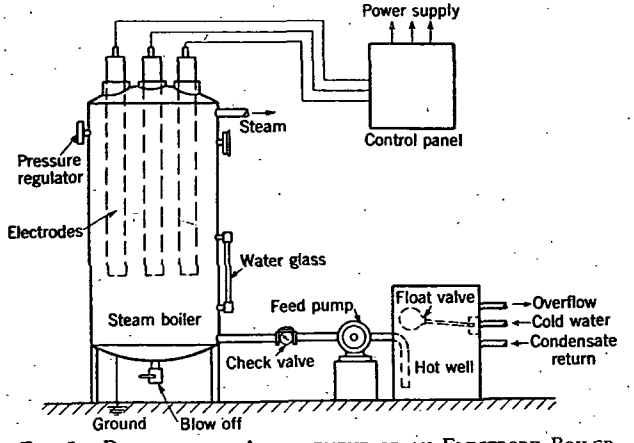


FIG. 3. DIAGRAMMATIC ARRANGEMENT OF AN ELECTRODE BOILER

ELECTRIC HOT WATER HEATING

Electric water heating, using an electric boiler in place of a fuel burning boiler, like electric steam heating, is generally confined to auxiliary or other limited applications. The use of insulated water storage tanks, in which to store heat generated by electricity during off-peak hours at extremely low rates, is a development which has some special applications.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with electric heating elements and automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the building may be of any standard individual radiator or fan-served indirect type or with provisions for the heating and humidification phases

of an air conditioning system. A system of this kind requires very careful design to avoid excessive over-all radiation losses during periods of low heat demand. It is also important to provide for sudden changes in heat demand. A typical hot water heating boiler is illustrated in Fig. 2.

HEATING DOMESTIC WATER BY ELECTRICITY [1]

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable. In many sections of the country low electric rates have been established by the electric utilities to secure this load. In many localities, electric rate schedules divide the current used for water heating into two classifications, regular and off-peak. A time switch automatically limits use of the off-peak heating element to the hours of off-peak load, while the regular heating element is a stand-by at all times. Storage of this two-element type of water heater is larger than average to carry over the periods when the off-peak element is timed

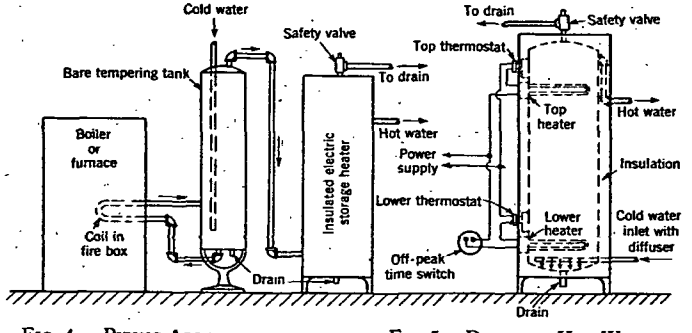


FIG. 4. PIPING ARRANGEMENT FOR CONNECTING ELECTRIC WATER HEATER TO FIRE-BOX COIL

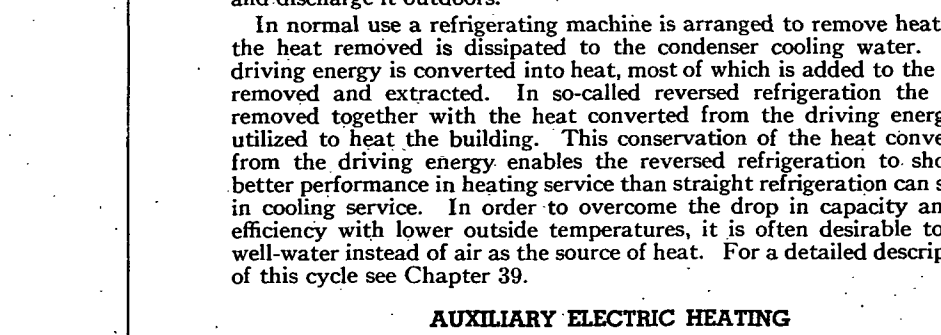


FIG. 5. DOMESTIC HOT WATER HEATER FOR OFF-PEAK SERVICE

out, without too frequent demands on the regular heating element which takes the higher domestic lighting service rate. Some utilities now offer a schedule which, beyond a stipulated minimum, lowers the rate for all electric service if an electric water heater is installed.

Competition with other fuels, especially gas, seems to be the major controlling factor in the use of electricity. The first cost of electric storage heaters is greater than for gas, owing to the need for larger tank storage due to off-peak service and slower recuperating capacity.

In residential work, to effect a saving in the cost of operation, it is sometimes desirable to use a furnace coil or indirect heater in connection with an electric water heater. In this case it is important to make the proper connections in order to benefit by any heat obtained from the furnace and at the same time to prevent dangerous overheating. The proper piping connections are shown in Fig. 4, and in this case the electric heater will only furnish heat when insufficient heat is supplied from the furnace. This arrangement has a further advantage in the summertime in that the bare tank through which the cold water passes on its way to the electric heater serves as a tempering tank, absorbing heat from the basement air and requiring the use of less energy in the electric heater.

A typical domestic hot water heater as shown in Fig. 5 is arranged with

upper and lower heating elements for the usual type of off-peak heating service. The lower heating element is under the control of the off-peak time switch. However, the upper heating element is usually connected to the line so that, in case the supply of hot water in the tank becomes exhausted, the top thermostat can turn on the top heater and heat a small supply of water. The top heater will not heat the water in the tank below its location, but when the off-peak period arrives the lower heater is turned on and the entire tank becomes heated.

HEATING BY REVERSED CYCLE REFRIGERATION

Reversed refrigeration is frequently referred to as a *heat pump* since the electric motor driving the refrigerating compressor furnishes the motive power to transfer heat from one temperature to a higher temperature level. The compressor acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and discharge it outdoors.

In normal use a refrigerating machine is arranged to remove heat and the heat removed is dissipated to the condenser cooling water. The driving energy is converted into heat, most of which is added to the heat removed and extracted. In so-called reversed refrigeration the heat removed together with the heat converted from the driving energy is utilized to heat the building. This conservation of the heat converted from the driving energy enables the reversed refrigeration to show a better performance in heating service than straight refrigeration can show in cooling service. In order to overcome the drop in capacity and in efficiency with lower outside temperatures, it is often desirable to use well-water instead of air as the source of heat. For a detailed description of this cycle see Chapter 39.

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electric heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a building comfortable. Likewise, such electric heating might be used on abnormally cold days to help out the main heating system and by this means reduce the necessary size of the system.

A few installations have been made using electric heating cable buried in the floors of bathrooms, etc., to provide auxiliary electric heating. At least one airplane hangar is heated in this manner.

Because of the feeling of comfort that a radiant type heater gives, bathrooms may be heated electrically with this type of heater while the rest of the house is cared for by some other system. Offices and rooms which require heat at periods when the main heating plant is shut down can be conveniently heated electrically.

CONTROL

Because the efficiency of electric heat production is the same for small and large units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Heaters are often controlled manually but thermostatic control is essential for economical operation. For duct systems having a variable volume of air flow the electric heater control must automatically vary the heat input in coordination with the changes in air volume and demand for heat.

CALCULATING CAPACITIES

In calculating electric heating capacity one kilowatt is equal to 3413 Btu per hour or 14.2 sq ft equivalent direct steam radiation.

INDUSTRIAL USES FOR ELECTRIC HEAT

Electric heating is valuable for many industrial processes in both low and high temperature ranges. It is easily controlled and can be justified where savings in labor or improved quality of product outweigh the inherent higher cost per unit of heat as compared with other fuels. Important examples are metal melting and heat treating furnaces, ovens, dryers, laboratory equipment, cooking vessels, oil preheaters, catalysts, and countless other special uses.

RADIANT DRYING

Lacquers and similar surface films can be very effectively dried by radiation. Special electric lamp bulbs have been developed which give off a high percentage of infra-red and similar heat rays[2]. These are mounted in very efficient reflectors. For continuous manufacturing processes these reflectors are mounted in tunnels through which conveyors pass. For local applications, as for example paint drying in automobile repair shops, they may be mounted on portable racks.

In the application of this type of drying the composition of the paint or lacquer is important. In general, lacquers and those enamels using synthetic resins react most favorably. Other applications include the drying of ink, glue, and water, the softening of celluloid and bakelite for punching or shearing, and a wide variety of other uses[3].

Objects of relatively large surface area in proportion to their weight, and fabricated materials having a rather high heat absorption, may be satisfactorily heated by such a source.

ELECTRONIC HEATING BY INDUCTION AND ELECTROSTATIC MEANS

These methods differ radically from resistance heating as they employ high frequency radio waves to apply the energy which produces heat. High frequency heating has many important industrial uses and opens up a whole new field of special applications where extreme accuracy and speed are vital. Some spectacular results are being attained with this modern industrial tool. Skillful engineering design and experience are necessary to produce safe and satisfactory performance, but this technical assistance is now available from many sources.

Metals can be heated by induction. When the work is placed in a magnetic field within a high frequency coil, eddy currents immediately produce heat in the body of the metallic piece. The speed and intensity of this heating can be regulated by controlling the high frequency currents producing the magnetic field and the location of the spot heated by the position of the work piece within the coil. Induction heating is very useful in special processes such as melting metals, brazing, forging, heat treating, etc. It is possible to apply localized heat so rapidly that conduction cannot draw the heat away before it has time to accomplish the desired purpose at a particular spot. One example is the rapid hardening of a tool edge or tip too quickly for scale to form.

Dielectric materials can be heated internally by introducing them into an electrostatic field between high frequency electrode plates. Foods can be sterilized, plywoods bonded, plastics heated, granular or crystalline materials dehydrated, and countless other products heated quickly and uniformly, although they are poor thermal conductors and resist heat applied to their exteriors. Electrostatic heating is ideally suited for continuous production processes as the materials can pass through the heating field quickly with very short exposures to the high frequency radio waves.

POWER PROBLEMS

The cost of electric energy varies because of several factors. Distribution costs differ for large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate electric plants at uniform loads; hence, even the time of use may affect the cost of electricity. Special low rates are sometimes available during certain prescribed hours of use.

Since the cost of production and distribution depends not only upon the quantity of energy used but also upon the maximum rate at which it is used, electric energy is often sold on a demand rate basis. In some cases, the demand charge is based upon the rated connected load, in other cases, upon the maximum demand as indicated by a demand meter.

Homes are almost universally supplied with lighting current of 115 volts, which can only be used economically for small heaters. Usually the service lines will not permit more than plug-in devices. The Underwriters permit approved heaters of 1320 watts or less to be plugged into approved baseboard receptacles, but such heaters cannot be served on a circuit supplying much other load without overloading the fuses. There is an increasing trend toward supplying homes with three wire 115-230 volt service. Where homes have such service, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 208, 220, 440 or 550 volts. All polyphase heaters should be balanced between phases. In ordering electric heaters the proper voltage must be specified as the heat produced will vary as the square of any variation in voltage.

REFERENCES

[1]—Application of Electric Water Heaters To Domestic Service, by C. G. Hillier (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1936, p. 632). Fourteenth Range and Water Heater Survey (*Electric Light and Power*, August, 1940).

[2]—Infra-Red Lamps Speed Up Drying Operations (*Automotive Industries* 82:376-7; April 15, 1940). Invisible Rays Build Visible Profits, by H. M. Archer (*Electric Light and Power*, May, 1940). Radiant Energy Drying and Baking for Organic Finishing (*Metal Industry* 38:294-6; May, 1940).

[3]—Infra-Red Heating, Section IV, Power Sales Manual (*Edison Electric Institute*).

BIBLIOGRAPHY

Electric Elements Well Adapted to the Air Conditioning Heating Cycle, by L. P. Hynes (*Heating, Piping and Air Conditioning*, January, 1940, p. 29).

Electric Heat for the Bathroom Floor (*Heating and Ventilating* 36:41; December, 1939).

Electric Heat Spreads in Sunny Climes (*Barron's National Financial Weekly* 20:5; January 8, 1940).

Electric Heating for Los Angeles Building (*Heating and Ventilating* 37:50-1; June, 1940).

Dracker Apartments, West Los Angeles, Calif. (*Heating and Ventilating* 37:32; July, 1940).

Electric Heat in California Homes, by P. F. Offerman, W. J. Walsh and H. H. Skilling (*Electrical West*, March, 1941, pp. 37-39).

Radiant Heat, by F. W. Hutchinson (*Electrical West*, January, 1941, pp. 29-31).

Electric Radiant Heat vs. Steam Convection, by Douglas Dow (*Electrical World*, August 10, 1940, pp. 61-62).

Practical Aspects of Heating Residences by Electricity, by F. L. Lawton and P. Tellier et al (*Electrical News & Engineering* (Canada) July 1, 1934, pp. 32-33 and 40; July 15, 1934, pp. 29-32; August 1, 1934, p. 17; July 1, 1933, pp. 16-20; July 15, 1933, pp. 21-22; August 1, 1933, pp. 23-25).

Off-Peak System of Electric Heating for Buildings, by Elliott Harrington (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 323).

Electric Heating of Residences, by Edgar Allan Loew (University of Washington, *Engineering Experiment Station*, Part I, *Bulletin* No. 15, December, 1921; Part II, *Bulletin* No. 20, November 15, 1923).

Reversed-cycle Refrigeration for Air Conditioning Work, by Regis D. Heitchue (*Refrigerating Engineering*, May, 1941, pp. 317-321).

South American Way on This Buenos Aires Job Was to Use a Heat Pump, Melvin A. Ramsey (*Heating, Piping and Air Conditioning*, March, 1941, pp. 167-170).

The Sixth Ingredient—The Heat Pump—The Importance of Its Development in Making the All-Electric Home a Reality, by Philip Sporn (*E.E.I. Bulletin*, August, 1944).

Description and Performance of Two Heat Pump Air Conditioning Systems (Using Well Water and Outside Air as the Heat Source), by Philip Sporn, and E. R. Ambrose (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944).

Heating of Non-Magnetic Electric Conductors by Magnetic Induction, by R. M. Baker (*Electrical Engineering*, June, 1944).

What High Frequency Heat Treating Can Do (*Electrical Manufacturing*, July, 1943, Vol. 32, Number 1).

Electronic Devices Aid Metallurgical Research, by E. V. Potter (*Electrical Engineering*, May, 1944).

Designing an Induction Heating Product, by Charles R. Underhill (*Electrical Manufacturing*, June, 1944).

CHAPTER 31

Panel Heating and Radiant Heating

Influence of Heat Radiation on Human Comfort, Objectives of Radiant Heating, Practical Problems of Radiant Heating from a Physiological Standpoint, Fundamental Computations, Application Methods, Calculation Principles, Measurement and Control

IT has been pointed out in Chapter 12 that the human body loses heat to its environment in three ways; by convection, radiation, and evaporation. The Effective Temperature Chart takes account of convection and evaporation, but does not provide for such radiative effects as occur when room air and its surrounding surfaces differ widely in temperature.

INFLUENCE OF HEAT RADIATION ON HUMAN COMFORT

When, however, the body is exposed to radiation from a hot surface or is radiating to a cold surface, the factor of radiative heat gain or heat loss may be important. This phenomenon is most marked in the case of exposure to the sun's radiative heat. On a cold day, with no wind blowing, while standing in the sunshine, one may feel perfectly comfortable but, when a cloud passes over the sun, one may instantly feel much cooler. The cloud acts as a shield to interrupt the radiant heat from the sun. The change in feeling of comfort is due to the instant change in rate of heat loss from the body caused by the shielding effect of the cloud. A shielded thermometer under the same condition would register no change in temperature.

The rate of heat loss by convection depends upon the average temperature difference between the surface of the body and the surrounding air, the shape and size of the body, and the rate of air motion over the body.

The rate of heat loss by radiation depends upon the exposed surface area of the body, and upon the difference between the mean surface temperature of the body and the mean surface temperature of the surrounding walls or other objects. This latter temperature is called the Mean Radiant Temperature (MRT).

Because these two types of heat loss supplement each other, a required rate of total heat loss can result either from a relatively low air temperature and a relatively high MRT, or vice versa.

At the temperature which produces comfort (and at all lower temperatures) the production of sweat is low and the heat loss by evaporation is relatively low and relatively constant, irrespective of the relative humidity of the atmosphere. Under such conditions the heat loss from the body is chiefly related to the combined effect of convection and radiation. The heat demand of the environment, so far as these two factors are concerned, may be measured by Operative Temperature, which is defined by the following formula, modified from that of Gagge [1] by the expression of air velocity in feet per minute and temperature in Fahrenheit degrees.

$$t_o = 0.81\, t_w + 0.135\, [\sqrt{V}\, t_a - (\sqrt{V} - 1.40)\, t_s],$$

where

t_o = operative temperature, Fahrenheit degrees.
t_w = mean radiant temperature; Fahrenheit degrees.
t_A = air temperature, Fahrenheit degrees.
t_s = mean skin temperature, Fahrenheit degrees.
V = air velocity in feet per minute.

At high environmental temperatures, a given Operative Temperature with cold air and hot walls produces a slightly greater cooling effect on the body than the same Operative Temperature with air and wall temperatures equal, probably on account of local cooling of the nose and throat[2]. This phenomenon is not important in the comfort zone.

Under ordinary conditions, with normally clothed human beings in still air, the mean skin temperature of the body is about 90 F (with lower values for the extremities); and the mean temperature of the surface of the clothing is about 86 F.

The normal rate of heat production in an average sized sedentary individual is about 400 Btu per hour. The heat production for persons subjected to various rates of activity is given in Chapter 12. The human body is of complicated shape, and radiation takes place freely only from the exposed outer surfaces; there are considerable portions of the body such as the legs, arms, lower part of the head, etc., which radiate most of their heat to other portions. It is necessary to determine the equivalent surface of the body from which heat is radiated and a similar value for convection. The total may be assumed to be about 19.5 sq ft for convection and 15.5 sq ft for radiation, in an average sized individual.

The loss by respiration and by evaporation from the nose and throat depends on the temperature and area of the moist surfaces (respiratory) of the body, the air temperature, air movement, and humidity. In air at a temperature of 70 F, this loss, for a sedentary individual of average size, will be approximately 90 Btu per hour; and at 60 F about 70 Btu per hour. These values are relative, because the total will vary materially with change of position, bodily activity, age, sex, race, etc.

The balance of the heat generated in the average human body, approximately 300 to 320 Btu per hour at about 70 F room temperature, is the approximate amount of heat given off by radiation and by convection from the external body surfaces. Under normal conditions (in still air), the radiation loss will be about 190 Btu per hour; and the convection loss about 120 Btu per hour. With an air velocity of 520 fpm, comfort will require an increase in Operative Temperature of nearly 12 deg; under such conditions the convection loss will rise to 250 Btu per hour but comfort may be attained if the subject is surrounded by heated walls which keep the radiation loss at about 50 Btu[3].

It is neither feasible nor desirable to change the relationships of convection and radiation very greatly in actual heating practice. In the laboratory, where the laws of radiative heat loss have been deduced, it is necessary to produce wide differences between radiative and convective heat loss. This can only be accomplished, however, by elaborate and powerful conditioning apparatus which simultaneously heats walls and cools air, or vice versa. Such a process would be very costly in practice and would not be justified unless marked improvement in comfort resulted from such a condition—an assumption which has not been demonstrated. In practice, where radiant heat is introduced into a room,

that heat is absorbed by surfaces, furniture, and the like, and then transformed into convective heat so that air and surfaces tend to attain a generally uniform temperature.

OBJECTIVES OF RADIANT HEATING

Under ordinary circumstances the human being, indoors, is not subjected to marked variations between the factors affecting convection and radiation. Air and walls are not commonly very far apart in temperature; air movement and relative humidity are usually low. Where such conditions obtain, the ordinary air thermometer is a good measure of comfort —which is the reason why it has enjoyed such universal use. Where considerable window surfaces create heavy radiation loss, or where stoves or open fires, or very hot ceilings contribute to large radiation gain, the picture is changed and the air temperature productive of comfort must be correspondingly modified.

In general, however, radiant heating of occupied spaces is not a procedure designed to create differences between air and walls, but is merely one method of introducing heat into that space. The engineering factors used in determining desirable heat input will be essentially the same as if the heat were introduced by convection, or in any other way.

PRACTICAL PROBLEMS OF RADIANT HEATING FROM A PHYSIOLOGICAL STANDPOINT

It is convenient to distinguish two different methods of introducing radiant heat into an enclosed space. The first, which may be called High-Temperature Radiation, involves direct exposure of the occupied parts of the room to radiation emitted from relatively small heating units of very high temperatures (perhaps 1,000 F); the second, Panel Heating, involves exposure to relatively large surfaces at not over 130 F.

High-Temperature Radiant Heating may be useful for temporary purposes, as in the use of a bathroom heater. It is, however, generally an undesirable process (except in rooms of great height) on account of the marked unevenness of the effect produced on the human body. Studies at the John B. Pierce Laboratory of Hygiene have shown that this type of heating produces uncomfortable differences in the temperature of different parts of the body (an over-heated head, for example, if the heat comes from the ceiling).

Panel Heating, on the other hand, is advantageous from the standpoint of temperature differentials. In actual practice, a well-designed system of this sort produces very uniform conditions, the air throughout the room differing at various points by only 5 deg. This is desirable from the comfort standpoint and may also be a factor in heat economy, since high temperatures in the upper part of the room favor excessive heat loss. The esthetic value of such a system is also considerable, since it avoids the presence of registers or free-standing radiators in the room.

In the design of Panel Heating, however, careful thought must be given to the location of the panels from the standpoint of comfort. The English commonly use the ceiling for their panels, but their rooms are generally high-studded, and outdoor winter temperatures moderate. With low ceilings even panels may produce an excessive directional heating effect if all the heat necessary in a cold climate is introduced from above. Similarly, if the floor alone is used, it may—in very cold weather—be necessary to make the floor too hot for comfort. Wall panels, or a combination of ceiling and floor panels, will perhaps produce the best results.

FUNDAMENTAL COMPUTATIONS

The mean surface temperature of an inert body, which will cause given rates of heat loss by radiation and by convection in a uniform environment, having a given air temperature and a given mean wall temperature, may be calculated from fundamental equations[4] for radiation and natural convection, with substitution of comparable cylinders for the irregular human body.

$$q_r = 0.1730\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \quad (1)$$

$$q_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \quad (2)$$

where

q_r = heat loss by radiation, Btu per (square foot) (hour.)
q_c = heat loss by convection, Btu per (square foot) (hour.)
T_s = absolute temperature of the body surface, Fahrenheit degrees.
T_w = absolute temperature of the walls, Fahrenheit degrees.
T_a = absolute temperature of the air, Fahrenheit degrees.
$T_m = \frac{T_s + T_a}{2}$
D = diameter of cylinder, inches.
e = the ratio of actual emission to black body emission.

If it is assumed that an average adult has a height of 5 ft 8 in., a body surface of 19.5 sq ft for convection, and 15.5 sq ft for radiation, an equivalent effect can be worked out for two cylinders, 5 ft 8 in. high by 13.15 in. diameter and 10.45 in. diameter, respectively. However, while the effects on a cylinder, of a particular size and shape may be used to estimate average similar effects on the human body, it should be remembered that the heat loss from the body varies greatly. Every movement alters not only its shape, but also the heat generated by the body, the velocity of the air passing over it and the surface exposed to radiation. This fact renders the results of any such computation only approximate.

APPLICATION METHODS

The several methods of applying panel and radiant heating to a structure are:

1. *By warming the interior wall and ceiling surface of the building.* Pipe coils are imbedded in the concrete or plaster of the walls or ceilings, the heating medium being hot water circulating through the pipe coils. These coils are generally constructed of small pipe ½ or ¾ in. I.D. and spaced about 6 to 9 in. apart. See Fig. 1. This has the effect of warming the entire concrete or plaster surface in which the pipes are imbedded. Since the temperature of the heating medium should never exceed about 130 F, due to the possibility of cracking the plaster the area of the warmed surface must be sufficient to supply the requisite quantity of heat at this low temperature. When carefully designed, this method produces very comfortable results and great operating economy, but offers some slight obstacles when alterations or additions to the building are desirable. Normally the hot water circulation is maintained by means of a circulating pump and facilities have to be provided to eliminate all air at the top of the system. All coils and circulating pipes are welded together and tested after erection to a hydraulic pressure of 300 psi.

2. *By circulating warm air through shallow ducts under the floor.* In this design the entire floor surface of a room is heated as in Fig. 2. This method was used 2000 years ago in many parts of the Roman Empire. While this method is more expensive in construction, it is effective and quite suitable for cathedrals and large public buildings. To provide a uniform floor temperature, one should give special consideration to the design of the air ducts so that equal heat distribution is obtained.

3. *By placing hot water pipes in or under the floor.* With this arrangement the whole floor surface of a room is raised to a temperature sufficient to give comfortable conditions. Floor heating is recommended for schools and hospitals where large quantities of outside air are desirable. The floor surface may be of concrete, wood blocks, marble or any other material unaffected by heat, and while it is true that heat will be conducted through all materials used in floor construction, it is important that due consideration be given to the emissivity of the floor. In some cases where pipe coils are installed in the air space

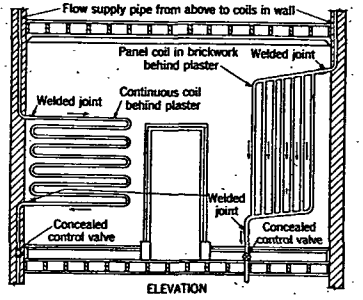


FIG. 1. COILS IN WALL SURFACES

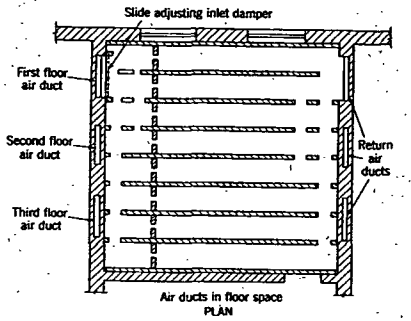


FIG. 2. AIR DUCTS FOR FLOOR HEATING

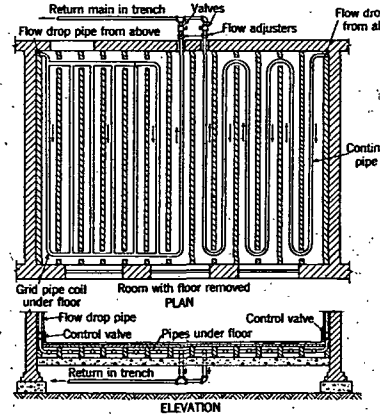


FIG. 3. CONTINUOUS COIL IN FLOOR

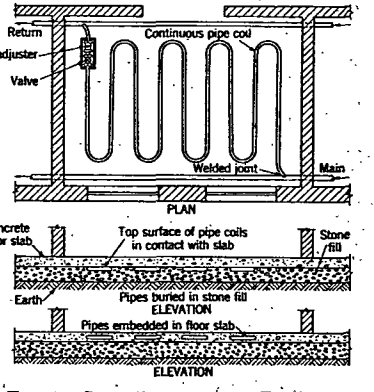


FIG. 4. COILS EMBEDDED IN FLOORS

under the floor, special floors are constructed in sections so that the whole floor can be lifted to examine the coils. See Fig. 3. Pipes supported thus may be larger and the heating medium maintained at a higher temperature than when pipes are actually imbedded in the floor. Pipes may be 1½ or 2 in. in the former, but for the latter ¾ or 1 in. pipes are recommended. See Fig. 4. Where the heat losses from a room are exceptionally high it may be necessary to supplement the warm floor by either adding some coils in the ceiling or forming heated panels in the side walls.

4. *By attaching separate heated metal plates or panels to the interior surfaces.* These plates or panels are placed either in an insulated recess so that the surface of the panel is flush with the surface of the walls or ceilings, or they may be secured to the face of the wall. They may be covered with wood veneers and decorated to harmonize with other

parts of the room, or they may be cast into panels to imitate oak or other wood designs. With flat plate panels it is common practice to use a frame of plaster, wood, metal or composition to allow for expansion. These plates may be heated with either hot water or steam and connected as in an ordinary radiator system. See Figs. 5 and 6.

5. *By electric heated metal plates or panels.* These plates or panels are either placed in insulated recesses of walls or ceilings or fastened to the construction, as found desirable. They should not have a surface temperature much above 200 F. Some have a much higher surface temperature but a lower temperature gives a more comfortable condition and is more efficient.

6. *By electrically heated tapestry mounted on screens and on the wall.* For this purpose the screen is woven with an electric continuous conductor. Such screens are useful to plug in at any position for emergency local heating without taking care of a large room or office.

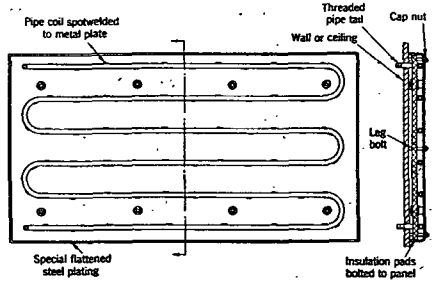


FIG. 5. PILLAR TYPE RADIANT HEAT PANEL

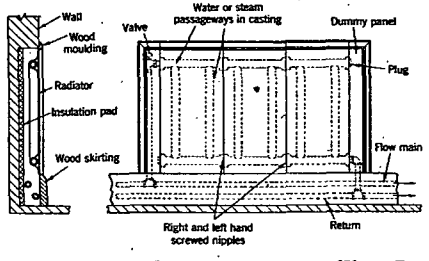


FIG. 6. FLAT TYPE PANEL INSTALLED IN WALL RECESS

Note. If all of a heating panel is installed at one end of a large room there may be a marked difference between the equivalent temperature on the two sides of the body. It is usually desirable, therefore, that the heat be distributed at different parts of the walls and ceilings so that no uncomfortable effect will be felt from unequal heating.

CALCULATION PRINCIPLES

Part I—Panel Heating

The term *panel heating*, involving both radiation and convection, is applied in this chapter to a system in which the heat is transmitted from panel surfaces to both air and surrounding surfaces, as is the case under indoor conditions.

Panel heating systems for buildings may be designed as illustrated and described in the following design of a panel heating system for the room shown in Fig. 7. The design is based on continuous heating. Panel

heating systems should, in general, be operated continuously since the panels have large thermal capacities. Where panels are heated by high temperature radiation from a heat source directed toward them, this qualification does not apply.

1. *Assume the location and the approximate size of the heating panel.*

Heating panels may be located in ceilings or floors or walls. Ceiling panels have the advantage that their heat emission is not affected by tapestry or furniture and that they can be used, to a limited extent, as cooling panels during the summer months. If used in low rooms, however, they may produce an undesirable heating effect upon the head. Floor panels have the advantage that they can be easily installed and that much of the radiated heat is delivered to the lower portions of the walls; they have the disadvantage that their heat emission is rather uncertain, since it may be affected by rugs, carpets, furniture, machinery, etc.

It is best to make the panels as large as practicable; for example, if the floor or ceiling is used as the heating panel, it is best to use the entire floor or the entire ceiling, or both. Heating a room by means of panels is very similar to lighting a room. Heat radiation is exactly like light radiation, except that it has a longer wave length. If a room is lighted

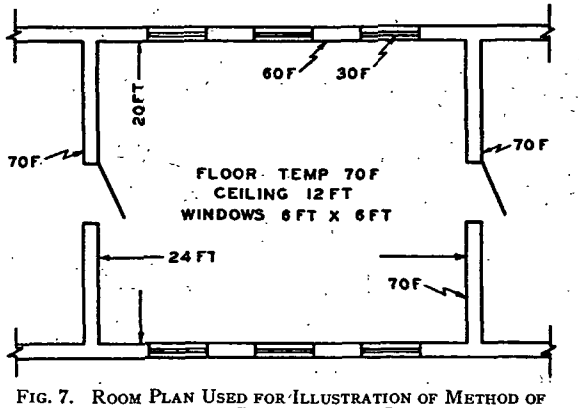


FIG. 7. ROOM PLAN USED FOR ILLUSTRATION OF METHOD OF DESIGNING A PANEL HEATING SYSTEM

by means of a large number of small units distributed uniformly over the ceiling, the room is lighted more uniformly than if it is lighted by means of a single unit of equal capacity. Similarly, if the entire ceiling is the heating panel, the room is heated more uniformly than if only a fractional part of the ceiling is used as the heating panel.

In the following example, *the entire ceiling will be used as the heating panel.*

2. *Select the desired mean temperature of the air in the room.*

In a panel-heated room, the mean air temperature is a few degrees lower than the mean temperature of the surfaces of the enclosing walls, floor, and ceiling. In a room heated by introduction of warm air or by means of radiators, convectors, or other similar heating appliances, located within the room, the mean temperature of the air is a few degrees higher than the mean temperature of the surfaces of the enclosing walls, floor, and ceiling. Under ordinary conditions, in a panel-heated room, the mean temperature of the air ranges from approximately 65 F to 72 F.

In the following example, *68 F is selected as the mean air temperature.*

3. *Determine as accurately as practicable, the mean temperature of the inside surfaces of the enclosing walls, floor, and ceiling.*

The two inside walls are assumed to separate rooms, which are filled with 68 F air, so that both surfaces of each wall are in close contact with 68 F air. The surface tem-

peratures of these walls, therefore, can not be lower than 68 F and must actually be higher than 68 F because, in addition to their contact with 68 F air and their contact with the heated ceiling, they are exposed to the heat radiation from the ceiling.

In the following example, *70 F will be selected as the mean surface temperature of the inside walls.*

For the two outside walls, the mean inside surface temperature can be calculated with fair accuracy. For a heat transmission coefficient of 0.25 and a temperature difference of 68 deg, heat flows through the wall at the rate of 17 Btuh per square foot. If the indoor film coefficient is 1.65, the temperature difference, indoor air to inside wall surface, is 17/1.65 or 10 deg and the wall surface temperature is 68 − 10, or 58 F. This value may be taken directly from Fig. 8. However, the film coefficient 1.65 was deter-

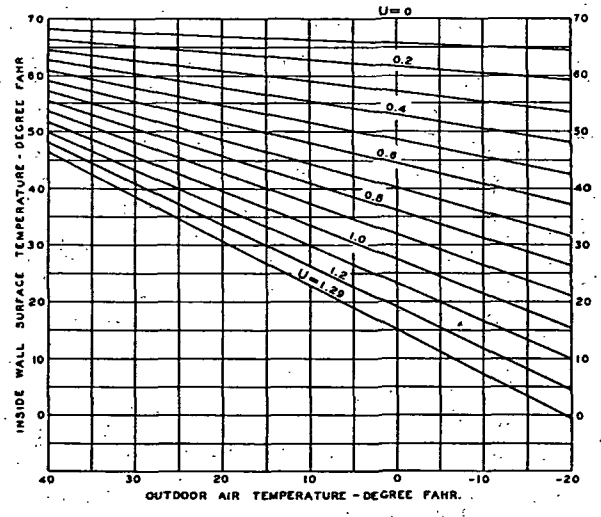


FIG. 8. CHART FOR ESTIMATING INSIDE SURFACE TEMPERATURES OF OUTSIDE WALLS[a]

[a]NOTE: The value of U, the over-all coefficient of heat transmission, cannot exceed 1.29 if the inside and outside film coefficients are 1.65 and 6.0 respectively (*e.g.* $\frac{1}{U} = \frac{1}{1.65} + \frac{1}{6} = \frac{1}{0.773}$. Therefore U = 1.29 maximum).

mined to represent the sum of the heat flow into the wall, by conduction from the air in contact with the wall, and by radiation from the warmer surfaces *seen* by the wall surface. In a panel-heated room, the rate of heat flow into the outside wall by radiation is greater than it is in a radiator-heated room; consequently, the film coefficient is higher, and the temperature difference, air to wall surface, is smaller, and therefore, the wall surface temperature is higher than the calculated 58 F. It is impossible to determine accurately how much higher than 58 F the temperature of the wall surface will be until the corresponding indoor air film coefficient has been determined accurately.

In the following example, *60 F will be selected as the probable mean inside surface temperature of the outside walls.*

The probable mean inside surface temperatures of the floor and the glass may be determined by calculations and by reasoning similar to that employed to determine the inside surface temperature of the outside walls.

In the following example, *30 F and 70 F will be selected as the probable inside surface temperatures of the glass and floor,* respectively.

TABLE 1. CALCULATED HEAT LOSS OF ROOM

SURFACE	AREA SQ FT	U	CALCULATION	HEAT LOSS BTUH
Outside Walls	360	0.25	360 x 0.25 x 68	6,120
Glass	216	1.13	216 x 1.13 x 68	16,597
Inside Walls	480	.....	No heat loss	.....
Ceiling	480	.....	Heating Panel	.....
Floor	480	0.10	480 x 0.10 x 38	1,824
Infiltration	.....	.....	5,760 cu ft x 1.50 x 68 x 0.018	10,576
			Total	35,117

4. *Determine the heat loss of the room.*

In the following example, the heat loss calculation will be based on an outdoor air temperature of 0 F. Since the functioning of a panel-heating system differs very little from that of a radiator-type heating system, the heat loss may be calculated according to Chapter 14 as shown in Table 1.

The heat loss through the outside walls and through the glass is probably a little greater than calculated because the calculation is based on an indoor air film coefficient of 1.65 Btuh, whereas, for a panel-heated room, this coefficient is a little higher, but the difference is probably not sufficiently large to be considered in design calculations for a heating system.

5. *Estimate the Mean Radiant Temperature.*

The Mean Radiant Temperature of the surfaces enclosing the room but not including the heating panels may be estimated as follows:

SURFACE	AREA	FAHR DEG	PRODUCT
Interior Walls	480	70	33,600
Exterior Walls	360	60	21,600
Glass	216	30	6,480
Floor	480	70	33,600
	1,536		95,280

The sum of these products divided by the sum of the surface areas is: 95,280/1,536 or 62.03 F, the required mean surface temperature.

In the following example, *62 F will be selected as the MRT of walls, glass and floors.*

6. *Determine the temperature of the ceiling panel.*

Determine the temperature of the ceiling so that the ceiling panel will deliver heat to the room at a rate equal to the rate at which the room is calculated to lose heat, namely, 35,117 Btuh.

When a room is heated by means of a panel, air convection currents are developed in the room similar to those which are developed when the room is heated by means of a free-standing radiator. Consequently, the heating panel delivers heat to the room partly by radiation and partly by convection. The proportion of the total heat flow delivered by convection varies with the location of the heating panel, with the height of the ceiling, and with the size, number, and location of pieces of furniture and other articles which interfere with the free flow of air along the floor and along the walls. It is generally sufficiently accurate to assume that a ceiling panel will deliver 70 per cent of its heat by radiation and 30 per cent by convection; a floor panel 55 per cent by radiation and 45 per cent by convection; and a wall panel 65 per cent by radiation and 35 per cent by convection.

In the following example, *it will be assumed that the ceiling panel must deliver 70 per cent of its heat or 24,582 Btuh by radiation,* since the total calculated heat loss is 35,117.

When two plane surfaces of infinite size are parallel to each other and their surfaces are at different temperatures, the exchange of heat between the two is proportional to the difference between the fourth powers of their absolute temperatures. This is also true when one surface is completely surrounded by another surface; for example, if one sphere is placed within another sphere, the flow of heat between the outer surface of the smaller sphere and the inner surface of the larger sphere is proportional to the fourth power of the absolute temperatures of the two surfaces.

In a panel-heated room, the heated panel may be considered to be completely enclosed by the remaining surfaces, because all heat radiated by the heated panel is intercepted by those surfaces. Consequently, the flow of heat from the heated ceiling to the room, by radiation, is proportional to the difference between the fourth powers of the absolute temperature of the ceiling and the absolute mean radiant temperature of the remaining surfaces.

The rate at which a surface emits heat varies with the temperature of the surface and with other characteristics of the surface. For ordinary heat flow calculations it is

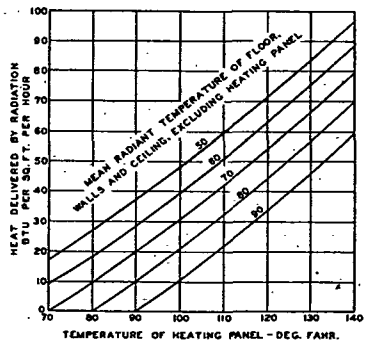


FIG. 9. HEAT DELIVERED TO ROOM BY RADIATION FROM PANEL

sufficiently accurate to assume that the materials which are commonly used in building construction emit heat at a rate of:

$$0.156 \left(\frac{T}{100}\right)^4 \text{ Btuh per square foot}$$

where

T is the absolute temperature of the surface in Fahrenheit degrees.

On this basis the flow of heat from the ceiling to its surrounding surfaces is at the rate of:

$$480 \times 0.156 \left[\left(\frac{T}{100}\right)^4 - \left(\frac{522}{100}\right)^4\right] \text{ Btuh.}$$

In order that this rate may be equal to 24,582 Btuh, T must be 572 and the ceiling temperature about 112 F.

Instead of calculating this temperature it may be taken from Fig. 9, as follows: The ceiling must deliver heat to the room, by radiation, at the rate of 24,582/480 or 51 Btuh per square foot. Find 51 on the left margin and move horizontally to the intersection with a 62 MRT line, and from the point of intersection to the lower margin and read about 112 F.

With a ceiling temperature of 112 F, the MRT of the room will be 480 × 112 + 1,536 × 62; the sum divided by 2,016, or 74 F.

If an air temperature of 68 F and an MRT of 74 F should not produce satisfactory conditions, the ceiling temperature can easily be changed as necessary by changing the temperature of the circulating water.

Calculations like the preceding may also be made with the aid of Table 2. The rate at which the ceiling must radiate heat exceeds by 51 Btuh per square foot the rate at

TABLE 2. TOTAL HEAT EMISSION BY RADIATION[a]

Body or Mean Radiant Temperature F Deg	Radiation in Btu per (square foot) (hour) emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature F Deg	Radiation in Btu per (square foot) (hour) emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1.00	e 0.95	e 0.90	e 0.80		e 1.00	e 0.95	e 0.90	e 0.80
30	99.7	94.7	89.8	79.7	71	137.0	130.1	123.3	109.7
35	103.9	98.7	93.6	83.1	72	137.9	131.0	124.0	110.3
40	108.0	102.8	97.2	86.4	73	138.9	132.0	124.9	111.0
45	112.5	106.9	101.3	89.0	74	140.2	133.1	126.1	112.1
46	113.3	107.7	102.0	90.8	75	141.6	134.4	127.4	113.2
47	114.3	108.6	102.9	91.5	80	147.2	140.0	132.5	117.9
48	115.2	109.5	103.8	92.3	85	152.9	145.2	137.6	122.4
49	116.0	110.3	104.5	92.8	90	158.5	150.5	142.7	126.9
50	116.9	111.0	105.3	93.6	100	170.3	161.7	153.2	136.2
51	117.9	112.0	106.4	94.4	110	182.3	173.2	164.2	146.0
52	118.8	112.9	106.9	95.1	120	195.6	185.7	176.1	156.5
53	119.8	113.8	107.8	95.9	130	210.9	200.4	189.8	168.8
54	120.6	114.6	108.6	96.6	140	224.1	212.9	201.8	179.2
55	121.7	115.5	109.4	97.3	150	238.0	226.1	214.4	190.5
56	122.6	116.4	110.3	98.1	160	252.1	239.8	226.9	201.8
57	123.5	117.4	111.3	98.9	170	271.6	258.0	244.5	217.2
58	124.4	118.2	112.0	99.6	180	289.1	274.9	260.1	231.3
59	125.3	119.0	112.8	100.3	190	307.7	292.1	276.9	246.1
60	126.3	119.9	113.8	101.1	200	326.5	310.2	293.9	261.3
61	127.1	120.7	114.4	101.8	210	349.3	331.9	314.3	279.5
62	128.2	121.8	115.3	102.6	220	373.0	354.4	335.7	298.2
63	129.1	122.6	116.2	103.3	250	439.5	417.6	395.6	351.6
64	130.1	123.5	117.1	104.1	300	577.3	548.2	519.6	461.8
65	131.0	124.4	117.9	104.8	350	743.0	705.8	668.6	594.3
66	132.1	125.5	118.8	105.8	400	945.8	898.5	850.8	756.5
67	133.0	126.4	119.7	106.4	450	1181.0	1121.0	1063.0	944.0
68	134.0	127.3	120.5	107.2	500	1470.0	1396.0	1323.0	1176.0
69	135.0	128.3	121.5	108.0	550	1798.0	1708.0	1619.0	1439.0
70	136.0	129.3	122.3	108.8	600	2181.0	2072.0	1962.0	1745.0

[a] These factors are calculated from the formula $q_r = e\left(\frac{0.173 \times T^4}{100{,}000{,}000}\right)$ *where* q_r = total radiation, Btu per (sq ft) (hr), e = emissivity, T = absolute temperature, Fahrenheit degrees.

which the ceiling receives radiant heat from its surroundings. Assuming the emissivity of the walls, floor, and ceiling to be 90 per cent of that of a *black* body, the heat radiated to the ceiling, from the surfaces whose MRT is 62 F, is (Table 2) at the rate of 115.3 Btuh per square foot; the ceiling must therefore radiate heat at the rate of 115.3 plus 51, or 166.3; its temperature must be (Table 2) between 110 F and 120 F, and, by interpolation, 112 F, as calculated.

7. *Select the medium for heating the ceiling panel.*

The medium may be electricity, steam, air, or water, but usually is air or water. If air is used it is generally heated in the basement, passed up through hollow inside walls or through ducts in those walls, allowed to flow between the ceiling and the floor above, and returned to the basement through hollow outside walls or through ducts in those walls.

If the walls and floors are constructed of hollow tile, the cells in the tile can be placed so that they will form continuous ducts through which the warm air can flow up the inside walls, then between the ceiling and the floor above, and down the outside walls. In this way the walls and ceiling become heating panels.

If water is used as the medium, the pipes through which the water circulates—almost always under forced circulation—are placed in the floor, walls, or ceiling in such a manner that as much as possible of the heat emitted by the pipes will be delivered to the space to be heated.

TABLE 3. HIGHEST SAFE SURFACE TEMPERATURES FOR HEATING PANEL

TYPE OF PANEL	SURFACE TEMPERATURE F DEG
Plastered Ceiling (Pipes Imbedded)	115
Plastered Walls (Pipes Imbedded)	120
Floor, Any Method	90
Floor, Border and Aisles	120
Iron, Hot Water Medium[a]	160
Iron, Steam Vapor[a]	180
Electrically Heated Panels[a]	200

[a]Low surface temperature radiation is recommended regardless of the heating medium employed.

Practical limits for surface temperatures of heating panels are given in Table 3.

In this example water will be selected as the medium.

8. *Determine the size, length, and location of the pipe coils in the panels.*

When hot-water pipes are imbedded in concrete slabs or attached to plastered surfaces, their rate of heat emission varies with many factors. If the pipes are imbedded in dense concrete slabs, it may be assumed that the rate of heat emission of ½-in. pipe, spaced 6 in. on centers; ¾-in. pipe, spaced 9 in. on centers; and 1-in. pipe spaced 12 in. on centers; per foot of length of pipe and per degree difference between the temperature of the water in the pipe and that of the air in the space to be heated, is 0.8, 1.0, and 1.2 Btuh, respectively. If the distance between the pipes is increased, the rate of heat emission, per foot of pipe, is also increased; if the distance is doubled, the rate of heat emission is increased about 15 per cent. If the pipes are attached to plastered ceilings, the rate of heat emission is slightly less, probably about 10 per cent less, than when the pipes are imbedded in concrete slabs. The data given regarding heat emission of panels are intended as general guides for the designer. Additional experience and research are needed to develop definite and complete data. However, after a heating panel has been designed and installed, any small error can easily be corrected by modifying the temperature of the water circulating through the coils.

When the heating pipes are attached to a plastered ceiling, a portion of the heat emitted by the pipes is delivered to the space below the ceiling and a portion to the space above the ceiling. The relative quantities depend on the degree of insulation applied above the heating coils.

When the heating pipes are imbedded in a concrete floor slab a portion of the heat emitted by the pipes will flow upward into the space to be heated, and the remainder will flow downward into the ground.

When the heating pipes are placed below the concrete floor slab instead of being imbedded in the slab, a larger portion of the heat will flow into the ground, and a smaller portion into the space to be heated.

In the following example *it is assumed that the insulation above the pipe coils is such that 90 per cent of the heat emitted by the pipe coils will flow into the room and 10 per cent into the space above.*

Since the room is to receive 35,117 Btuh, and since the room is assumed to receive only 90 per cent of the heat emitted by the coils attached to the plastered ceiling, the coils must emit 35,117/0.9 or 39,000 Btuh. If ¾-in. pipe and a mean water temperature of 140 F are selected, the heat emitted, per foot of pipe, will be 0.9 (140 − 68) or 65 Btuh. The quantity of pipe required will therefore be 39,000/65 = 600 lineal feet.

The pipe coils can be arranged in any convenient manner, but should be arranged so that the temperature of the water in the pipe will vary only slightly; otherwise, the temperature distribution over the ceiling will not be uniform. Generally, it is best to arrange the pipes so as to form two-pipe reversed-return systems, as suggested by the two sketches in Fig. 10. By using 33 runs of ¾-in. pipe, welded to two 1¼ in. mains, sufficient pipe surface is secured; the ¾-in. pipes will then be spaced about 8½ in. on centers, which is satisfactory.

The friction heads of water flowing in pipes and fittings are so well known that the pipe coils can be designed so that each will receive its proper share of the hot water without the use of balancing valves. However, it is desirable to divide large heating systems into sections and to install valves so that individual sections can be disconnected without interfering with the operation of the system as a whole.

Part II—Radiant Heating

The term *radiant heating* is applied in this chapter to a system in which only the heat radiated from the panel is effective as in outdoor and semi-outdoor conditions.

The outstanding example of radiant heating is the transfer of heat from the sun to the earth. The sun radiates large quantities of energy of which a very small portion is intercepted by the earth. A part of the intercepted radiation is transformed into heat when it strikes the earth's

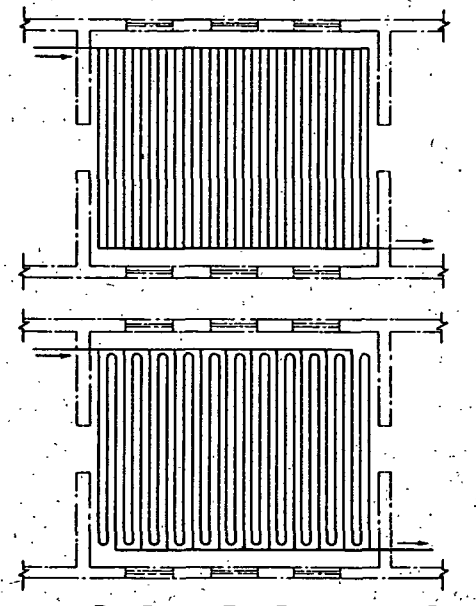


FIG. 10. ARRANGEMENT OF PIPE COIL FOR EVEN DISTRIBUTION OF PANEL TEMPERATURE

surface. In this manner *heat* is received by the earth from the sun by *radiation*.

In industry, radiant heating is employed in manufacturing processes, particularly in drying, baking, and dehydrating operations; in agriculture, it is employed to improve living and growing conditions for young plants and young animals.

The heating engineer employs radiant heat primarily in the heating of open-air schools and open-air hospitals. When a surface radiates heat, and every surface does unless its temperature is absolute zero, every point of the surface radiates heat in all directions. The total quantity of heat radiated by a point or by an elementary area is π times the quantity of heat radiated at right angles to the surface.

Thus, if in an elementary cube the upper face is the heating panel, the lower face would receive only about 32 per cent of the radiated energy and the four sides would receive each about 17 per cent.

For larger surfaces the conditions are different. If two parallel plane surfaces of considerable size are near each other, the rate of heat exchange between the two can be determined fairly accurately by means of the chart of Fig. 9. This is possible because the larger part of the heat radiated by one of the surfaces is intercepted by the other surface and only a small portion is radiated in such directions that it will not impinge upon the opposite surface.

As the distance between the two surfaces is increased, the proportion of the heat radiated by one of the parallel plane surfaces and intercepted by the other decreases almost as the square of the distance between the surfaces increases, because the intensity of heat radiation, like the intensity of light radiation, varies inversely as the square of the distance from the source of radiation.

For the purpose of designing radiant heating systems in which the heating panel is practically square and is radiating heat toward a parallel surface of equal size and shape, as shown in Fig. 11, the rate of heat exchange between the two surfaces will be equal to that shown in Fig. 9, multiplied by a factor, p, which depends upon the ratio of h to s (Fig. 11) as shown in the following table:

h/s =	1	2	3	4	5
p =	0.200	0.070	0.034	0.020	0.013

For example, if a panel 3 ft square is located parallel to, and 9 ft above, a bed in an open-air hospital, and if the temperature of the panel is 112 F and that of the bed is 70 F, the rate of heat transfer from the panel to a 9 sq ft section of the bed directly beneath the panel, will be 3.4 per cent of the rate shown in Fig. 9, or $0.034 \times 9 \times 44$, or 14 Btuh, approximately. The rate of heat transfer from the panel to a section of the bed other than the 9 sq ft directly beneath the panel will be lower than 14/9 Btuh per square foot.

This is a crude way of designing a radiant heating system for an open-air hospital, but it is sufficiently accurate, because the required temperature of the bed and the required rate of heat flow into it will vary with the temperature of the outdoor air, with the air movement over the bed, with the thickness and the character of the bedding, and with the physical condition of the patient.

A radiant heating system for an open-air school may be designed as described for the open-air hospital. The heating panel in such a case should be almost as large as the ceiling and, in order to keep the heat loss by radiation at a minimum, should be placed so that a maximum portion of the heat radiated by the panel will be directed toward the pupils and a minimum toward the outside walls and particularly the windows.

MEASUREMENT OF RADIANT HEATING

Radiant heating is intended to control the rate of radiant heat loss from the human body and should be measured by calorimetric methods.

The apparatus for this purpose consists essentially of a cylinder, maintained at the accepted mean surface temperature of the human body, together with an accurate (usually electrical) measuring of the varying rate of heat supply required to maintain this exact temperature. This instrument, the *eupatheoscope*, is readily adapted to function like a thermostat so as to turn heat on or off, when the desired temperature of 80 F, or any other predetermined surface temperature of the cylinder,

decreases or increases as a result of changes in the Operative Temperature.

For testing work, the *globe thermometer* is a useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 to 9 in. in diameter, usually made of thin copper and painted black and sometimes covered with cloth. The temperature recorded by thermometer with its bulb in the center of the sphere is termed the *radiation-convection temperature.* See Chapter 11.

CONTROL OF PANEL AND RADIANT HEATING

The effectiveness of any type of control will depend largely on the time lag of the system. With warm air passing through floor ducts the time lag is usually too long for any kind of room thermostat, in fact a thermostat will not prove suitable with any system if the building is constructed with massive brickwork and masonry, unless it operates in conjunction with a time control responsive to changes in outside conditions.

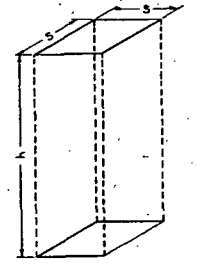


FIG. 11. EFFECT OF HEIGHT UPON RADIATION RECEIVED FROM A PANEL

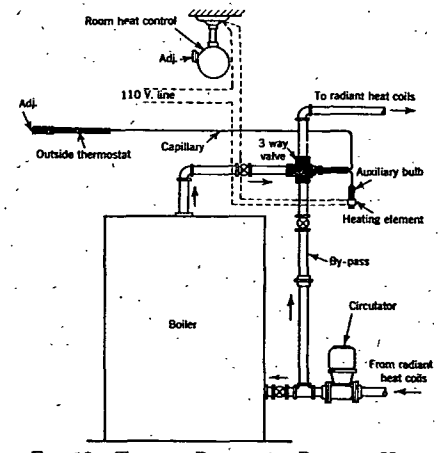


FIG. 12. TYPICAL PANEL AND RADIANT HEAT CONTROL SYSTEM

The heat emitted by hot water pipes imbedded in the plaster of the ceiling and walls or in the concrete base of a floor can be effectively controlled by an instrument designed to modulate the temperature of the water circulating in the system according to the outside conditions. Metal panels which can be installed in the ceiling or side walls may be either controlled by an instrument responsive to outside weather conditions or by a specially designed instrument responsive to both air temperature and radiation. Any purely *on or off* control system is not recommended for panel heating.

A typical control system operated from an outside thermostat and supplemented with a room heat control instrument is illustrated in Fig. 12. The outside thermostat modulates the temperature of the circulating water in the coils by mixing some of the hot water leaving the boiler with a proportionate amount of return water which is diverted to the three-way valve.

One type of room instrument consists of a blackened copper sphere of 6 or 8 in. in diameter, in which a cylindrical sump contains a volatile liquid. A small electric heating coil creates in the sphere a vapor pressure which remains constant as long as the total heat loss from the sphere is at the desired rate. If the Operative Temperature becomes too high for comfort, a greater vapor pressure results from the smaller heat loss from the sphere. This acts on a diaphragm and reduces the supply of heat to the room. With too low an Operative Temperature the reverse action occurs. A similar instrument which has an electric heating element for warming the air inside the sphere and the thermostat operated switch is also used for controlling room conditions.

In addition to a thermostatically controlled device for modulating the temperature of the circulating water, it is advantageous to insert in each coil a locked flow control or adjustable resistance to give uniform conditions throughout all rooms. Owing to unforeseen difficulties with varying frictional losses in pipes, emission factor, and exposures, it is an advantage to be able to regulate permanently the flow through each circuit by means of a key operated valve as indicated in Fig. 4.

REFERENCES

[1]—Standard Operative Temperature, A Generalized Temperature Scale, Applicable to Direct and Partitional Calorimetry, by A. P. Gagge (*American Journal Physiology*, 1940, Vol. CXXXI, p. 93).

[2]—Physiological Reactions and Sensations of Pleasantness Under Varying Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge (A.S.H.V.E. TRANSACTIONS, 1938, Vol. 44, p. 190).

[3]—The Influence of Air Movement Upon Heat Losses from the Clothed Human Body, by C.-E. A. Winslow, A. P. Gagge and L. P. Herrington (*American Journal Physiology*, 1939, Vol. CXXVII, p. 505).

[4]—Surface Heat Transmission, by R. H. Heilman (*A.S.M.E. Transactions, Fuels and Steam Power Section*, Vol. 51, No. 22, September-December, 1929).

BIBLIOGRAPHY

Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 425).

A.S.H.V.E. RESEARCH REPORT No. 1192—Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 35).

A.S.H.V.E. RESEARCH REPORT No. 1193—Radiation as a Factor in the Feeling of Warmth in Convection, Radiator and Panel Heated Rooms, by F. C. Houghten, Carl Gutberlet and E. C. Hach (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 55).

Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 285).

Operating Results of a Residence Radiant Wall Heating System, by E. J. Rodee (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 123).

Performance of a Residential Panel Heating System, by H. F. Randolph and J. B. Wallace (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 235).

Radiant Heating (*Heating and Ventilating*, March, 1941, p. 35).

Radiant Heating and Cooling, by F. E. Giesecke (*Heating, Piping and Air Conditioning*, June, July, August, September and October, 1940).

Calculations for Radiant Heating, by T. Napier Adlam (*Heating and Ventilating*, October, 1931).

Radiant Heating and Cooling, Part I, by C. O. Mackey, L. T. Wright, Jr., R. E. Clark, and N. R. Gay (*Cornell University Engineering Experiment Station*, Bulletin No. 32, 1943).

CHAPTER 32

Fans

Classification, Performance, Fan Efficiency, Characteristic Curves, System Characteristics, Selection of Fans, Fan Designations, Control, Motive Power

IN heating, ventilating, and air conditioning practice, fans and blowers are used to produce air flow. All fans or blowers are classified according to the direction of air flow through the fan with relation to the axis of rotation and are either of the (1) *axial flow* or *propeller* type, in which the flow is parallel to the axis, or (2) *radial flow* or *centrifugal* type, in which the flow is at right angles to the axis.

Axial flow fans are made in various designs and sizes. The form and number of blades may vary for impellers of the same or different diameters. The blades may be of uniform thickness and made of cast or sheet metal, and either flat or cambered or of screw form; or they may vary in thickness, in the latter case usually being designed to conform to so-called *airfoil* sections of known characteristics, similar to those which have been developed for airplane propellers. Likewise, blade angle, or the angular relation of the blades to the plane of rotation, varies over a wide range. For operation against comparatively high pressures, it is customary to resort to enlarged hubs in proportion to fan diameter (large hub ratio) and correspondingly short blade length. The term *disc fan* has sometimes been loosely applied to such large hub fans, though it has long been generally used in connection with any propeller fan of comparatively short axial length whose blades are relatively flat; in other words, for fan wheels which occupy a space which is more or less disc-shaped. Single stage air foil axial flow fans can operate at pressures up to 4 in. at a quite low noise level and, at considerably higher pressures, they are comparable in regard to noise to the centrifugal fan type.

Radial flow or *centrifugal fans* used in heating, ventilating, and air conditioning practice, are in general of two types; one with forward-sloped blades, and the other with backward-sloped blades. Many modifications may be made in the proportions, the curvature, and the slope or angularity of the blades. The slope or angularity of the blades determines the operating characteristics of a fan; a forward-curved or sloped blade is found in a fan having low speed operating characteristics, while a backward-curved or sloped blade is found in a fan having higher speed operating characteristics.

A wide variety of heating, ventilating, and air conditioning systems creates a wide variety in the demands that have to be met by the fan or blower. The requirement may be to move small or large quantities of air against little or no resistance, or to move small or large quantities of air against higher resistance. Between the two limits innumerable specific requirements must be met. Although fans of any class in either of the two general types can in general be made to perform the same duty, mechanical difficulties, space and noise limitations, efficiency and power limitations usually determine the selection.

FAN PERFORMANCE

Fans of all types follow certain laws of performance 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 equip-

ment due to the space, power, or speed limitations. The following laws in which Q = air volume and P = static, velocity or total pressure, apply to all types of fans:

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 square of 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 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 the 200 F temperature, what will be the speed, capacity, and 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?

$$\text{Speed} = 400 \times \frac{0.075}{0.0602} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.075}{0.0602} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.075}{0.0602} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.075}{0.0602}\right)^2 = 6.20 \text{ hp.}$$

Laws of Homologous Fans

The laws applying to different sizes of homologous fans are as follows:

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 EFFICIENCY

The efficiency of a fan may be defined as the ratio of the horsepower output to the horsepower input.

The horsepower output is expressed by the formula:

$$\text{Air Horsepower}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356} \quad (1)$$

When the static pressure is used in the computation in place of total pressure it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but, owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure. In the standards for published capacity tables as adopted by the *National Association of Fan Manufacturers*, the term *static pressure* refers to the true resistance to air flow. Such tables charge both the inlet and outlet velocity of the fan to the fan performance, and may be used directly where the static pressure of the system as calculated represents only the actual resistance to flow of the air.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static Efficiency}^1 = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \quad (2)$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total pressure. This efficiency is variously known as the total, or mechanical efficiency, and may be expressed as follows:

$$\text{Mechanical or Total Efficiency}^1 = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \quad (3)$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different percentages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans based upon tests performed in accordance with the Standard Test Code for Centrifugal and Axial Fans [1] prepared jointly by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers* are generally plotted to show total and static pressure, mechanical and static efficiency, and horsepower in relation to air delivery as a basis. Results may also

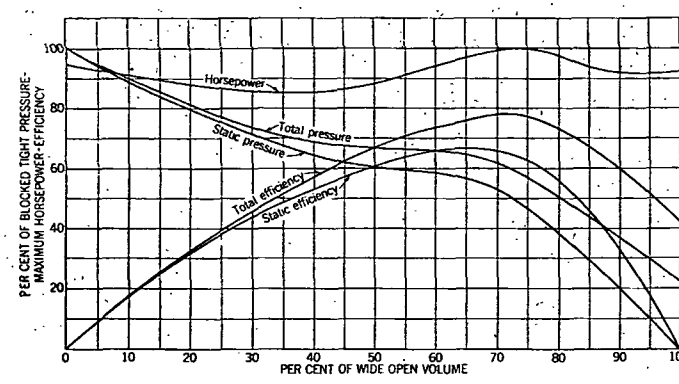


FIG. 1. OPERATING CHARACTERISTICS OF AXIAL FLOW AIRFOIL TYPE FAN

be plotted against per cent of wide open volume or discharge. Examples of fan performance curves are shown in Figs. 1, 2 and 3.

In the selection of all but very small fans, power consumption is usually a major consideration. It must be borne in mind that the horsepower at peak efficiency alone may be misleading, as actual operation is apt to occur at some point on the pressure-volume curve varying considerably from that specified, due to inaccuracies of the estimated system resistance or to fluctuating resistance caused by damper or louver adjustments. To cope with such variations a fan should be selected having a high efficiency over a wide range, that is, a *flat* or broad efficiency curve is more desirable than a sharp or narrow curve which, though reaching a high peak, falls off rapidly to either side of a narrow range. When the point of operation varies only within narrow limits and both volume and pressure requirements are accurately known in advance, the designer can select a fan operating at maximum efficiency, irrespective of performance over the entire range.

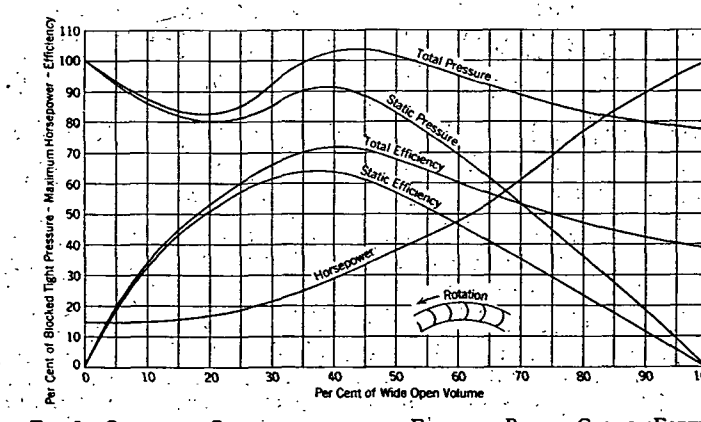


FIG. 2. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

Generally, fans are selected either at the peak of the static efficiency or to the right of the peak depending on the requirements of the particular installation. Fans selected to the right of the peak will be smaller but will require more power, run at higher speeds and may have a higher sound rating. Where first cost is important and added horsepower and noise are not important, smaller fans may be used. Where efficient and quiet operations are most important, fans are selected at or near the peak of the static efficiency curve. Fans are not ordinarily selected to the left of the peak of the static efficiency curve as this results in larger, more costly fans, requiring more power and in some cases producing objectionable noise.

The curves in Figs. 1, 2, and 3 show operating characteristics for axial flow and two classes of centrifugal fans, namely, forward-curved multiple blade and backward-inclined blade design, for comparison purposes. These curves are not applicable for rigid comparison or actual selection, but are shown to indicate variations in operating characteristics.

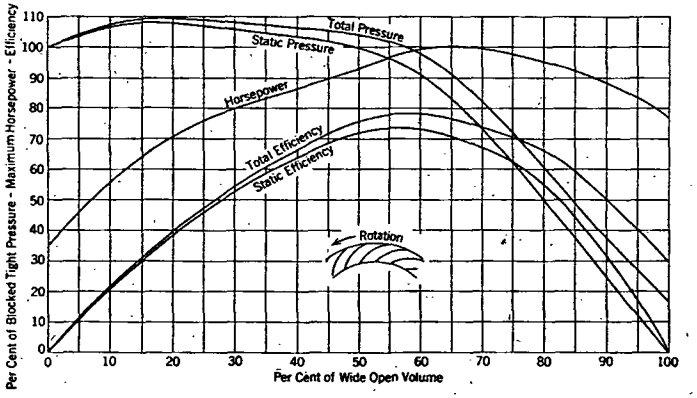


FIG. 3. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

The curves in Fig. 1 of a *typical axial flow* fan show characteristics of non-overloading horsepower and high efficiency. These results are obtained by producing a more uniform pressure throughout the blade annulus, so that back flow does not occur except at high pressures. This avoidance in turbulence has a tendency to reduce noise. Fans of this type are operating against static pressures as high as 4 in. water. The capacity and efficiency of axial flow fans when operating above the low pressure range can be improved by the use of either inlet or outlet guide vanes or both. The effect of such vanes is to increase the level of the pressure volume curve, and properly designed vanes on the discharge side of the fan have the advantage of eliminating the rotational component of the air stream, thus restoring uniform axial flow. As high pressures usually require large hubs in proportion to the fan diameter, performance is improved by the use of round-nosed or conical forms mounted coaxially with the direct-connected fan (sometimes partly or wholly enclosing the motor) so as to make the changes in velocity to and from the fan blade annulus as uniform as space conditions permit. When axial flow fans are installed in ducts, provisions may be made to install the driving motor outside the duct by employing slots in the duct to permit

a belt drive from the motor to the fan sheave, or by extending the shaft for a direct-connected motor placed outside of a Y fitting or elbow in the duct system.

The *forward-curved multiblade* fan and the *backward-curved* type are used extensively in heating, ventilating, and air conditioning work. The forward-curved type has a low peripheral speed and a large capacity. (See Fig. 2.) The point of maximum efficiency for this fan occurs near the point of maximum pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continuously from low to peak capacity and, if reasonable care is exercised in calculating resistance, a moderate reserve in power in the motor selection will prevent overloading.

The *backward-sloped type* includes the full backward-curved blade and the double-curved blade having a forward-curved heel and a backward-curved tip. This type has steep pressure curves, non-overloading power characteristics, and relatively high speed (see Fig. 3). This fan operates at a peripheral speed approximately 175 to 200 per cent of that of the forward-curved multiblade fan for like performance. Pressure curves for this type begin to drop at very low capacity, with the most rapid drop beginning at about 60 per cent of wide open volume. The steep portions of the pressure curves tend to produce nearly constant capacity under changing pressures. Where wide fluctuations in demand occur, especially where the regulation is obtained by damper control and particularly through by-passes, this type of fan is desirable to prevent overloading of motor. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type makes it adaptable for direct connected electric motor drives. The dimensional bulk is usually greater than that of the forward-curved multiblade type.

Between the extremes of the forward and backward curved blade type centrifugal fans there exists a number of modified designs differing in angularity and in the shape of the blades. Characteristic curves of these types show varying degrees of similarity to the curves in Figs. 2 and 3.

SYSTEM CHARACTERISTICS

Any ventilating system consisting of duct work, 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. 4. 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 a constant speed fan curve for a given size fan is super-imposed 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. 4, 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.

The curves in Fig. 4 also illustrate the effect of errors which 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. 4. If a 100 per cent 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 over estimated and

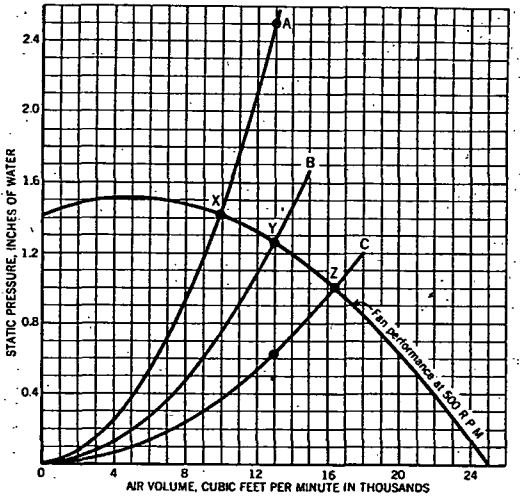


FIG. 4. PARABOLIC SYSTEM CHARACTERISTIC CURVES

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 per cent in volume; and in the second example, a system estimated at twice what it should have been resulted in an increase of 26 per cent 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. 4, 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.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures.

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 per cent 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 of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Other important factors to be considered in selecting fans are: (1) efficiency, (2) space occupied, (3) sound emission, (4) first cost, and (5) speed (both peripheral and revolutions per minute). These factors are not necessarily shown in the order of importance. In some installations space occupied may be of first importance. In others lowest power consumption is desirable. In many cases quietness of operation of the entire system is essential. Practically all fans operate at their lowest sound level when selected at or near the peak of the static efficiency so that in selecting a fan for highest static efficiency the quietest operating range of the fan will also be obtained. Table 1 shows desirable outlet velocities and tip speeds, or peripheral velocities, for various static pressures. Fans selected accordingly will operate at or near the peak of

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR MULTIBLADE VENTILATING FANS

Static Pressure Inches of Water	Forward Curved Blade Fans		Backward Tipped and Double Curved Blade Fans	
	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
¼	1000–1100	1520–1700	800–1100	2600–3100
⅜	1000–1100	1760–1900	800–1150	3000–3500
½	1000–1200	1970–2150	900–1300	3400–4000
⅝	1200–1400	2225–2450	1000–1500	3800–4500
¾	1300–1500	2480–2700	1100–1650	4200–5000
⅞	1400–1700	2660–2910	1200–1750	4500–5300
1	1500–1800	2820–3120	1200–1900	4800–5750
1¼	1600–1900	3162–3450	1300–2100	5300–6350
1½	1800–2100	3480–3810	1400–2300	5750–6950
1¾	1900–2200	3760–4205	1500–2500	6200–7550
2	2000–2400	4000–4500	1600–2700	6650–8050
2¼	2200–2600	4250–4740	1700–2800	7050–8550
2½	2300–2600	4475–4970	1800–2950	7450–9000
3	2500–2800	4900–5365	2000–3200	8200–9850

the static efficiency with resulting low power consumption and noise levels. Smaller fans with higher outlet velocities may be used if the installation requirements are such as to warrant the additional power and increased sound level. When space for duct expansion from a fan outlet is not available there may be advantages in selecting a larger fan for reducing duct noises, although lower outlet velocities generally result in lower fan efficiencies which cannot always be justified on the basis of increased cost and space requirements. Fans for schools, churches, residences, and all public buildings should be selected for lower outlet velocities and tip speeds than would be required for other types of installations.

Having selected a fan for its quietest operating point consistent with the requirements of the installation, it must be recognized that ventilating fans, even so selected, emit noise and precautions must be taken in the installation of the fans to prevent this noise from being transmitted to occupied portions of the building. Fans operating against high static pressures produce more noise than fans operating against low static pressures. Consequently, from a noise standpoint, the system should be designed to operate against the lowest static pressure possible. In many modern air conditioning systems it is necessary to introduce devices into the air stream for conditioning the air in various ways, the result of which is to set up a rather high static pressure against which the fan must operate. In such cases the sound level at the fan may be too high to be neglected and special sound treatment of the installation must be considered. When a fan is operating against higher pressures it should be located in a room either removed from the occupied areas, or in a room which has been acoustically treated to prevent sound being carried through the walls to adjoining spaces. The fan should be mounted on a resilient base along with its driving motor to absorb any noise or vibration which might be transmitted to the floor and thence to the building structure. All ducts should be connected to fans with unpainted canvas, or other flexible material, to prevent any vibrations being transmitted to the duct work. Ducts leading into the fan room or from the fan should be acoustically treated on the interior and in special cases should be provided with sound traps or filters. Many ventilating systems encounter noises which are connected with the fan in no way. Noises due to high duct velocities, abrupt turns, grilles, etc., may be present. Treatment of such problems is covered in Chapter 42.

If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. Proper installation and location of fans is as important as the selection of the size of fan for that particular installation. All fans will work according to their characteristic curves if they are properly selected and properly installed and operated.

FAN DESIGNATIONS

In order to prevent misunderstandings, which may cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 5 are suggested.

Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is clockwise, the fan, blower, or blast wheel will be designated as *clockwise*. If the proper direction of rotation is counter-clockwise, the designation will be *counter-*

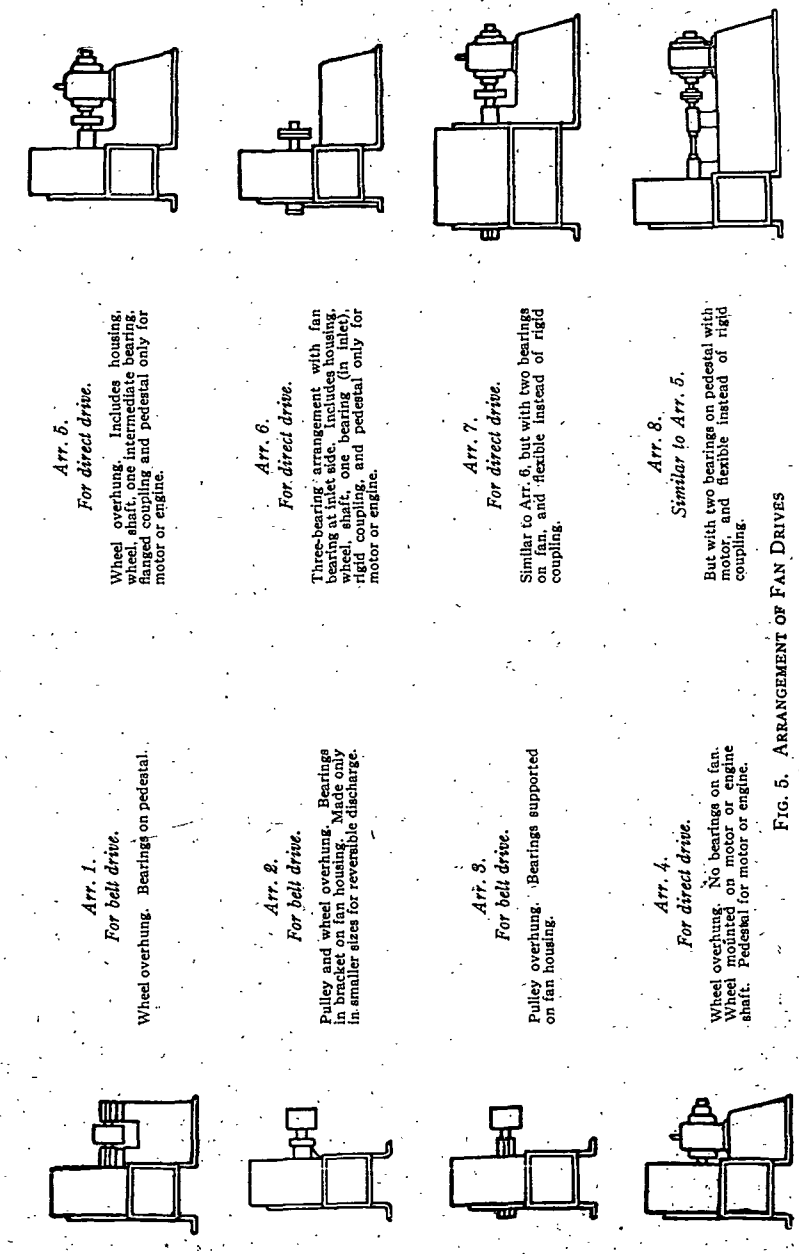


FIG. 5. ARRANGEMENT OF FAN DRIVES

clockwise. (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.)[2]

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word *hand* but specify *clockwise* or *counter-clockwise.*

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of air discharge is vertically up.

Down blast: If the line of air discharge is vertically down.

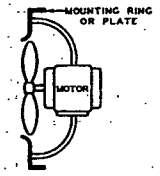
All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

Fig. 6 shows the names and definitions of types of fans recently distributed by the *National Association of Fan Manufacturers*[3].

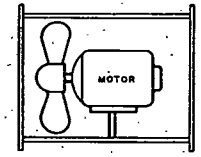
FAN CONTROL

In some heating and ventilating systems it is desirable to vary the volume of air handled by the fan, and this may be accomplished by a number of methods. Where the change is made infrequently, the pulley



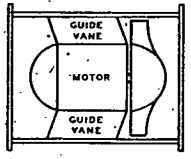
Propeller Fan

A propeller fan consists of a propeller or disc type wheel within a mounting ring or plate and including driving mechanism supports either for belt drive or direct connection.



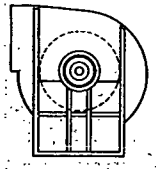
Tubeaxial Fan

A tubeaxial fan consists of a propeller or disc type wheel within a cylinder and including driving mechanism supports either for belt drive or direct connection.



Vaneaxial Fan

A vaneaxial fan consists of a disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports either for belt drive or direct connection.



Centrifugal Fan

A centrifugal fan consists of a fan rotor or wheel within a scroll type of housing and including driving mechanism supports either for belt drive or direct connection.

FIG. 6. NAMES AND DEFINITIONS OF TYPES OF FANS

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, electric or hydraulic couplings, may be used to vary the fan speed. Variable speed motors and variable fan inlet vanes may also be used to adjust the fan volume. All of these methods will give control. From a power consumption standpoint, a reduction of the fan speed is most efficient. Inlet vanes save some power and dampers save the least. From the standpoint 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 of each case will dictate which type of control is most desirable. Where noise is a factor, lowering the fan speed if possible is preferred as a control means, because of the resulting reduction in sound level.

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 by-pass 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.

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 accurate, 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. 4 shows that the system resistance varies as the square of the volume and 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. 4 indicates that there is no justification for allowing large spare motor capacity, and it is generally more economical to operate motors well loaded.

REFERENCES

1—See Standard Test Code for Centrifugal and Axial Fans, 3rd Edition, 1938, Bulletin No. 103, *National Association of Fan Manufacturers.*

2—Recommendations adopted by the *National Association of Fan Manufacturers.*

3—Supplement No. B to Form X-12, *National Association of Fan Manufacturers.*

BIBLIOGRAPHY

Mine Ventilation, by J. J. Walsh (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 659).

Fan Blower Design, by H. F. Hagen (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 175).

The Specific Characteristics of Fans, by M. C. Stuart and J. B. Lusk (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 57).

Non-Dimensional Fan Characteristics, by H. Carlton Moore (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, September, 1937, p. 580).

Fans, by Theodore Baumeister, Jr. (McGraw-Hill).

Fan Engineering, Buffalo Forge Co.

Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932 (John Wiley & Sons).

The Centrifugal Fan, by Frank L. Busey (A.S.H.V.E. TRANSACTIONS, Vol. 21, 1915, p. 43).

The Theory and Performance of Axial-Flow Fans, by L. S. Marks and J. R. Weske.

The Axial Flow Fan and Its Place in Ventilation, by W. R. Heath and A. E. Criqui (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944).

The Centrifugal Pumps and Blowers, by A. H. Church (John Wiley & Sons).

CHAPTER 33

Air Cleaning Devices

Air Cleaners, Dust and Lint, Classification of Air Cleaning Media, Viscous Impingement Type Cleaners, Automatic Viscous Filters, Dry Air Filters, Electric Precipitators, Performance and Testing, Selection and Maintenance, Safety Requirements, Adsorption of Vapors

FOR the purposes of this discussion, an air cleaner is defined as a device for capturing and removing solid matter from a stream of air. This solid matter includes fibrous material such as lint as well as particulate matter such as dust, fumes, smoke, cinders, etc. Air cleaning is distinct from air purification in that the latter deals with removal of harmful or unpleasant gases, vapors, or bacteria from an occupied space. Air pollution and purification are discussed in Chapters 10 and 12 and air sterilization in Chapter 13.

In general, air cleaners are not installed in buildings for the specific purpose of improving the health of the occupants. However, the removal of matter such as pollen, house dust and similar allergens which motivate attacks upon persons of allergic sensitivity[1], does have a beneficial effect on health. The toxic elements in some war gases are in reality fine particles capable of air flotation.

It is known by experience that air cleaners greatly reduce the rate of dirt accumulation in buildings, but, in the present state of knowledge, an exact numerical expression for the cleaning effect of the filter cannot be given because of the many unknown or variable factors involved. For this reason air cleaners are usually selected on a basis of experience or judgment, guided by the results of various test procedures.

AIR CLEANERS

A typical air cleaner consists of a frame, which may be of metal, wood, cardboard, etc., and a filtering medium. The frame is designed to support the medium in a duct or chamber forming part of the air conditioning or ventilating system, so that the air passes through the medium while en route to spaces to be ventilated.

Unit Air Cleaners or Filters

Many air cleaners are available in the form of units of convenient size for handling during installation, cleaning or replacement. 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, often, in the recirculating 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.

Settling chambers, air washers and electrostatic precipitators are also

air cleaners. Settling chambers are used in boiler plants for capturing cinders, but they occupy too much space for general use in air conditioning or ventilating systems. Unless they are inordinately large, they are not effective in capturing small particles, since the air velocity is not sufficiently reduced to permit such particles to settle.

Air washers have generally become recognized for the purpose of adjusting the temperature and humidity of the air, rather than for cleaning air. It happens that insofar as dirtiness is concerned, carbon is the most troublesome dust. Carbon particles are likely to be greasy and since there is a natural repulsion between grease and water, the water spray in an air washer is not effective to a desirable degree in capturing them. However, air washers do capture considerable amounts of dust and lint which become sludge in the sump. Much of the cleaning action occurs at the eliminator plates where the dust particles are thrown against the film of water on the plates by their momentum.

The electrostatic precipitator is an effective available means of capturing the finer dust particles. This device does not entirely displace other types of air cleaners at the present time because of relatively high cost, large space requirements, and the fact that the air must enter and leave the apparatus in substantially parallel and straight flow.

Dust and Lint

Air-borne solid matter from the viewpoint of air cleaning is considered as lint and dust. Some lint originates outdoors, as animal hair, vegetable fibers, etc., but much is generated within buildings by the wear and brushing of fabrics in the form of clothes, draperies, carpets, etc. Lint is comparatively easy to capture in an air filter because of its comparatively great length. So far as air filter performance and testing are concerned, lint is chiefly important because of its tendency to impede or stop the flow of air through the filter. In general, lint, if not captured, will accumulate in corners and under furniture in a building in areas of slight air motion and in some cases may seriously obstruct heating and cooling coils. Dust settles, or is precipitated by heat or air motion, upon furniture, fixtures and walls and the only satisfactory treatment is washing or re-painting. Dust is more difficult to capture than lint, and, obviously, small particles are more difficult to capture than large ones. The air cleaning problem is complicated by the vast difference in size of dust particles, the range of which is shown in Fig. 1, Chapter 10.

Even if the discussion is limited to the range from 0.1 micron to 50 microns, that is; between the smallest particle observable in the microscope and the smallest particle distinguishable to the naked eye, this range is so far outside the usual experience that it is difficult to visualize. 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 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. Picturing this range in particle size from a golf ball to a sphere 50 ft in diameter will emphasize the difficulty of cleaning air or of devising any single test to adequately measure the performance of air cleaning devices under all conditions of service.

It may be contended that the removal of the finer particles from ventilating air is relatively unimportant insofar as cleanliness in a house or building is concerned, since it is possible that such particles may remain suspended in the air and in large part be removed from the building without settling by the circulating air. However, since some of the particles are undoubtedly deposited by contact and by thermal precipitation, the ability to remove small particles is desirable in an air cleaner.

The migration of dust particles from a warm region toward a cool surface, to which they will adhere, is called thermal precipitation[2]. This fact is responsible for the *lath marks* often observed on walls or ceilings. The laths form barriers to the passage of heat so that the surface of the plaster in front of them is warmer than the surfaces between them. Dust is therefore deposited more rapidly between joists or laths than it is in front of them, resulting in the streaks observed. If the entire ceiling or wall is insulated, the differences in temperature across the surfaces cease to exist, lath marks do not form, and deposition of dust is much slower.

A laboratory apparatus has been designed employing thermal precipitation in capturing dust particles for microscopic examination[3]. So far as is known thermal precipitation has not yet been used as a practical means of cleaning air.

CLASSIFICATION OF AIR CLEANING MEDIA

Air cleaners of such a variety of types have been used in the past that a single classification is difficult. Considering the wide diversification of materials and particle sizes to be removed and the varying requirements which have to be met in the field, it is natural that many kinds of air cleaning devices are used which cannot be shown satisfactorily in a simple outline. Classifications on three different bases are enumerated herewith:

1. Principle of air cleaning.
   - a. Viscous-impingement filters.
   - b. Dry filters.
   - c. Washers.
   - d. Centrifugal devices.
   - e. Electrical precipitators.

2. Methods of servicing.
   - a. Automatic.
   - b. Non-automatic.
     - (1) Throw-away (replaceable elements).
     - (2) Manually cleaned in place (including one type of electrostatic).
     - (3) Removable for cleaning.

3. Classification according to application.
   - a. General air conditioning.
     - (1) Central cleaning system.
     - (2) Unit ventilator.
     - (3) Window installation.
     - (4) Warm air furnace.
   - b. Removal of smoke and fumes from stack gases.
   - c. Collection of dusts from exhaust systems.

VISCOUS IMPINGEMENT TYPE CLEANERS

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 abruptly change direction 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. This includes 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 where the air carries a high percentage of lint.

The resistance of air filters obviously increases with the 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. Special filters with low resistances are available for use with gravity warm air furnaces and for other uses where only low pressure is available.

The resistance of these filters increases with dust or lint loading and it is the resistance due to this cause which 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. Manometers are often installed to indicate the pressure drop across filter banks and they serve to 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, providing the difference in resistance between the clean and dirty filters is small.

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 use. The frame is frequently a combination of cardboard and wire.

Cleanable types 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 latter may serve both to clean the filter and add the necessary adhesive.

Automatic Viscous Filters

In an automatic air filter, means are provided to remove the dust from the medium mechanically. Automatic filters with moving cloth media have been constructed. The media is supported on rollers and moves slowly and continuously across the air stream and then through some cleaning mechanism, such as a beater, a vacuum cleaner, or a brushing arrangement. Such filters, however, are not now in wide use, possibly because of mechanical difficulties and the rapid and sometimes permanent increase of resistance caused by oily matter present in the air.

The medium in a typical automatic filter at present 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 filtering 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 many small air passages are formed between them. These air passages turn abruptly one or more times in order to give the impingement effect.

An electrically driven rotating device is usually supplied with an automatic 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. Such a switch will cause the gear to move the curtain when the resistance of the filter to air flow becomes excessive and will stop it when the resistance becomes sufficiently low.

In operation, the resistance of an automatic filter will remain approximately constant as long as proper operation is obtained. A resistance of $\frac{3}{8}$ in. water at a face velocity of 500 fpm is typical of this class.

DRY AIR FILTERS

As the name implies, adhesives are not used on dry air filters. The media in such 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 are likely to have a comparatively high lint-holding capacity on account of the large area of medium used. Wool felt media are troublesome to clean when impregnated with greasy dust and they are too expensive to discard frequently. Both vacuum cleaning and dry cleaning have been used for reconditioning wool felt filters.

ELECTRIC PRECIPITATORS

The fact that a particle exposed to an electric field will assume a charge and migrate toward one of the electrodes has been utilized for some years in boiler plants as a means of smoke abatement. The same principle has

been used in equipment developed for air cleaning in air conditioning without generating ozone in intolerable quantities. The air stream in a precipitator passes first through a relatively high-tension electric field, known as the ionizing field, and then through a secondary field where the precipitation of the dust occurs. The arrangement is as shown in Fig. 1.

In a typical case, a potential of 12,000 volts may be used to create the ionizing field, and some 5000 volts between the plates upon which the precipitation of dust occurs. These voltages, which are capable of shock to personnel similar to that of a spark plug, necessitate some safety measures. A typical arrangement provides means for automatically making the unit inoperative when a door to the precipitator is opened.

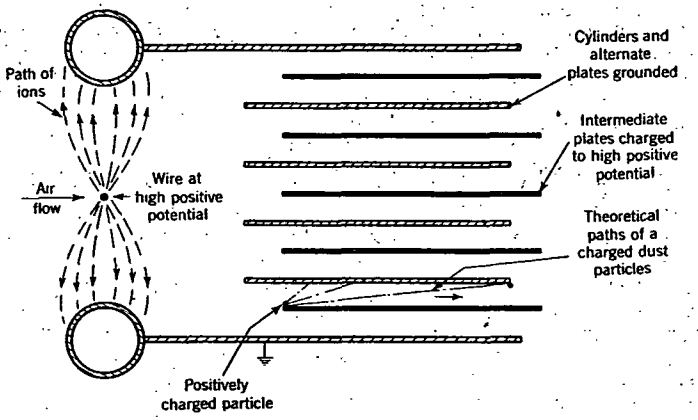


FIG. 1. DIAGRAMMATIC CROSS-SECTION OF ELECTROSTATIC PRECIPITATOR

To resume operation the procedure necessitates closing the door and turning an electric switch, the latter of which should be located at a reasonable distance from the equipment. The voltages necessary for the operation of the precipitator are usually obtained from an alternating current building service line by means of a step-up transformer. Precipitation with alternating current is possible but is not nearly so effective; so the current is usually rectified by means of vacuum tubes. The transformer and tubes are collectively termed the power pack.

Only a very small amount of electric energy is necessary to operate an electric precipitator and the resistance to air flow through the device is practically negligible. Some care is necessary in arranging the duct approaches on the entering and leaving sides of precipitators to assure that the air flow is distributed uniformly over the cross-sectional area. The efficiency of the precipitator is sensitive to air velocity and the device itself has much less tendency to rectify the air stream than filters, which have much higher resistances.

Electric precipitators are available in both automatic and non-automatic types. The plates of non-automatic precipitators are commonly coated with a light oil as an adhesive. Cleaning is accomplished with a water hose and, for this reason, the bottom of the equipment is made water tight and provided with a drain. In one automatic type, precipitation units are mounted on chains and are alternately dipped in oil and exposed to the air stream with an action similar to that of an automatic impingement filter. An arrangement of sliding contacts maintains the necessary electric circuits.

PERFORMANCE AND TESTING

The rating of an air cleaner is the air flow for which it is designed expressed in cubic feet per minute. Face velocity is defined as the average velocity of the air entering the cleaner, and it is determined by taking the air flow and dividing it by the area of the duct connection to the cleaner in square feet. Cleaners are often rated at a face velocity in the range of 250 to 500 fpm. The resistance of an air cleaner to air flow is usually measured in inches water. The resistances of filters when new and clean and when operated at rated capacity are generally available from the manufacturer (see *Catalog Data Section*).

The ability of air cleaners to clean air is called the efficiency or the arrestance, and may be denoted by the symbol E. The efficiency of an air cleaner differs with the size and nature of the dust on which the cleaner operates. Obviously, large particles and lint are more easily captured than minute particles which are small in all dimensions. The efficiency of an air cleaner, algebraically expressed, is:

$$E = \frac{D_1 - D_2}{D_1} \qquad (1)$$

where

D_1 = amount of dust per unit volume in uncleaned air.
D_2 = amount of dust per unit volume in cleaned air.

Several methods have been investigated for evaluating D_1 and D_2. The particle count method is not used for efficiency evaluation except in investigation of filter performance on specific particles 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 an inspection and analysis of dust is desired, but particle counting is not sufficiently precise for evaluating the efficiency of a cleaner operating on a heterogeneous dust.

The weight method of evaluating efficiency has found wide utility and was recognized by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and incorporated in a code [4]. For this test, 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 the 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 *dust-spot* or *blackness* test for cleaner efficiency was developed at the *National Bureau of Standards* [5]. The test consists of drawing samples of cleaned air and of uncleaned air through filter papers simultaneously. The ratio of the areas of paper through which the air samples are drawn and the ratio of the amount of air drawn through the papers are adjusted during successive trials to yield spots of approximately equal blackness on the papers. The ratios of the areas and of the volumes of the air samples are then indicators of the filter effectiveness. A special photometer is provided for comparing the blackness or opacity of the papers by transmitted light. For tests of ordinary air filters by this method, a

dust is injected into the air stream. The dust consists of precipitated smoke particles from a Cottrell precipitator used in a local power plant for smoke abatement. For tests of electrostatic air cleaners, no dust is added to the air. Tests are commonly made with the dust existing in the air at the location of the installation on a clear day. Some specifications for this type cleaner have required that in dust spots of equal area the downstream spot shall not be any dirtier than the upstream spot when 10 times as much air is drawn through it as is drawn through the upstream spot. When this condition is met the cleaner is said to have an efficiency or arrestance of 90 per cent or better on atmospheric air.

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 non-automatic air cleaners. Determination of dust-holding capacity is an objective of each test under the A.S.H.V.E. Standard Code [4]. Curves are obtained during such tests to show the relation between dust load and resistance. Typical curves are shown in Fig. 2. Type A is a dense pack used in bacterium control; Type B is a medium pack used for general ventilation work; and Type C is a low

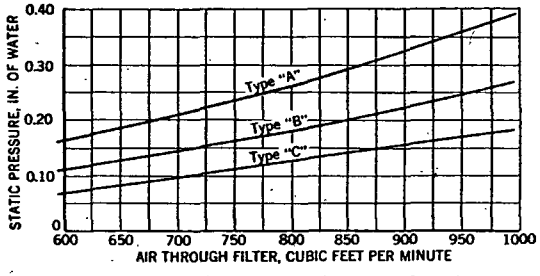


FIG. 2. RESISTANCE TO AIR FLOW OF TYPICAL UNIT AIR FILTERS

resistance unit, for use where low resistance is the important factor and maximum cleaning efficiencies are not essential.

At the *National Bureau of Standards* two injectors are provided on the air cleaner testing apparatus. One injector is used to contaminate the air stream with Cottrell precipitate, previously described. This dust is used to make both efficiency determination and dust-holding capacity tests. The other injector contaminates the air stream with cotton linters with which lint-holding capacity tests are made. The curves in Fig. 3 illustrate the difference in the characteristics of two filters, one a viscous-impingement type and the other a dry filter with a cellulose fiber medium. The two injectors can be operated either separately or simultaneously. A total dust deposit of 4 per cent cotton linters and 96 per cent Cottrell precipitate gives a deposit on a filter closely resembling those that occur in Washington, D. C.

SELECTION AND MAINTENANCE

If effectiveness in arresting dust were always the primary consideration, an electrostatic cleaner might be used for all air cleaner installations. Where the dust load is heavy, a filter bank may be installed ahead of the precipitator. At the present time electrostatic equipment is comparatively expensive and bulky and is therefore only used when the expense is justified by the need for the cleanest air possible.

The advantage of the automatic 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. This type of equipment is not any better in dust arrestance than some unit filters, and it ranks next to precipitators in first cost.

Unit filters constitute the majority of air cleaners now in use, and some choice is possible between the types available. Where lint in an eminently dry state predominates, a dry filter obviously may be preferable to other types on account 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. The cleaning difficulty is avoided if a throw-away type of medium is used. Some dry filters are capable of high efficiencies, compared to other unit

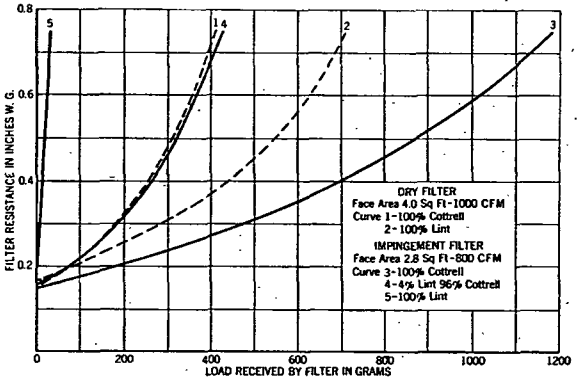


FIG. 3. DUST AND LINT-HOLDING CAPACITY OF TWO FILTERS

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 represent the general type of air cleaner now in use. They have approached standards in size and their over-all 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 automatic oil type if the resistance is to be maintained within reasonable limits.

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 a combustible filtering media may not be permitted in accordance with some existing local regulations. Combus-

tion 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.

The selective action of charcoal with a mixture of water vapor and organic gases is shown in Fig. 4.

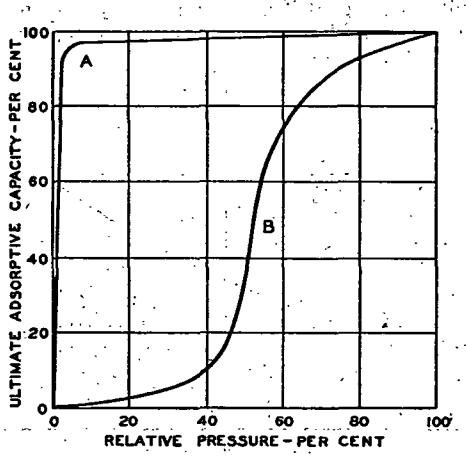


FIG. 4. ADSORPTION AT RELATIVE PRESSURES BY CHARCOAL OF A TYPICAL ORGANIC GAS (A) AND WATER VAPOR (B)

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 per cent of their own weight in 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 per cent or above when placed in intimate contact with the carbon at 70 F. In commercial apparatus there may be a by-pass effect which depends on the physical arrangement of the charcoal containers. This by-passing 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 the requirements for contaminant-free air, minus the outdoor air, divided 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.

REFERENCES

1—Bronchial Asthma and Allied Allergic Disorders, by S. S. Leopold and C. S. Leopold (*Journal of the American Medical Association*, March 7, 1925, Vol. 84, p. 731-734).

2—Dirt Patterns on Walls, by R. A. Nielsen (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1941, p. 247).

3—Industrial Dust, by Philip Drinker and Theodore Hatch (McGraw-Hill Co., New York, N. Y.).

4—A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225).

5—A Test Method for Air Filters, by Richard S. Dill (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 379).

BIBLIOGRAPHY

Design and Application of Oil-Coated Air Filters, by H. C. Murphy (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 73).

Operation and Maintenance of Air Filters, by W. G. Frank (*Heating, Piping and Air Conditioning*, May, 1931, p. 378).

Size and Characteristics of Air-Borne Impurities, by W. G. Frank (*Heating, Piping and Air Conditioning*, January, 1932, p. 35).

Fundamental Principles in the Design of Dry Air Filters, by Otto Wechsberg (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, April, 1933, p. 217).

The Economic Factors in Converting Recirculated Air for Ventilation, by H. E. Ziel and Henry Sleik (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1943, p. 367).

Operation, Maintenance of Cloth-Screened Dust Collectors, by W. F. Terry (*Heating, Piping and Air Conditioning*, May, p. 259, June, p. 304, 1933).

Testing and Rating of Air Cleaning Devices Used in General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 277).

A.S.H.V.E. RESEARCH REPORT No. 1094—Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed, and Air Velocity Through Filter, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 415).

A.S.H.V.E. 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, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 339).

A.S.H.V.E. RESEARCH REPORT No. 1145—The Effect of Lint on Air Filter Performance, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 25).

A.S.H.V.E. RESEARCH REPORT No. 1169—Comparison of the Weight, Particle Count and Discoloration Methods of Testing Air Filters, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 29).

A.S.H.V.E. RESEARCH REPORT No. 1187—Economical Air Velocities for Mechanical Air Filtration, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 391).

A.S.H.V.E. RESEARCH REPORT No. 1218—Overloading of Viscous Air Filters During Accelerated Tests, by F. B. Rowley and R. C. Jordan (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 437).

The Dust-Free Space Surrounding Hot Bodies, by H. H. Watson (*Transactions of the Faraday Society*, Vol. 32, 1936, Pt: 2, p. 1073).

A New Electrostatic Precipitator, by G. W. Penney (*Electrical Engineering*, January, 1937, p. 159).

Electrostatic Precipitation for Aircraft, by Howard E. Corbitt and Norman J. Clark (*Aero Digest*, December, 1940, p. 132).

Cleaning of Air and Gas by Thermal Repulsion, by Samuel C. Blacktin (*Journal Society of Chemical Industry, 68*, 1939, p. 334-8, *59*, 1940, p. 153-4).

Pointers on Selecting Equipment for Industrial Gas Cleaning, by C. E. Miller (*Chemical and Metallurgical Engineering, 45*, March, 1938, p. 132-5).

Electrical Precipitation, by W. A. Schmidt and E. Anderson (*Electrical Engineering, 57*, August, 1938, p. 332-338).

Electrostatic Precipitation for Cleaning Industrial Gases, by H. W. Wagner (*Fuel Economist, 10*, 1935, p. 895-9, 942-5, 971-3).

Dust Precipitation, by M. Pauthenier and Mme. Moreau-Hanot (*Electrician, 113*, August 10, 1934, p. 187-9).

Alternating Current Precipitators for Sanitary Air Analysis: II. Acid Formation in Electric Precipitators, by W. G. Hazard and Tomoyoshi Ishikawa (*Journal of Industrial Hygiene*, October, 1932, p. 367-70).

Electrical Precipitation, by A. W. Simon and L. C. Kron (*Electrical Engineering, 51*, February, 1932, p. 93-5).

The Mathematical Theory of the Cottrell Electric Precipitator, by A. W. Simon (*Iron and Steel Engineering, 6*, 1929, p. 143-6).

Progress in the Art of Electrical Precipitation Since 1900, by P. E. Landolt (*Transactions American Electrochem. Society, 51*, 1927).

Some Factors and Principles Involved in the Separation and Collection of Dust, Mist and Fume from Gases, by Evald Anderson (*Transactions American Institute of Chemical Engineers, 16*, 1924, Pt. 1, p. 69-86).

Electrical Precipitation of Solids from Smelter Gases, by Ross B. Rathbun (*Transactions American Institute of Electrical Engineers, 41*, 1922, p. 815).

Electrical Engineering Features of the Electrical Precipitation Process, by G. H. Horne (*Transactions American Institute of Electrical Engineers, 41*, 1922, p. 808).

CHAPTER 34

Automatic Control

Basic Types of Control, Types and Functions of Controllers, Designation of Temperature Control, Automatic Control Terminology, Residential Control Systems, Control of Automatic Fuel Burning Appliances, Zone Control, Control of Unit Systems, Control of Refrigeration Equipment, Application of Control Devices to a Typical System

THE function of *automatic control*, as applied to the heating, ventilating and air conditioning industry, may be broadly subdivided into the maintenance of *temperature*, *humidity*, *pressure*, and *liquid levels* within predetermined ranges. It automatically coordinates the operation of the various controlled mechanisms in proper sequence to produce the desired result. A thermostat, for instance, is a controller responsive to temperature which initiates a force that repositions valves, dampers, etc., as required to maintain selected temperatures. Similarly, a hygrostat or humidistat is a controller responsive to relative humidity which initiates a force that repositions the controlled sources as required to maintain a selected relative humidity.

BASIC TYPES OF CONTROL

There are two basic types of controllers, (1) the *two-position*, *on-off*, or *positive acting controller*, which functions to move a valve, damper or electric switch through its full travel (not necessarily 100 per cent) from open to closed position and vice versa, and (2) the *throttling*, *proportioning*, or *intermediate acting controller* which functions to reposition the valve, damper, or other controlled device by small increments of its total travel as the controller senses a slight change in the controlled condition. The available equipment may be broadly sub-divided into three main groups, (*a*) *self-contained equipment*, (*b*) *electrically operated equipment*, and (*c*) *pneumatically operated equipment.*

Self-Contained Equipment

In *self-contained controllers* the primary source of power is the vapor pressure of an enclosed volatile liquid which varies with the temperature changes of the controller bulb. Such pressure changes are transmitted directly into the bellows or diaphragm mechanism of the controlled valve. Devices of this type produce gradual operation and are used to regulate the flow of heating and cooling media to coils, radiators, and liquid tanks. Each instrument is designed to operate within a predetermined selected temperature range and for a given condition. Care must be exercised to select a type which has the ability to withstand any possible over-run of temperature at the control bulb without damage to the instrument. Instruments of this type are available either with a rigid bulb or with a flexible tubing which connects the control bulb to the head of the instrument. This tubing is usually protected by a flexible metal covering or armor and is available in varying lengths.

Electrically Operated Equipment

Electrically operated equipment for either two position control or throttling control is available in either low voltage or high voltage type. Low voltage equipment usually is designed to operate on 20 to 24 volt current. High voltage equipment may be used in connection with any of the

standard voltages. The individual units of this type of system are interconnected by wiring circuits which carry the commands of the controller to the motors operating the controlled valves or dampers and to stop and start electric motors driving the controlled equipment. Simple make and break electric thermostats are used to stop and start motors and for two-position control of valves and dampers. Modulating electric control may be accomplished by the unbalancing of some form of Wheatstone bridge which in turn causes the repositioning of a motor element which moves the valve or damper. In some controllers the impulse from more than one Wheatstone bridge can be integrated in one motor element.

Controllers for modulating electric control vary from the contact type in that the electrical mechanism consists of a variable potentiometer. The potentiometer includes a contact finger which moves across a coil of resistance wire wrapped on a suitable bobbin. The contact finger is actuated by a temperature, pressure, or humidity sensitive element.

When the same amount of current is flowing through both coils of a balancing relay the contact blade will be in the center of the space between the two contacts and the motor will be at rest.

Changes in the conditions at the controller will affect the circuit so that the amount of current flowing through the two relay coils will not be identical. Whenever the current through these two coils becomes unbalanced, one of them will become stronger and the *U* shaped armature will be tilted. The movement of the armature will close one of the relay contacts which in turn will start the motor running either clockwise or counter-clockwise, until the circuit is again balanced by the resulting movement of the motor balancing potentiometer, whereupon the relay contact will break and the motor will stop.

The development of specialized industrial control equipment using *electronic* principles has made rapid progress during recent years. Temperatures may be measured and controlled to fractions of a degree by new applications of the photo-electric cell. Its accuracy and flexibility make it applicable to a variety of automatic devices and controllers.

Pneumatically Operated Equipment

In *pneumatically operated equipment* the primary source of power is compressed air at a pressure of 15 to 25 lb per square inch which should be clean, dry and free from oil vapor. It is desirable to compress the air to a pressure of 70 lb per square inch or more to provide for a precipitation of moisture due to temperature drop in the high pressure receiver and to reduce the dew-point of the air to the controllers below that of the lowest ambient temperature of the air lines.

The compressed air piping from the source of supply to the inlet side of the controller is known as the *main*. The connections from the outlet of the controller to the pneumatic motors are known as *branches*.

Throttling type controllers, of the pneumatic type, are basically air pressure regulating valves actuated by the controlled condition to vary the *branch line pressure* and thereby reposition the controlled valves or dampers as required to maintain the desired condition of the controlled media. A direct acting controller increases its branch pressure on an increase of the controlled condition. A reverse acting controller increases its branch pressure on a decrease of the controlled condition. Several different devices may be operated from the branch line of a single controller in selected sequence by the proper spring loading of the respective pneumatic diaphragm motors. Small air pressure gages connected into the branch at each controller will indicate the controller action and are of great convenience to the operator in adjusting the system.

The simplest form of pneumatic thermostat is based on the leak stat principle which consists of a restrictor in the air supply to the instrument, a leak port having a larger port than the restrictor, and a bi-metallic strip or diaphragm chamber which expands on a rising temperature and contracts on a falling temperature. This power is used to reposition the lid seat to close or open the leak port. As the leak port is gradually closed the pressure in the *branch* increases until it approaches the full *main* pressure in the fully closed position. As the lid seat moves away from the leak port the pressure in the *branch* decreases until the *branch* is fully exhausted when the lid seat restriction of the leak port is entirely removed. The basic leak stat principle of operation has been refined by the addition of supplemental relays in nearly all current types.

TYPES AND FUNCTIONS OF CONTROLLERS

Descriptions or functions of various types of controllers follow.

A *single-duty room thermostat* controls the heating or cooling devices to maintain the desired temperature.

A *dual or two-temperature room thermostat* controls the respective heat sources to maintain alternately two selected temperatures. They may be indexed manually or by time switch from central points in selected zones.

A *high-low room thermostat* is similar to the dual thermostat, except that it is indexed individually by the manual operation of a push button or switching device built into the individual instrument.

A *sub-master or remote readjustable room thermostat* is a pneumatic room type instrument fitted with a readjusting diaphragm to which a separate pilot air line connection is made. As the pressure in the pilot line is raised from 1 psi to 13 psi, or as selected, the operating point of the controller may be increased or decreased within predetermined limits. Indexing may be done by a manual gradual switch or master thermostat.

A *pneumatic master or pilot thermostat* varies the pilot pressure to the readjusting diaphragm of one or more sub-master thermostats to vary their operating point in accordance with a predetermined change in the temperature sensed by the master thermostat.

A *summer-winter room thermostat* operates at any selected temperature as a direct acting controller during the heating season and operates at the same or any other selected temperature as a reverse acting controller during the cooling season. Groups of these instruments may be indexed from a central point or individually indexed by a manual push button or a switching lever built into each instrument. They may be used to control volume dampers of a summer-winter air conditioning system to maintain a desired temperature condition in the controlled space, and will operate to close normally open dampers on a rising temperature during the heating season and to open same on a rising temperature during the cooling season.

A *summer-winter sub-master or summer-winter remote readjustable room thermostat* is similar to the summer-winter thermostat but is arranged to permit resetting of the control point by a master thermostat or by a manual gradual switch or other selected device.

A *humidity compensated thermostat* is a room type gradual acting thermostat combined with, and readjusted by, a hygroscopic element. It may be used to maintain a constant *effective* temperature.

The humidity sensitive element usually functions to raise the operating point of the temperature sensitive element one degree for a predetermined drop in relative humidity and lower the operating point of the temperature sensitive element proportionately for a similar increase in relative humidity.

An *insertion or duct thermostat* of tube type is usually equipped with a rigid bulb, consisting of a brass tube enclosing an invar rod which usually has a maximum range of 0 F to 250 F, or consisting of a vapor disc type which operates through a similar range but in which the type of disc must be selected to function at given temperature increments throughout that range. It is available in the remotely readjustable or sub-master type, which may be reset in the same manner as the *sub-master* room type thermostat. This type of instrument may also serve as a *master* thermostat.

An *immersion thermostat* is similar to an insertion thermostat, except that the bulb fitting is threaded for insertion through the wall of a vessel containing a liquid. A union connection and separable well or socket are available where it is desirable to provide for

the removal of the controller bulb without the necessity of draining the vessel to the bulb level.

A *differential thermostat* has two flexible capillary tubes fitted with plain bulbs, or with bulbs having flanges or having threaded connections. One sensitive element is subjected to the controlled temperature and the second element is exposed to the temperature which is to determine the controlled temperature. The instrument functions to maintain a definite relation between the two temperatures.

A *remote bulb or extended tube controller* may be of liquid, vapor, or gas filled type. A bulb of this type of instrument is connected to the instrument head by means of a flexible capillary tube of the desired length. The external diameter, bore, and material used to form the capillary vary with the type of control system.

An *indicator controller* is a controller fitted with a pointer, thermometer, or gage which indicates the controlled condition.

A *recorder controller* is a controller, combined with a clock mechanism and chart, which both controls and records the controlled condition.

A *time cycle controller* is a controller equipped with a clock mechanism which automatically raises and lowers the control point of the instrument to meet a required time schedule.

A *hygrostat or humidistat* is an instrument for controlling relative humidity. It may be of a room, insertion, indicating or recording type, and may be direct or reverse acting.

Many forms of hygroscopic materials, such as human hair, wood, bi-wood and membrane of selected type, are used as the sensing medium for these instruments. Where the controlled condition ranges below 20 per cent or above 70 per cent, or the dry-bulb temperature is above 100 F, careful selection of the hygroscopic element is essential.

A *static pressure controller* is frequently used to provide a constant pressure in a duct system in which the volume of air supply is varied to maintain temperature or some other condition, and usually it operates to reposition dampers installed in the fan vortex or other selected location. It usually functions to maintain a pressure differential between the atmosphere and the pressure or vacuum of the controlled condition. It is frequently necessary to pipe the neutral chamber to a selected location to obtain a true basic condition.

A *velocity regulator* is a controller actuated by the velocity of the medium which it controls.

A *solenoid air valve or electric pneumatic switch* is an electrically actuated three-way air valve which may be either direct or reverse acting. It is frequently used in connection with fan ventilating systems to close outdoor intake dampers and heater coil valves during the period when the fan motor is stopped.

A *pneumatic relay or cumulator* is a device fitted with a readjusting diaphragm to which a separate pilot connection is made. It is installed in the branch between the controller and the controlled valves or dampers, and functions to accelerate, magnify or reverse the action of the controller. Direct acting relays increase the branch pressure on an increasing pilot pressure. Reverse acting relays decrease the branch pressure on increase of pilot pressure.

A *duplex cumulator or relay* is a device having two pilot connections either of which will vary the branch pressure on a changing pilot pressure.

An *averaging cumulator or relay* is a device arranged to vary the branch pressure as dictated by the average of the pressure of two or more pilot connections.

A *mulliple cumulator or relay* is a device having more than two pilot connections any one of which will vary the branch pressure on a changing pilot pressure.

A *control valve* may be considered as a variable orifice designed to control the flow of liquids, gases and other fluids.

The use of the flat disc type of valve is usually restricted to on-off control requirements. Throttling valves are generally equipped with selected forms of plug to produce the flow characteristics required for the special problem. Special plugs can be shaped to produce almost any desired flow characteristics. The three types of plugs most widely used are the V-port skirt type, the throttle plug, and the ratio plug.

A *diaphragm valve* is a valve which is operated by a diaphragm attached to the valve stem. Increasing air pressure on the diaphragm moves the valve against the pressure of opposing springs which return the valve to its normal position on a decreasing diaphragm pressure. The pressure range required to fully reposition a valve has four limiting factors, viz., the size of the diaphragm top, the tension of the selected spring, the packing gland friction and, for all single seated valves, the pressure of the controlled media against which the valve must hold.

A *direct acting or normally open diaphragm valve* will assume an *open* position due to spring action when all operating power is removed.

A *reverse acting or normally closed diaphragm valve* will assume a *closed* position due to spring action when all operating power is removed.

A *double seated balanced or semi-balanced valve* is designed for use with high pressures and is practically balanced regardless of the pressure differential across the plug. The diaphragm tops for valves of this type need only have ample power to close the valve against the tension of the opening spring plus the friction of the valve packing gland. While plugs and seats for this type of valve can be arranged to provide for tight seating for special purposes, such a requirement is not ordinarily expected.

A *three-way mixing valve*, having two inlets and one outlet, is fitted with a double faced disc, operating between two inlet ports, which functions to close one port as the other is opened, and is used for mixing service.

A *three-way diverting or by-pass valve* has one inlet and two outlets with two separate discs operating on the outside of the respective valve seats to close one as the other is opened; and is used for diverting service.

A *pilot positioner or valve positioner* is an auxiliary control device repositioned by the controller pressure and having a direct connection to the valve stem. It permits the full available air pressure to be used on the diaphragm top to overcome hysteresis, packing gland friction and pressure variation of the controlled media, to position the valve precisely in accordance with the controller pressure.

A *pneumatic damper motor* is a diaphragm or bellows assembly operating against resisting springs to reposition an arm or lever assembly. They may be attached directly to the damper frame, either inside or outside of same, or fitted with brackets for wall or floor mounting. The springs used to return the motors to their normal positions are available in several tensions to permit their operation in selected sequence. Pilot positioners may be applied to damper motors in the same manner as described for diaphragm valves.

Pneumatic dampers in either the single blade or multi-louver type should be substantially built in heavy frames, properly braced to hold true against any sag in the supporting structure, in order to prevent distortion of the louvers and binding of their supporting trunnions. When large dampers are used for open and shut control, the regular multi-louver type of damper serves the purpose. Where accurate volume control is desired with gradual positioning of the damper blades, the adjacent louvers should be arranged to move in opposite directions in order to properly proportion the available free area through the damper from its fully open to its fully closed position. Multi-louver dampers with louvers which all move in the same direction provide full free area through all but their top and bottom louvers when the louvers reach 45 deg open position. Where fairly tight closing is desired, the damper frames should be fitted with solid stops against which the louvers can close and where additional precautions are necessary, rubber or felt stripping may be glued and riveted to the louver edges. The metal frames of the smaller dampers are usually made of ¼-in. stock. Solid stops are usually ½ in. deep. The free area of the duct enclosing the damper is thereby reduced 1½ in. in each dimension through the damper. Where channel frames are used, the restriction is increased accordingly. Damper louvers should be suspended on non-corrosive trunnions. Thrust bearings are necessary to properly support large vertical louvers. Where ball or roller bearings are required for special conditions, they should be of the non-corrosive type.

DESIGNATION OF TEMPERATURE CONTROL

Temperature may be controlled from three basic conditions:

Dry-bulb Control: The controller element senses the dry-bulb temperature.

Wet-Bulb Control: The controller element senses a true wet-bulb temperature. Wicking of selected material suspended from the controller element to a water trough directly below the element is one of the mediums frequently used to obtain wet-bulb temperature. True wet-bulb temperatures can only be maintained with clean wicking and air motion of not less than 600 fpm. The use of distilled water in the wet-bulb reservoir is desirable. Since the wet-bulb temperature indicates the heat content, this form of control while requiring additional and intelligent care to maintain true wet-bulb conditions, has many definite uses.

Dew-Point Control: A saturated condition at a predetermined temperature is maintained.

AUTOMATIC CONTROL TERMINOLOGY

The following is a generally accepted terminology referring to the operations of controls in a central air conditioning system.

Control Point: The desired constant condition of the controlled media to be main-

tained. Some change of the controlled condition must occur to alter the prevailing controller action.

Thermometer Lag: The instantaneous temperature differential existing at any time between the true temperature of the medium and the temperature indicated on the thermometer.

Controller Lag: The time delay in the controller's ability to reposition the controlled sources to properly compensate for a change in the controlled condition.

Process Lag: (Process Time Lag) The time which elapses between the instant the control valve is positioned by the controller and the instant the controller element senses the effect of the change. This depends upon the rate of heat transfer, the specific heat, and the velocity of the fluid movement. Generally a decrease in process time lag simplifies the control problem.

Rate of Load Change: Time element of change in the magnitude of load.

Magnitude of Load Change: The per cent load at a specified time. It directly affects the size of the controlling valve. Where the inlet temperature varies constantly, the control problem becomes more difficult as the thermal difference between the inlet temperature and the leaving temperature of the controlled media becomes greater.

Drift: The deviation from the control point due to change in load conditions from 0 to 100 per cent.

Drift increases as the sensitivity adjustment of the controller is lowered, thereby tending to decrease hunting. Drift decreases as the sensitivity adjustment of the controller is raised, thereby increasing hunting.

Hunting or Cycling: The temperature variation between the high and low points of the control cycle due to lag. This balances out at the point where each successive wave of temperature reaching the controller is as large as its predecessor. As the sensitivity is lowered, hunting decreases but drift increases. Oversized control valves increase hunting.

Throttling Range or Sensitivity Adjustment: The total rise in temperature at the controller bulb required to reposition the control valve from a fully open to a fully closed position or to operate several devices controlled in sequence throughout the full sequence range. In connection with chart type controllers, throttling range is usually expressed in per cent of the total range of the chart. For other types of instruments, sensitivity adjustment is expressed in pounds variation of branch pressure per degree change in temperature, or per other unit of change of the controlled media.

Low Sensitivity: A small change in branch pressure for a comparatively large temperature or pressure change in the controlled medium. This results in a greater deviation from the control point. Low sensitivity tends to reduce hunting but the lower the sensitivity, the greater will be the drift.

High Sensitivity: A large change in branch pressure for a comparatively small temperature or pressure change of the controlled medium. The extreme of high sensitivity is the on-off or two-position control. As the sensitivity is raised, drift decreases, but the tendency to hunt increases.

Automatic Reset: A mechanical addition to a controller which corrects for drift by constantly resetting the instrument to operate on the control point, while repositioning the controlled valve or damper from its fully open to its fully closed position. Reset action is superimposed on throttling range and both function simultaneously. The rate of reset is manually adjustable and must be set to meet the load requirements of the individual system. This supplementary device provides the accuracy of high sensitivity adjustment with a minimum of hunting.

Diaphragm hysteresis is the amount by which the stem travel of a given motor fails to assume the identical position on the upward and downward strokes for an identical air pressure in the diaphragm top. It prevails in varying degrees in connection with the operation of all pneumatic diaphragm motors, and is due to spring and diaphragm hysteresis and friction of guide bearing and stuffing box. Assume that a direct acting valve moving to a closed position will travel 50 per cent of its total with 8 lb air pressure on the diaphragm top. If the diaphragm pressure is increased to further close the valve and then reduced to 8 lb, the valve may be repositioned to be only 45 per cent open, indicating 5 per cent hysteresis. Hysteresis is usually expressed in per cent of the full stroke of valve stem.

RESIDENTIAL CONTROL SYSTEMS

The control installation in a residence may vary from the simple regulation of a coal-fired heating plant to the completely automatic all year air conditioning system. Residential installations with automatic fuel burning appliances, such as oil burners, gas burners or stokers, are normally equipped with single room thermostat, limit and safety controls as outlined under Control of Automatic Fuel Burning Appliances.

Coal-Fired Heating Plant

Control in the normal coal-fired domestic heating plant consists of regulating the combustion rate in accordance with requirements. This function is accomplished by a spring or electric-driven damper motor which, under the command of a room thermostat and through chain linkage, operates the draft and check dampers of a boiler or warm air furnace. Such installation should be protected against excessive temperature or pressure by means of a limit control serving to check the fire when conditions at the boiler or furnace reach a predetermined maximum.

All Year Domestic Hot Water Supply

Hot water or steam heating boilers with automatic fuel burning appliances can be used for all year heating of domestic water supply. The fuel burning appliance in this case is controlled from the temperature of water or pressure of steam in the boiler to maintain uniform boiler conditions and domestic hot water is heated by means of an indirect heater. The heating of the residence is normally governed by means of a thermostat which operates a control valve in the flow line of a gravity hot water or a steam system or which controls the circulating pump in a hot water system.

Air Conditioning Systems

Residential air conditioning systems normally include a heating source and a motor-driven fan for circulating air. In addition, such installations may involve spray-head equipment to supply humidity. Such installations distribute heated and humidified air during the heating cycle, and during the summer or cooling cycle may be used effectively as conditioners if equipped with refrigeration means.

Regulation of the humidity during the heating cycle is normally accomplished by opening and closing a solenoid water valve supplying water to the spray-heads, the solenoid valve being under control of a room type humidity control. In the average installation the fan is permitted to run only during such intervals as the thermostat is calling for heat or at the command of a limit control to prevent the overheating of the bonnet of a warm air furnace. The limit control should also prevent the operation of the fan at the command of the thermostat until the circulating air temperature has increased to a predetermined point.

For the cooling equipment provided in such installations, control during the cooling cycle will be an adaptation of the control principles described for central fan systems selected for the type of cooling equipment utilized.

CONTROL OF AUTOMATIC FUEL BURNING APPLIANCES

It is essential that automatic controls be used with oil burners, gas burners, and stokers in order to maintain even temperatures and provide safe and economical operation of the heating plant.

Oil Burner Controls

In the normal oil burner installation as encountered in residential and small commercial installations, the burner operation is frequently regulated by electric controls and primarily governed by a room thermostat. It is essential that a limiting control be incorporated in the control system to prevent the temperature of the heating medium from exceeding any

predetermined safe maximum. The type of limit control selected will depend on the type of the heating system. In a warm air furnace installation, a limit control would be used, reacting to the temperature of the heated air in the bonnet of the furnace; in a hot water system a control reacting to the temperature of the water in the boiler; and in a steam system a control reacting to the pressure of the steam in the boiler.

In addition to the normal control of the burner from the room thermostat and limit control, it is necessary that a combustion safety device be used to prevent operation of the burner under hazardous conditions. The oil fire is automatically ignited by means of gas, electric spark or incandescent element and the combustion safety control acting through a sequence device permits the burner operation only when the fire is properly established as the burner starts up. A further function of the combustion safety control is to react to any major disturbance in the flame during the running operation, shutting down the burner and preventing the discharge of unburned fuel if for any reason the flame is extinguished.

Gas Burner Controls

In the case of the domestic burner, full automatic operation is the normal requirement and the burner is started and stopped at the command of a room thermostat which, in turn, opens and closes a control valve in the gas supply line. Modulating controls and controls providing a high and low fire are also available for gas burners. For purposes of preventing abnormally high temperatures in the bonnet of gas-fired furnaces or in the temperature of the water in gas-fired hot water heating boilers or excessive pressures in gas-fired steam boilers, temperature and pressure limit controls are used. Ignition is normally secured through the use of a gas pilot flame and a safety device is provided, utilizing the heat of the pilot flame in such a manner that if the pilot light is extinguished for any reason, the main gas valve cannot be opened. For satisfactory and economical operation, all automatically-fired gas burners should be equipped with pressure regulators on the gas supply line.

Stoker Controls

Domestic stokers are normally placed under command of a room thermostat for primary operation subject also to the command of a limit control to prevent their operation when conditions in the boiler or furnace exceed predetermined safe maximums. Utilizing coal as fuel, automatic ignition is not provided and the stokers, once ignited, maintain their fire, merely changing the rate of combustion by changing the draft and the rate at which the coal is fed. Thus, at the command of the room thermostat the stoker motor is started, driving a forced draft fan and fuel feeding mechanism. The rate of combustion is thus increased and this operation continues until the thermostat has been satisfied when the motor is stopped and the fuel in the combustion chamber continues to burn at a slow rate with reduced draft.

Automatic controls may be used to operate the stoker sufficiently to maintain a fire in mild weather and also to prevent feeding of fuel if the fire is extinguished.

ZONE CONTROL

Zone control for winter heating of buildings, in which a single room thermostat will not provide satisfactory conditions and where the refinement of control from individual room thermostats is not a requisite, may be secured in various ways. According to its size, use and exposure, the

building may be divided into zones where the general requirements will be relatively constant. The number of zones is determined by (1) size of building, (2) number and character of exposures, (3) variation in occupancy and other inside conditions, and (4) cost of additional zones. For large buildings it is desirable to have at least one zone for each exposure and for high buildings, each exposure may be sub-divided into upper and lower zones in order to properly provide for the stack effect. In buildings of this type it is advisable to provide a separate main with local thermostats for the street floor heat sources, especially those adjacent to entrances and exits.

For small buildings or where cost or other conditions limit the number of zones, a frequent compromise is to combine the North and West exposures in one zone and the South and East exposures in a second zone.

For steam heating systems, the radiator output may be proportionately reduced as the outdoor temperature rises by several methods. Some of those in common use are:

1. Throttling steam pressure to allow flow through orifices in proportion to the needs for heating.
2. Turning steam on and off on time intervals proportioned to the needs for heating.
3. Varying the absolute pressure of steam in the system.

The usual zone controlling device is arranged to sense the effect of sun, wind, or rain, as well as the outdoor local zone temperature, and in some instances is fitted with additional elements sensing the indoor zone or zone radiator temperatures. It functions to regulate the rate of flow or the flow impulse as required to maintain the desired indoor zone temperature condition. A switch panel containing manual switches arranged to raise or lower the operating point of each zone controller is frequently mounted at a selected location. These panels may contain time switches for automatic cycling of day-night or other predetermined control programs. In buildings where some zones or all zones have no night occupancy, provisions may be made to index the control system to maintain predetermined low economy temperatures during the unoccupied periods and short morning warm-up period, and to obtain normal operation throughout the occupied period. Either manual operation or automatic cycling by time switch may be used. For hot water heating systems, the zone controller functions to regulate the water temperature in accordance with the prevailing outdoor temperature.

For discussion of zoning in air conditioned buildings see section on Zoning in Chapter 43.

CONTROL OF UNIT SYSTEMS

Because of the usual segregated location of unit equipment throughout a building and its consequent lack of competent supervision, complete automatic control is essential to its satisfactory operation.

Unit Heaters

In its simplest form, unit heater control consists of a room thermostat to start the unit heater motor when heat is required and shut it off when the demand is satisfied. With this limited control, it is possible in some instances that, with no steam available at the heater, the operation of the fan would cause objectionable drafts. To avoid this, limit controls are available which will prevent the operation of the fan at the command of the room thermostat except when steam is available, as determined by

the temperature of the steam or return pipe or the pressure of the steam supply.

Where several unit heaters serve a limited area, they may be grouped for purposes of automatic control, and several heaters placed in operation at the command of one thermostat. By properly grouping the units which will operate together, the benefit of zone control can often be obtained with a minimum of control equipment. Where such group operation is utilized, the thermostat and limit control usually function through a relay, as the combined load of the several motors may exceed the current capacity of the thermostatic control device.

In some cases where cold drafts will not result, it is desirable to operate the unit heaters continuously for circulation of air. In such instances the room thermostat regulates the supply of steam to the unit through a control valve in the steam supply line and the unit heater motor operation is manually controlled.

Unit heaters equipped with dampers arranged for by-passing air around the heating coils are controlled by room thermostats operating modulating damper motors attached to these dampers so that as the temperatures rise, a decreasing amount of air is heated. When the by-pass is wide open the heating effect is so much reduced that control of the steam supplied to the coil is not generally important. If valve control is added, the throttling of the steam may be concurrent with, or subsequent to, the opening of the by-pass.

Cooling Units

The recommended form of temperature control for a cooling unit contemplates the continuous operation of the fan, with automatic regulation of the compressor or cooling coil, or both, as determined by a thermostat in the room, or in the return air to the cooling unit. Such operation insures continuous circulation of air in the room, and in addition to providing the cooling effect of moving air, overcomes the tendency of the air to stratify. As the temperature begins to rise, the controller opens the valve to a cold water cooling coil, or for direct expansion coils, opens a valve in the refrigerant line, closes a by-pass around the coil or starts a compressor.

Cooling units may also be controlled by arranging the room thermostats to start and stop the fan motors or by a combination of motor and refrigerant control.

Unit Ventilators

There are various types of unit ventilators available but in general all types are designed to draw air from the outside or to mix outside and recirculated air, heat it and introduce it into the room under control of a thermostat.

The design of unit ventilators has to an extent been based on the requirements for automatic temperature control and the cycles of control have been developed to include other heating devices in the rooms with unit ventilators. Unit ventilators are frequently used in schools and other types of buildings where states have laws or regulations governing the minimum amount of ventilation to be provided. The control of the amount of outdoor air is designed to conform to the various laws. Usually the device circulates a constant amount of air and the amount automatically taken in from outdoors is controlled in one of these ways:

1. Full recirculation until the room temperature reaches a certain point, generally two degrees, below the desired room temperature; then a minimum amount of outdoor air for ventilation while the temperature is maintained by throttling steam; and if the room temperature rises with all steam shut off, a gradual increase in amount of outdoor air up to 100 per cent.
2. Full recirculation until the room reaches a set point below room temperature, after which all air is taken from outside.
3. Gravity recirculation while the fan motor is not running, with full outside air as soon as the fan starts, obtained by a relay in the motor circuit.
4. Full recirculation or all outdoor air as determined by a manual switch which can be operated at any time whether or not the fan is running. All the unit ventilators in a single building may be operated by one or many switches.

With arrangements 1 or 2, it is desirable to include a relay to prevent the intake dampers from opening while the fan is not running, regardless of room temperatures. With a dual system of control this is essential to prevent the thermostat keeping the outside damper open until the temperature falls to the reduced setting.

The intake and recirculated air quantities are determined by a single damper or by a pair of dampers working together, and operated by a damper motor. Although this affects the temperature of the air delivered, the main heat control comes from the throttling of the steam supplied to the heating coil, with or without by-pass damper control. To prevent air being delivered at too low a temperature, a low limit thermostat is commonly installed in the air stream and set at some point between 55 and 70 F. The lower settings may cause discomfort, the higher ones overheating, depending on circumstances. The air stream thermostat can be used to turn on steam, reduce the amount of outside air, or both.

Rooms with unit ventilators frequently have auxiliary heating devices, such as direct radiators, convectors or unit heaters, all under control of a single room thermostat. A common control cycle for such rooms is composed of the following functions, assuming that 72 F is desired:

1. Below 70 F the unit ventilator intake damper is in full recirculating position and all heat is turned on.
2. At 70 F the intake damper moves to a position that will admit a predetermined minimum amount of air from outdoors.
3. At 71 F the auxiliary heating devices are shut off.
4. From 71 to 72.5 F, the heating effect of the unit ventilator is throttled.
5. From 72.5 to 74 F, the intake damper is gradually moved to increase the amount of outside air from the set minimum to 100 per cent.
6. If the room thermostat calls for too much cooling, the air stream thermostat holds the delivery temperature at a proper minimum.

Other similar cycles may be used. One additional feature is the use of an air stream thermostat that has its control point reset by the room thermostat. Then as the room temperature rises, the delivery temperature is gradually reduced from a maximum to a minimum.

CONTROL OF REFRIGERATION EQUIPMENT

The most common means of providing cooling for air conditioning may be divided into four general classifications as follows:

Refrigeration compressors may furnish refrigerant to direct expansion cooling coils through which air is being passed, or to coils in cooling tanks through which water is passed which is then pumped to air washers or cooling coils through which the air is passed.

In either case the compressor motor may be started and stopped in order to meet the demand for refrigeration or a pressure controller may be used to regulate the low side or suction pressure of the compressor. When the latter method is used, the flow of refrigerant to cooling coils may be regulated by the opening and closing of a solenoid refrigerant valve at the command of a temperature controller or thermostat.

A high pressure cutout as an individual unit or in combination with either a temperature or pressure controller provides a safety feature against excessive pressures on the high side of the compressor.

Many compressors may be *unloaded* by instruments sensing room or duct conditions, or by refrigerant pressures, thus reducing the frequency of starting and stopping. If two or more compressors are used for a single cooling system, *step controllers* are used to start them in sequence at intervals of a few seconds to avoid the large momentary electric input that simultaneous starting would demand.

When condensers are water cooled, thermostatic control to vary the quantity of water is needed for economical operation. Mechanical air condensers may be started and stopped with temperature demands.

Chilled water may be stored in tanks at temperatures slightly lower

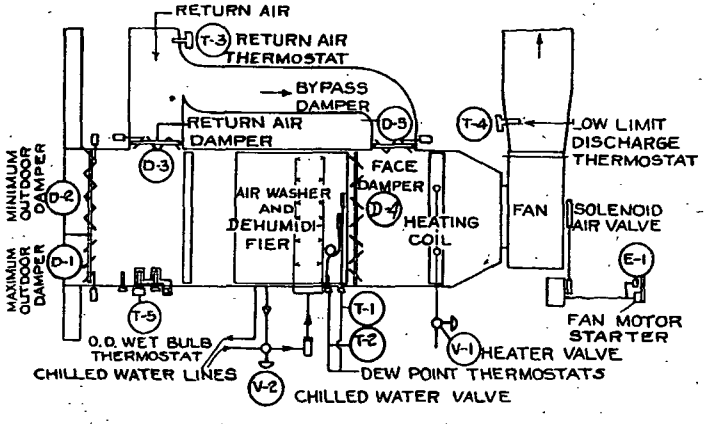


FIG. 1. LOCATION OF CONTROL DEVICES FOR YEAR 'ROUND AIR CONDITIONING SYSTEM

than required for air cooling coils. The control of temperature for the water distribution system is as described for Ice Cooling.

When ice is used for the cooling or dehumidification of air, it is usually placed in bunkers and water is sprayed over it. This water, after being cooled, may be used in air washers or surface cooling coils and is usually returned to the bunker for additional cooling after being used.

Control of the water temperature leaving the cold water tank may be maintained by a temperature controller, which measures the temperature of the water in the tank and modulates a control valve in a by-pass which permits a portion of the return water to return directly to the tank instead of passing through the sprays.

A vacuum refrigerating system consists of an evaporator, compressor, condenser and auxiliaries. The refrigerant used is water, and water vapor (steam) is the power medium.

Water which has been passed through an air washer or cooling coil is sprayed directly into the evaporator or water cooler where it is cooled by its own evaporation. A condenser is attached directly to the compressor

discharge and its function is to recondense the water vapor drawn from the evaporator, plus the steam which supplies the energy for compression.

The temperature of the cold water leaving the flash chamber should be measured by a temperature controller which will in turn operate a two-position or positive-control valve installed in the steam line to the jet so as to permit steam to flow only when cooling is required. If city water is

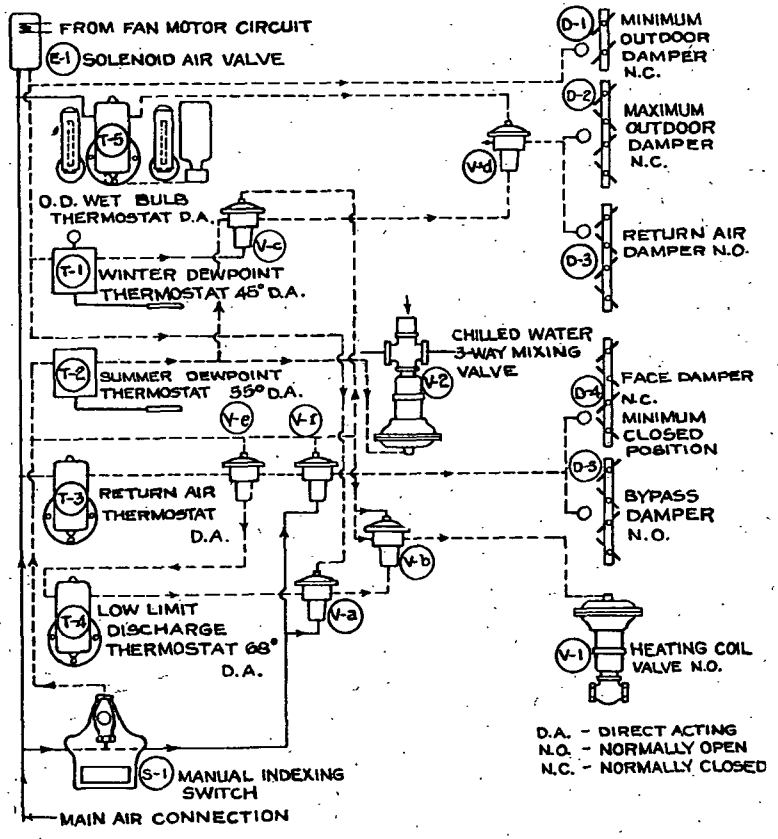


FIG. 2. CONTROL DIAGRAM FOR YEAR 'ROUND AIR CONDITIONING SYSTEM

used in the condenser, the amount of water should be modulated according to the demand as measured at the condenser outlet by means of a temperature controller and control valve.

Well water, if available in sufficient quantities at low temperatures during the cooling season, may be pumped directly to air washers or cooling coils. Control is usually effected through control valves on the water supply to the cooling unit actuated by temperature or humidity controllers, or both, located either at the outlet of the conditioner or in the conditioned space.

APPLICATION OF CONTROL DEVICES TO A TYPICAL SYSTEM

Fig. 1 shows the location of controlling devices for a year 'round air conditioning system such as shown in Chapter 43, Fig. 3. A control diagram for pneumatic control equipment is shown in Fig. 2 for convenience in explaining the function and sequence of operation of the control in Fig. 1, but obviously, the individual controls may also be of self-contained or electrically operated type provided they obtain the same control of valves and dampers.

The auxiliary controlling devices indicated may all be mounted together on an instrument board which may also contain a framed copy of the control diagram and the description of the automatic control cycle. Air gages, identified by suitable inscription plates, may be installed in the branch connections to and from the auxiliary devices on the instrument board to indicate the functioning of the various devices.

A description of the automatic control cycle follows:

When the fan motor is stopped, solenoid air valve E-1, actuated from the fan motor circuit, is de-energized and exhausts its branch to close minimum outdoor damper D-1, reposition three-way air valve V-a to close heating coil valve V-1, and remove main air from remote bulb indicator dew-point thermostat T-1, thereby closing maximum outdoor damper D-2 and opening return air damper D-3.

When the fan motor is started, E-1 fills its branch, thereby opening minimum outdoor damper D-1, repositions V-a to permit T-3 and T-4 to control V-1 and supplies main air to T-1 to permit same to operate D-2 and D-3.

During the summer cooling season, manual indexing switch S-1 is positioned to fill its branch, whereby it supplies main air to dew-point thermostat T-2, positions three-way air valves V-e and V-f to permit T-3 to control face damper D-4 and by-pass damper D-5, positions three-way air valve V-b to close heating coil valve V-1, positions three-way air valveV-c to remove the control of maximum outdoor damper D-2 and return air damper D-3 from dew-point thermostat T-1 and place these dampers under the control of dew-point thermostat T-2.

When S-1 is positioned as noted, remote bulb dew-point thermostat T-2 functions on a rising temperature to first gradually open maximum outdoor damper D-2 while simultaneously closing return air damper D-3 and on a further slight temperature rise, gradually positions chilled water valve V-2 to pass chilled water to the dehumidifier. The reverse operating sequence occurs on a falling dew-point temperature. Should the outdoor wet-bulb temperature rise above the desired indoor wet-bulb temperature, positive acting outdoor wet-bulb thermostat T-5, positions three-way air valve V-d to remove dampers D-2 and D-3 from the control of T-2, thereby closing outdoor damper D-2 and opening return air damper D-3.

On a rising return air temperature T-3 functions to open face damper D-4, while simultaneously closing by-pass damper D-5 as required to maintain the desired return air temperature. The reverse operation occurs on a falling return air temperature.

During all seasons except the summer cooling season, manual switch S-1 is positioned to exhaust its branch, thereby making dew-point thermostat T-2 inoperative and positions chilled water valve V-2 for continuous recirculation, positions three-way valve V-b to permit heating coil valve V-1 to be operated as required, positions three-way valve V-c to permit T-1 to control D-2 and D-3, positions three-way valve V-e to permit T-3 to operate through low limit discharge thermostat T-4 to control heater valve V-1 and positions three-way valve V-f to open normally closed face damper D-4 and close normally open by-pass damper D-5.

When switch S-1 is positioned as noted, remote bulb dew-point thermostat T-1 functions on a rising temperature to gradually open maximum outdoor damper D-2, while simultaneously closing return air damper D-3 as required to maintain the desired dew-point temperature. The reverse operation occurs on a falling dew-point temperature. When the outdoor wet-bulb temperature rises above the desired dew-point temperature, the dew-point temperature will rise accordingly until such time as the system is indexed for summer cooling and chilled water is made available to drop the dew-point temperature.

Return air thermostat T-3 functions on a rising temperature to pass air through low limit discharge thermostat T-4 to gradually close heating coil valve V-1. Should the discharge temperature fall below the operating point of T-4, this thermostat will release air from its branch to gradually open V-1 as required to maintain the desired low limit discharge temperature regardless of the operation of T-3.

CHAPTER 35

Motors and Motor Controls

Motor Rating; Functions of Motor Control Equipment; Direct Current Motors, Types, Control Equipment and Specifications; Alternating Current Motors, Types, Control Equipment and Specifications; Gear Motors; 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 prime mover. 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.

In selecting a motor for a particular application consideration must first be given to the type of power supply available. All machinery has certain load characteristics which may vary with speed. Some types may have a constant torque over wide ranges of speed, while others may have changing torques with changing speed. Consideration should be given to selecting the motor and the motor control which best suit the requirements of the drive.

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 with accompanying reduction in motor life.

In general, the electrical insulation is the portion of the motor most susceptible to injury from high operating temperatures. Of the several types of insulation which are available, the most common type, specified as Class A by the *National Electrical Manufacturers Association*, consists of cotton, felt, paper or similar organic materials and permits a 55 C rise in temperature over a 40 C ambient temperature. Class B insulation consists of mica, asbestos, fiber glass, or similar inorganic materials and permits a 75 C rise in temperature over the 40 C ambient temperature. Other types of insulation such as silicone resin are available and permit extremely high operating temperatures.

The mechanical construction of the different types of motor enclosures and the rise in temperature with Class A insulation for each type are enumerated in the glossary at the end of this chapter. Since the difference in temperature between the hottest spot and the nominal temperature, as measured by a thermometer, is greater for a completely unprotected machine than it is for an enclosed machine, the permissible temperature rise is smaller for an open motor.

FUNCTIONS OF CONTROL EQUIPMENT FOR MOTORS

In general, control equipment for all types of motors should provide: (1) means of disconnecting the motor from the power supply; (2) means

TABLE 1. CLASSIFICATION OF MOTORS

Power Supply	Type	Speed Characteristics	Full Voltage: Starting Torque	Full Voltage: Starting Current	HP Range	Type of Application See Footnote †
			Constant Speed Drives			
d-c	1. Shunt	Constant	Normal (with controller)	Normal	All	(*a*) Fans and (*c*) centrifugal pumps and centrifugal compressors
	2. Compound	Variable	High (with controller)	Normal	All	(*b*) (*c*) (*e*) Reciprocating pumps and frequent or hard starting
	3. Series	Variable	High (with controller)	Normal	Small	(*d*) Fans direct connected
Poly-phase a-c	4. Squirrel-Cage General Purpose Class A	Constant	Normal 0.8–1.5 times	High 6–8 times	All	(*a*) Fans and (*c*) centrifugal pumps and centrifugal compressors
	5. Squirrel-Cage Class B	Constant	Normal 0.8–1.5 times	Normal 5–6 times	Medium Small	(*a*) Fans and centrifugal pumps and centrifugal compressors
	6. Squirrel-Cage Class C	Constant	High 2–2.6 times	Normal 5–6 times	Medium Small	(*b*) Reciprocating pumps (*e*) and compressors started loaded
	7. 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
	8. 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
	9. 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	10. Capacitor	Constant	High	Normal	Small	(*b*) Pumps and compressors
	11. Capacitor Fan	Constant	Normal	Normal	Small	(*a*) Fans, centrifugal pumps
	12. Capacitor Start Induction Run	Constant	High	Normal	Small Fractional	(*a*) Fans (*b*) pumps and compressors
	13. Repulsion Induction	Constant	High	Normal	Medium Small	(*a*) Fans (*b*) pumps and compressors
	14. Split Phase	Constant and Adjustable	Normal	Normal	Fractional	(*a*) Fans (*b*) pumps and compressors (*d*) fans—direct

†Applications:

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.

TABLE 1. CLASSIFICATION OF MOTORS—(Concluded)

Power Supply	Type	Speed Characteristics	Full Voltage: Starting Torque	Full Voltage: Starting Current	HP Range	Type of Application See Footnote †
			Adjustable Speed Drives			
d-c	15. Shunt Field Adjustment	Constant	Normal (with controller)	Normal	All	(*a*) Fans and (*e*) centrifugal pumps
	16. Armature Resistance Adjustment	Variable	Normal (with controller)	Normal	All	(*a*) Fans and (*e*) centrifugal pumps
	17. Variable Voltage Control	Constant	Normal (with controller)	Normal	All	(*d*) Fans and centrifugal pumps
Poly-phase a-c	18. Squirrel-Cage High Slip. Transformer Adjustment	Variable	Normal	Normal	Medium Small	(*a*) Fans
	19. Squirrel-Cage Separate Winding or Regrouped Poles	Constant Multi-Speed	Normal or High	Normal or Low	All	(*a*) Fans (*b*) pumps and (*c*) compressors
	20. Wound Rotor	Variable	High (with secondary control)	Low	All	(*a*) Fans (*b*) centrifugal pumps and compressors
Single Phase a-c	21. Repulsion	Variable	High	Normal	Low and Fractional	(*a*) Fans—centrifugal pumps (*b*) compressors
	22. Capacitor Low Torque Tapped Winding	Variable Two Speed	Low	Normal	Fractional	(*d*) Fans, direct
	23. Capacitor Low Torque Transformer Adjustment	Variable	Low	Low	Fractional	(*d*) Fans
	24. Split Phase Regrouped Poles	Constant	Normal	Normal	Fractional	(*d*) Fans

for starting the motor; (3) overload protection for the motor; (4) protection against low voltage; and (5) means for varying the motor speed.

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 at 125 per cent overload, the element trip-

ping after a definite interval of time. The National Electric 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 only used 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 control device located elsewhere.

DIRECT CURRENT MOTORS

Direct current motors are classified (see Table 1) according to type of winding as: *shunt wound, compound wound,* and *series wound.*

Shunt Wound motors, being suitable for application to fans, centrifugal pumps, or similar equipment where the amount of starting torque required is relatively small, are used for the majority of direct current applications in the field of heating, ventilating, and air conditioning. They may be used on reciprocating pumps and compressors if started under unloaded conditions.

Without auxiliary control the shunt wound motor is designated as *constant speed.** Fig. 1 illustrates the characteristics of direct current motors, showing speed, horsepower, and torque as a function of current. The speed regulation* of small size shunt wound motors from ¾ hp to 5 hp is 12 per cent as specified by the *NEMA* while on larger motors it is 10 per cent.

Compound Wound motors are required for application to reciprocating compressors, stokers, reciprocating pumps when started under loaded

*Refer to Glossary at end of chapter.

conditions, and other similar equipment requiring high starting torque. The characteristics of this type of motor are such that for starting torques above full-load torque the starting current required is somewhat less than for the shunt wound motor. Compound wound direct current motors are normally used whenever frequent starting makes high starting and accelerating torque desirable. Without auxiliary control, compound wound motors are designated as *varying speed,** and have a speed regulation of 25 per cent.

Series Wound motors find only limited application in a few special cases and are available in a limited range of sizes. The motors are used where extremely high starting torques are required and must be applied only to direct coupled continuous loads due to the fact that the speed

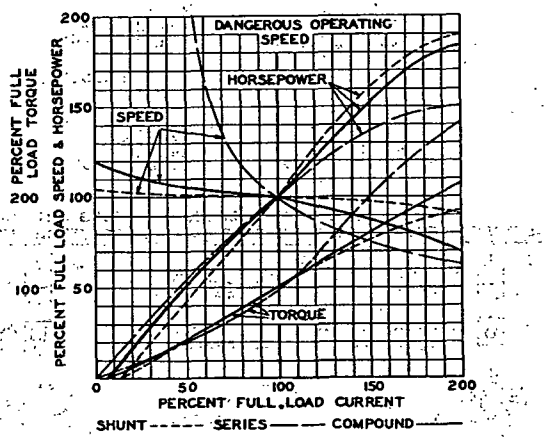


FIG. 1. CHARACTERISTICS OF DIRECT CURRENT MOTORS

of the motor becomes dangerously high when the motor is operated at a light load.

Typical d-c motor specifications are shown on page 613.

DIRECT CURRENT MOTOR CONTROL

Direct current motors are usually started through starting controllers which use a resistance, in series with the motor armature, which is gradually cut out as the motor comes up to speed. Motors up to 2 hp may be linestarted providing the inrush current causes no serious voltage fluctuations in the power supply line.

Constant Speed and Varying Speed Motors. As shown in Fig. 2, the recommended practice for manual starting of motors over ½ hp requires the use of a fused safety switch or circuit breaker and a face-plate type starter. For automatic push button starting a safety switch or circuit breaker and an automatic starter are recommended.

*Refer to Glossary at end of chapter.

Adjustable Speed Motors are normally shunt wound and are operated at various speeds by varying a resistance connected in series with the motor field. A maximum range of speed of about 5 to 1 can be obtained by this means. Rated speed regulation is 22 per cent for motors of this type from 2 to 5 hp; 15 per cent is standard in larger sizes. The *NEMA* practice on rating adjustable speed d-c motors is defined in the Glossary at the end of the chapter.

The control for the adjustable speed motor consists of the addition of a field rheostat for speed control to the equipment specified for the constant speed motor.

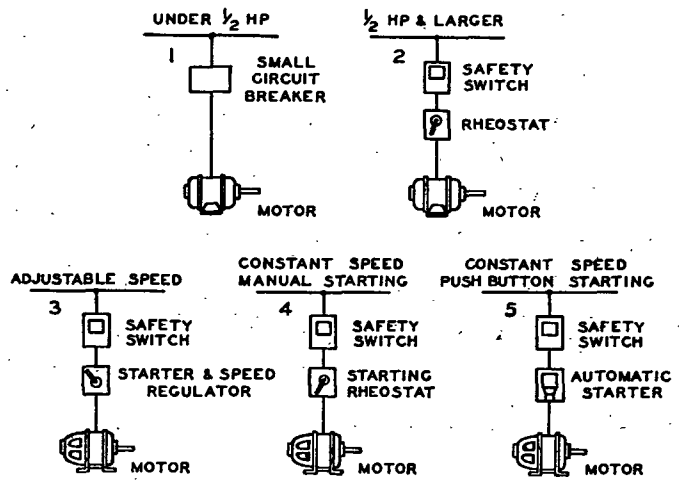


FIG. 2. RECOMMENDED CONTROLS FOR d-c MOTORS

Adjustable Varying Speed * Motors are d-c motors in which the speed is varied by the addition of resistance in series with the armature. The speed of the motor by this means is always less than the rated full field speed and varies widely with a change in load, especially with high series resistance. Fig. 3 illustrates typical speed characteristics of this type of motor for different values of armature resistance.

The addition of series resistance in the armature circuit reduces the motor speed by lowering the voltage on the armature. At one-half speed the voltage is approximately one-half of line voltage. Consequently, with rated full load current the power delivered by the motor will be only one-half of the maximum, *e.g.*, it will be 5 hp from a 10 hp motor, because the other 5 hp will be lost in the resistance. It is, therefore, evident that the efficiency of the motor is reduced at reduced speeds due to the loss in the resistor.

Control for the adjustable varying speed motor is similar to that for the constant speed motor with the exception that the starting resistor

*Refer to Glossary at end of chapter.

Typical Specifications for d-c Motors

.....Hp, ...RPM, {Constant / Adjustable / Varying} Speed, {Shunt / Compound / Series} Wound, {115 / 230 / 600} Volts d-c Motors For Driving ..... (Application) Motor Shall Be Arranged For {Horizontal / Vertical} Mounting And Shall Be Provided With A ..... (Open, Splashproof, etc.) Type Of Enclosure, *NEMA* {Class A / Class B} Insulation, And {Ball / Sleeve} Bearings.

Typical Specifications for d-c Motor Control

Control For {Constant / Adjustable / Adjustable Varying} Speed d-c Motors Shall Be {Manual / Magnetic} And Shall Consist of a {Safety Switch / Circuit Breaker} And an Enclosed Controller Providing Overload Protection [And Low Voltage {Protection / Release}].

Typical Specifications for Squirrel-Cage Motors

.....Hp, ...RPM, {25 / 60} Cycles, {208 / 220 / 440 / 550} Volts, {2 / 3} Phase, Squirrel-Cage Induction Motors of the *NEMA* {Class A—Normal Starting Torque, Normal Starting Current / Class B—Normal Starting Torque, Low Starting Current / Class C—High Starting Torque, Low Starting Current / Class D—High Slip} Type Driving ..... (Application) Motor Shall Be Arranged For {Horizontal / Vertical} Mounting And Shall Be Provided With A ..... (Open, Splashproof, etc.) Type of Enclosure, *NEMA* {Class A / Class B} Insulation, And {Sleeve / Ball} Bearings.

must be designed for speed regulating duty which means that it must be capable of carrying the motor current continuously. Speed controllers are available both for constant torque applications and varying torque drives, such as required by fans, in which the torque is reduced considerably at reduced speed.

The *Adjustable Voltage* type of speed control is also often known as the variable voltage or Ward-Leonard system. For machines requiring a wide range in speed control and a large number of steps of control this type of system is used most extensively. The drive consists of one or more d-c motors, the armatures of which are supplied with power from a d-c generator and the fields of all machines are excited from a constant voltage exciter. A schematic diagram of connections for an adjustable voltage drive is shown in Fig. 4. In most cases the d-c generator is a part of a three-unit set including a constant speed a-c driving motor, and a constant voltage exciter. As the voltage on the generator and consequently on the d-c motor or motors is adjusted by a rheostat in the

generator field circuit, a great many steps are thus obtained in an efficient manner. With constant field excitation on the motor the speed of the motor will vary approximately as the voltage on the generator.

Extremely wide speed ranges are possible with the adjustable voltage type of drive. Ranges as high as 10 to 1 are common and, by the addition of field control on the motors, ranges as high as 40 to 1 are permissible. This type of drive provides the advantage of good speed regulation over the entire speed range, as shown in Fig. 3 in which this type of drive is compared with the adjustable varying speed drive.

Typical specifications for d-c motor control are shown on page 613.

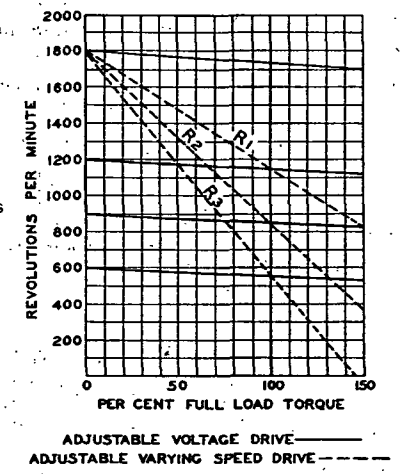


FIG. 3. SPEED TORQUE CHARACTERISTICS OF ADJUSTABLE VARYING SPEED DRIVE AND ADJUSTABLE VOLTAGE DRIVE

ALTERNATING CURRENT MOTORS

Alternating current motors are divided into two main classifications: *polyphase* and *single phase* (see Table 1), according to the type of power supply used. They are further subdivided as to the 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 Class B squirrel-cage motor. In addition, the polyphase motor has the advantages of higher power factor and 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* in classes providing a variety of speed and torque characteristics. Class A motors provide normal starting torque at normal starting current and are suitable for constant speed application; to equipment such as fans and blowers, in which *starting current need not be limited.* Class B motors provide normal starting torque at low starting current and are used for the same type of application as Class A *where starting current must be limited.* Class C

motors provide high starting torque with low starting current and are used on compressors, started without unloaders, and on reciprocating pumps. Class 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.

Figs. 5, 6, 7, and 8 illustrate the characteristics of squirrel-cage motors. It will be noticed by inspection of Fig. 6 that both power factor and efficiency are improved if the motors are operating as near rated load as possible. In addition, as shown in Fig. 8, power factor and efficiency are better for higher speed motors.

Typical specifications for squirrel-cage motors are shown on page 613.

Wound Rotor motors are used for applications requiring high starting torque at low starting current, because a wound rotor motor with its

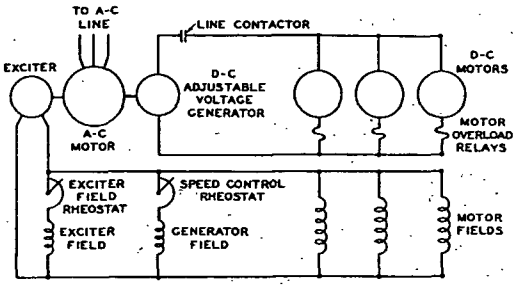


FIG. 4. COMPONENT PARTS OF AN ADJUSTABLE VOLTAGE DRIVE

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. Typical specifications for wound rotor motors are shown on page 616.

The addition of resistance to the secondary winding of the wound rotor motor changes the speed torque characteristics as indicated in Fig. 9. 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 to values below 50 per cent.

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 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

*Refer to Glossary at end of Chapter.

Typical Specifications for Wound Rotor Motors

.....Hp, ...RPM, {25 / 60} Cycles, {208 / 220 / 440 / 550} Volts, {2 / 3} Phase, Wound Rotor Induction Motors For Driving.....(Application) Motor Shall Be Arranged for {Horizontal / Vertical} Mounting And Shall Be Provided With A.....(Open, Splashproof, etc.) Type of Enclosure, *NEMA* {Class A / Class B} Insulation and {Ball / Sleeve} Bearings.

Typical Specifications for Synchronous Motors

.....Hp, ...RPM, ...Per Cent P.F., ...Volts, {2 / 3} Phase, {25 / 60} Cycles, Synchronous Motors of the {Belted / Coupled / Engine} Type for Driving.....(Application) Motor Shall be Arranged for {Horizontal / Vertical} Operation and Shall Be Provided With A.....(Open, Splashproof, etc.) Type of Enclosure. Motor Shall be Capable of Developing A Starting Torque of ..... Per Cent Full Load Torque, A Pull-In Torque of ..... Per Cent Full Load Torque, And A Pull-Out Torque of..... Per Cent Full Load Torque. The Motor Field Shall Be Excited From A {Direct Connected Exciter / Belted Exciter / M-G Set Exciter / d-c Bus} Which {Shall / Shall Not} Be Included With The Motor.

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-qut 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

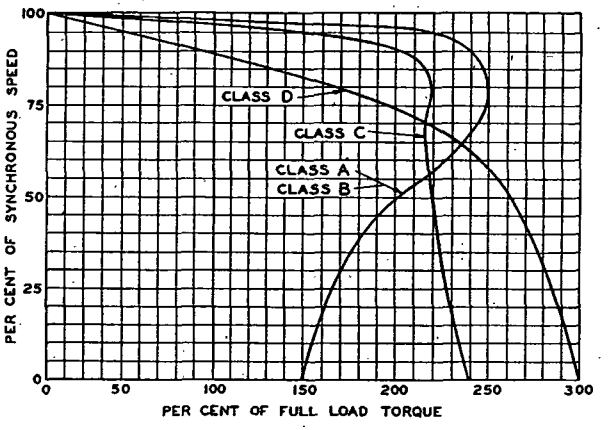


FIG. 5. SPEED TORQUE CHARACTERISTICS OF SQUIRREL-CAGE MOTORS

excitation is applied. Table 2 tabulates typical application requirements of synchronous motor drives, listing starting, pull-in, and pull-out torques.

Typical specifications for synchronous motors are shown on page 616.

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

TABLE 2. 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
Blowers	Blowing Engines Reciprocating	Engine Type	Unloaded	40	40-60	150	
Blowers	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	40	150	Flywheel Effect Important
Compressors	Ammonia and Ammonia Booster	High Speed—Belted Low Speed—Engine Type Occasionally Coupled	Unloaded (By By-Pass)	40	40	150	Flywheel Effect Important
Compressors	Freon	High Speed—Belted Low Speed—Engine	Unloaded (By By-Pass)	45	60	150	Flywheel Effect Important
Compressors	Gas Reciprocating	High Speed—Belted Low Speed—Engine	Unloaded (By By-Pass)	40	40	150	Flywheel Effect Important

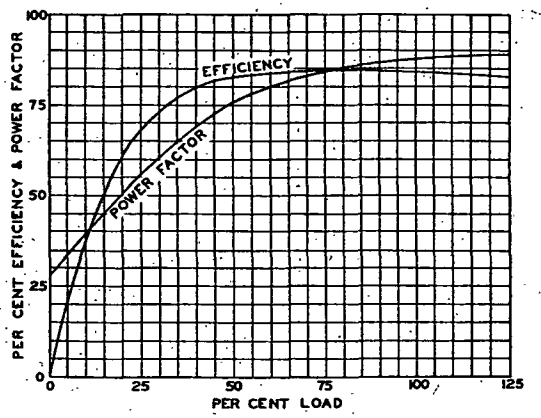


FIG. 6. VARIATION OF EFFICIENCY AND POWER FACTOR WITH LOAD FOR A TYPICAL 5 HP, 4 POLE 60 CYCLE, CLASS B, SQUIRREL-CAGE MOTOR

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

TYPE OF SQUIRREL CAGE MOTOR	STARTING TORQUE % OF FULL LOAD TORQUE 100 200 300	STARTING CURRENT % OF FULL LOAD CURRENT 200 400 600 800	MAXIMUM TORQUE % OF FULL LOAD TORQUE 100 200 300	EFFICIENCY AT FULL LOAD - % 80 85 90	POWER FACTOR AT FULL LOAD-% 80 85 90	SLIP AT FULL LOAD - % 1 2 4 6 8 10
CLASS A						
CLASS B						
CLASS C						
CLASS D			*			

* SAME AS STARTING

FIG. 7. COMPARATIVE PERFORMANCE OF SQUIRREL-CAGE MOTORS OF 30 HP AND SMALLER SIZES

winding motor 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 inherently develop no starting torque and are provided with auxiliary windings and devices for starting pur-

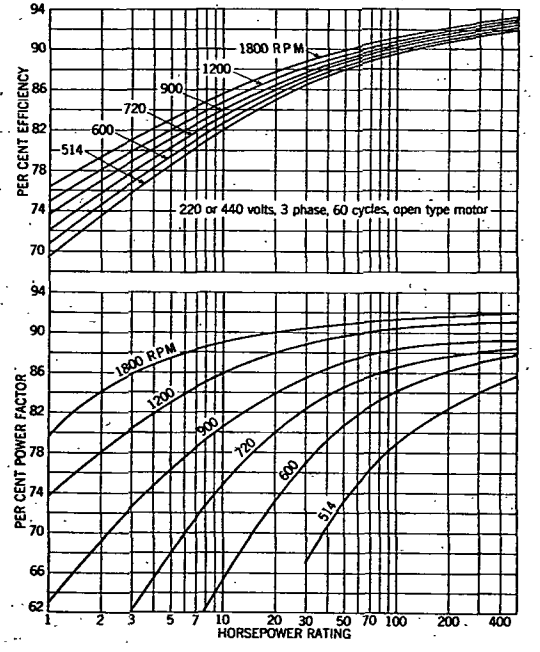


FIG. 8. EFFICIENCIES AND POWER FACTORS FOR SQUIRREL-CAGE INDUCTION MOTORS

poses. The motors are classified as to the type of auxiliary winding provided.

Capacitor Start-Induction Run motors develop high starting torque with low starting current and are used for all types of constant speed heavy duty drives such as compressors, pumps, and stokers. During the starting period, a winding with a capacitor in series is connected in the motor armature circuit and when the motor comes up to speed a centrifugal switch cuts the capacitor and second winding out of the circuit.

Capacitor motors, which are ideally suited for small fan drives, are similar to the capacitor start-induction run type except that the capacitor is not cut out when running.

Repulsion Start-Induction Run motors develop extremely high starting torque. They are supplied with a short-circuiting switch which cuts out

Typical Specifications for Single Phase Motors

.....Hp, ...RPM, {25/60} Cycles, {115/230} Volts, Single Phase Motor of the.....Type for Driving..... Motor Shall
(Capacitor, Split Phase, etc.) (Application)
Be Arranged for {Horizontal/Vertical} Mounting And Shall Be Provided With a.....Type of Enclosure, *NEMA* {Class A/Class B} Insulation, And {Ball/Sleeve} Bearings.
(Open, Splashproof, etc.)

Typical Specifications for Squirrel-Cage Motor Control

Control for Squirrel-Cage.....Motors Shall Consist of An Enclosed
(Fan, Pump, etc.)
{Manual/Magnetic} Type {Across The Line/Reduced Voltage} Starter Providing Overload Protection And Low Voltage {Protection/Release} Control Shall Include A Safety Disconnect Switch {Separately Mounted/In a Common Enclosure} With the Starting Controller.

Typical Specifications for Wound Rotor Motor Control

Control for Wound Rotor Motor Control for.....Applications Shall Consist of a Safety
(Fan, Pump, etc.)
Disconnect Switch, a {Separately/Common} Mounted Across the Line Starter Providing Overload Protection, Low Voltage {Protection/Release}, and a Secondary {Starting/Speed Regulating} Controller for {Separate Mounting/Mounting in Same Enclosure}. Primary and Secondary Control Shall Be Interlocked so as to Provide Complete Control from the Rheostat Handle.

the commutator when the motor comes up to speed. These motors are suitable for applications such as industrial compressors where high break-away torque is required 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

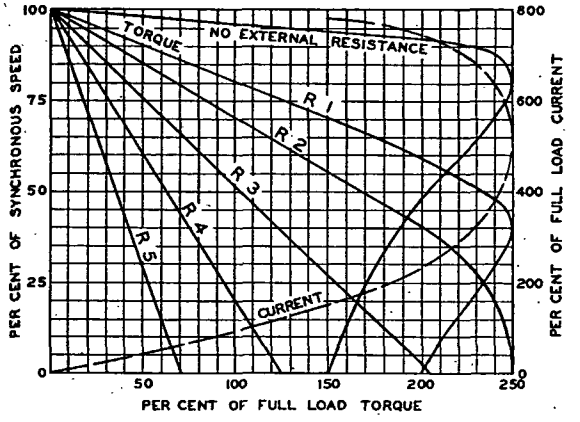


FIG. 9. PERFORMANCE CHARACTERISTICS OF A WOUND ROTOR MOTOR WITH EXTERNAL RESISTANCE

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. 10.

Typical specifications for single phase motors are shown on page 620.

CONTROL FOR ALTERNATING CURRENT MOTORS

Squirrel-Cage motors are usually linestarted where power company limitations permit. In sizes up to 2 hp the motors are started by means of manual switches with an overload current element for motor protection.

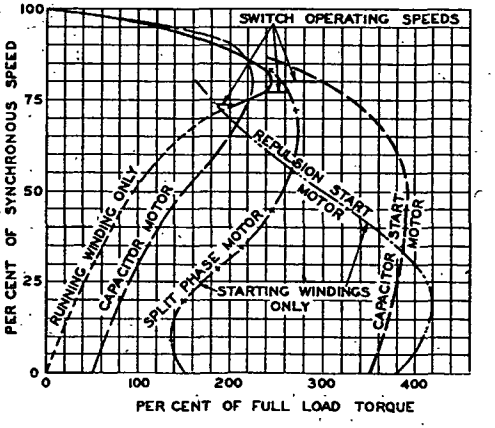


FIG. 10. SPEED-TORQUE CHARACTERISTICS OF SINGLE PHASE MOTORS.

Typical Specifications for Synchronous Motor Control

Synchronous Motor Control for _____ (Application) Motors Shall Provide for {Full Voltage / Reduced Voltage} Starting And Shall Be {Full Magnetic / Semi-Magnetic}. (Reduced Voltage Starting Shall Be Obtained By Means of {Autotransformers / Resistors / Reactors} And Shall Limit The KVA Inrush to a Maximum of ..... per cent of Full Load KVA.) The Control Panel Shall Be For {Isolated Switchboard} Assembly And Shall Be of {Open Switchboard / Enclosed Cubicle} Construction. It Shall Provide Overload, Under Voltage Protection And After Pulling Out of Step Will {Automatically Resynchronize / Automatically Stop} The Motor.

Typical Specifications for Single Phase Motor Control

Control For Single Phase _____ (Fan, Pump, etc.) Motors Shall Consist of a {Manual / Magnetic} Type {Across The Line / Reduced Voltage} Starter Providing Overload Protection (And Low Voltage); And A Separate Safety Disconnecting Switch.

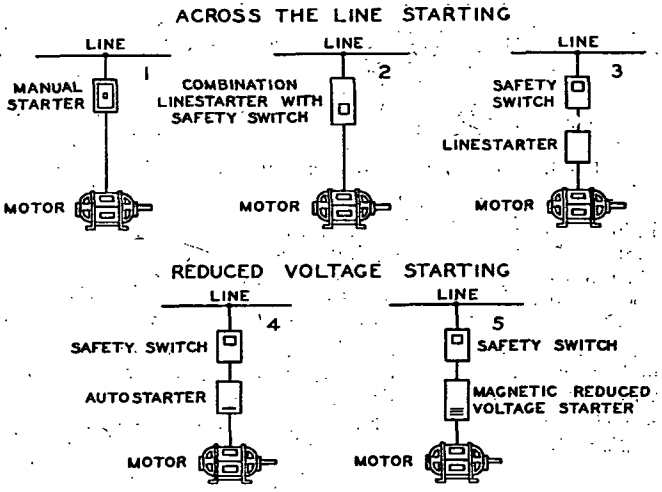
In larger ratings a linestarter is usually provided with either an additional safety switch or circuit breaker for disconnecting and short circuit protection. Reduced voltage starting may be either of manual or push button controlled magnetic type. In specifying this type of starter consideration should be given to the fact that starting torque of squirrel-cage motors varies as the square of the applied voltage. For example, a motor developing 100 lb-ft starting torque on full voltage would produce only 25 lb-ft torque on starting on half rated voltage. Fig. 11 illustrates recommended control practice for squirrel-cage motors. Typical specifications for squirrel-cage motor controls are shown on page 620.

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. Fig. 12 illustrates recommended control practice for wound rotor motors. Typical specifications for wound rotor motor control are shown on page 620.

*Refer to Glossary at end of chapter.

Synchronous motor starters should provide pull-out protection, automatic synchronization or automatic stopping of the motor after pull-out, and insurance 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.

In applying reduced voltage starters to synchronous motors it should be remembered that, since these motors are started on damper windings and during the acceleration period function similarly to squirrel-cage motors, the starting torque varies as the square of the applied voltage.



Arrangements 2, 3, 4 and 5 provide automatic push-button starting.

FIG. 11. RECOMMENDED CONTROLS FOR SQUIRREL-CAGE MOTORS

Consideration should be given to insure development of sufficient motor torque to accelerate the load. Typical specifications for synchronous motor control are shown on page 622.

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 it necessary for the operator to press the first speed button before regulating the motor to the desired speed. This insures that the motor is always started at low speed before adjusting to a higher speed.

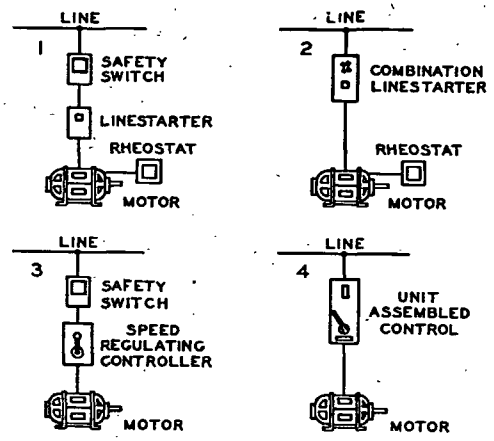
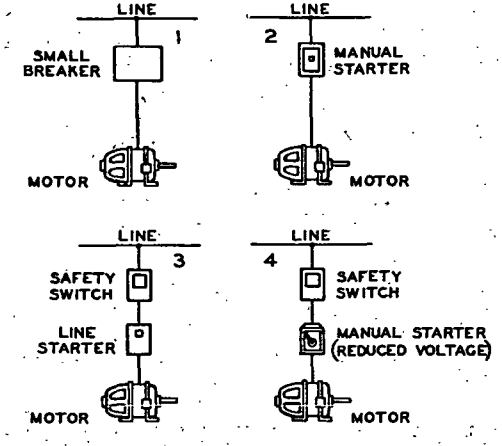


FIG. 12. RECOMMENDED CONTROLS FOR WOUND ROTOR MOTORS

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 to the motor and drive when the speed is reduced from a higher to a lower speed.

Single Phase motor control usually consists only of a linestarter, either manual or magnetic. In some cases it is desirable also to provide a disconnect switch. Fig. 13 illustrates the recommended controls.

Typical specifications for single phase motor control are shown on page 622.



Arrangements 2, 3 and 4 are optional for motors up to 7½ hp, 220 volts.

FIG. 13. RECOMMENDED CONTROLS FOR SINGLE PHASE MOTORS

GEAR MOTORS

A gear motor is a self-contained combination of any type of a-c or d-c motor and an enclosed speed-reducing gear, providing a more compact and readily adaptable unit than is obtained by using a motor coupled to a gear reducer. Gear motors are available in sizes up to 75 hp with output shaft speeds from about 4 to 1430 rpm, making it possible to couple or to connect by gear or chain to nearly any machine. High speed motors are used, generally 1800 rpm on 60 cycles, thus obtaining the advantages of high motor power factor and efficiency. The gearing efficiency is also high, usually about 98 per cent for a single reduction of the helical or spur type, that is, a 2 per cent loss for one reduction or 4 per cent loss for a double reduction. Consequently, the over-all performance of the gear motors is much higher than a combination of open gearing, belting, countershaft, or other arrangement, which would otherwise be required. Gear motors are used extensively to drive numerous types of slow speed drives. Besides being more effective than other combination drives in saving space, they are important in reducing maintenance and operating hazards.

GLOSSARY

General Definitions

NEMA is the abbreviation for the *National Electrical Manufacturers Association.*

Speed Regulation (d-c motors) is the change in speed between no-load and full-load. expressed in per cent 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 5.6 per cent.

Slip (a-c induction motors) is the difference between the motor speed and synchronous speed expressed in per cent of synchronous speed, *e.g.*, a 1200 rpm motor operating at 1140 rpm would have a slip of 5 per cent.

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 of Motor Enclosures

Open motors (40 C rise) are self-ventilated machines having no restriction to ventilation other than that necessitated by mechanical construction.

Protected motors (50 C rise) have all ventilating openings in the frame protected by perforated covers.

Semi-Protected motors (50 C rise) have the ventilating openings in the top half of the frame only protected by perforated covers.

Drip Proof motors (50C rise) are so constructed that drops of liquid or solid particles falling on the machine at any angle not greater than 15 deg from the vertical cannot enter the machine either directly or by striking and running along a horizontal or inclined surface.

Splash Proof motors (50 C rise) 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 deg from the vertical cannot enter the machine either directly or by striking and running along the surface.

Totally Enclosed Non-Ventilated motors (55 C rise) are so constructed as to prevent exchange of air between the inside and outside of the case.

Totally Enclosed Fan-Cooled Motors (55 C rise) are similar to totally enclosed, non-ventilated machines except that exterior cooling is provided by means of a fan or fans integral with the machine.

Explosion Proof motors (55 C rise) have an enclosing case designed to withstand an explosion of a specified gas or vapor which may occur within it, and to prevent the

ignition of the gas or vapor surrounding the motor by sparks, flashes, or explosion of the gas or vapor which may occur within the machine casing.

Water Proof motors (55 C rise) are so constructed as to exclude water applied in the form of a stream from a hose.

Dust Tight motors (55 C rise) are so constructed that the enclosing case will exclude dust.

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 per cent 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 per cent 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 per cent 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, agitators, and so forth. 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 36

Unit Air Conditioners, Unit Air Coolers, Attic Fans

Definition of Types, Unit Air Conditioners, Heating, Humidifying, Cooling and Dehumidifying, Filtering, Ventilating, Types of Units, Application, Ratings, Unit Air Coolers, Design and Performance, Types of Units, Ratings, Defrosting, Economics, Attic Fans

AN assembly of functional elements, as indicated by the name, comprises the unit air conditioner or unit air cooler. Such a unit when complete in itself, employing its own direct means of air distribution and source of refrigeration is known as a self-contained unit. When used in various combinations with remote sources of refrigeration, heat or air supply, it is termed a remote unit, indicating that the source of refrigeration is not contained within the unit cabinet. Either the self-contained or remote type units may be located within or without the conditioned area, and are a separate classification from the central plant type of system, as described in Chapter 43.

The code, Standard Method of Rating and Testing Air Conditioning Equipment [1], defines the various types of unitary equipment:

1. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.

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.

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.

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 [2] 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 outside, ventilation by exhausting room air to the outside, 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.

6. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

7. A *Forced-Circulation Air Cooler* is a factory encased assembly of elements by which heat is transferred from air to evaporating refrigerant [3].

UNIT AIR CONDITIONERS

This equipment takes the form of an encased assembly including the apparatus necessary to perform either some or all of the functions of cooling, dehumidifying, filtering, ventilation, air circulation, heating, and humidifying. Control of air conditions is provided by manual switches, manual dampers, and automatic devices, in combination. The controls are either mounted on the units, or in some suitable location in the conditioned area. See Chapter 34 for a discussion of controls.

The various conditioning elements and their functions, which produce the required effects on air, are discussed under separate headings.

Heating

Heating is usually accomplished by means of a heating coil in the unit,

supplied with steam or hot water from an external source. Electric strip heaters may also be considered, where installation and operating costs justify their use. They are often used in special control applications. Reverse cycle heating as described in Chapter 39 has been developed as a feature in some unit air conditioning equipment, but this form of heating has not yet been generally adopted.

Humidifying

Adding moisture to the air involves the absorption of heat, by the humidifying water, for conversion to water vapor. Heat may be supplied by heating the humidifying water, or supplied from the air to be humidified. In the latter case the air may be warmed or the water finely divided to present a large evaporating surface to the existing air. The source of heat may be from electricity, steam or hot water coils, or from a heat transfer surface as in a direct-fired unit. Occasionally vapor is added directly by means of steam jets but this is usually confined to industrial applications because of the presence of some odor from the steam. Methods and types of humidifying apparatus are dealt with in detail in Chapter 37.

For unit application, some of the available methods are the spray nozzle, or the atomizing nozzle, the impact-jet, the drip screen and the evaporating pan. The first method is usually employed where humidification on a large scale is desired, as with large remote industrial units. Eliminator plates are necessary and the water supply may be recirculated by a pump or wasted to the drain. The drip screen, impact-jet, evaporator pan or small atomizing jets are used when humidification is desired in the smaller remote and self-contained units.

Wetted surfaces exposed to the air stream and utilizing the capillary action of water on porous substances such as fabrics and ceramics, are used for adding moisture to air. Frequent cleaning or replacement is necessary to avoid closing of the pores and to maintain freedom from odors or growths on the humidifying element.

Cooling and Dehumidifying

These two functions of air conditioning are usually performed simultaneously, although both may be done separately. For example, air may be dehumidified or dehydrated without sensible cooling by the process of adsorption. Sensible cooling of air may be accomplished without dehumidifying by maintaining the cooling surface temperature above the dew-point temperature of the air to be treated. Chapter 7 explains these fundamentals in detail.

Unit equipment commonly utilizes heat transfer surface such as pipes or coils, through which a cooling medium such as water or a refrigerant, is circulated. Types and methods of generating cooling mediums are covered in Chapter 39. Brine or water sprays may be used if desirable, either separately or in combination with coil or pipe surface. However, these methods find their best application in the larger, remote type units. Surface temperature and area of the coil or pipe, air volume and velocity, and spray temperature and volume are some of the controlling factors in unit air conditioner design and application. Ice as a cooling medium is practical but is seldom used in connection with units.

Filtering

Cleaning of outside or recirculated air discharged to the conditioned area is one of the important functions of air conditioning, and air filters

should be included on all units which condition air for comfort. Protection is also afforded the cooling and heating coils as well as the condenser coil in the case of the air cooled type.

Means of filtering may vary from the lint screen to the electrostatic filter, with the degree of efficiency covering a wide range. Inexpensive throw-away type filters lend themselves well to the compact design of the unit conditioner. Air cleaning devices form the subject of Chapter 33.

Ventilating

Provision for the introduction of outside air should be an essential part of unit conditioner design. Odors and air vitiation are avoided and better load control is possible when a positive means of introducing outside air is available.

On the small, air cooled room units, it has been found practicable to control the air in such a way as to permit variation from all recirculated air to 100 per cent outside air. An added feature is a dampering arrangement whereby it is possible to exhaust air from the room to remove smoke and generally ventilate the space.

A code [4] sponsored by a Joint Committee of the A.S.H.V.E. and the *American Society of Refrigerating Engineers* may be consulted, although it should be realized that individual applications may often show a need for ventilation in excess of these minimum requirements.

Types of Units

Unit air conditioners fall into two general classifications, depending on the location of the refrigeration source. Those units having the condensing unit completely enclosed in the same cabinet as the evaporator are known as self-contained types, while those units having the condensing unit remotely located from the evaporator and requiring piping of refrigerant from the condensing unit to the evaporator and return are known as remote types.

Self-contained units are further divided into two groups in accordance with their condensing mediums, being either air-cooled or water-cooled. Evaporative cooled types are included in this latter class.

The air-cooled types are small in capacity, ranging from ⅓ to 1½ hp. Their principal application is for conditioning such spaces as hotel rooms, offices and residential living quarters. A duct connection between the unit and an outside window or ventilated air shaft is required to permit disposal of the heat extracted from the conditioned area. The unit may stand in front of the window or be mounted on the window sill. Various styles and types of windows are encountered which increase the difficulty of making the window connections. The evaporation of condensate on the condenser coils, as a means of disposing of this moisture, tends to increase the condensing capacity and reduce the operating head pressure. Some units add supplementary water so that increased capacity may be obtained from constantly wetted condenser coil surface. Connections to an electrical outlet may be by means of a conventional cord and plug or a permanent electrical connection, depending on local code rulings pertaining to the installation of small motors. The exterior finish of the unit in metal, wood or fabric is decorated to harmonize with office or bedroom furnishings.

A unit of the air-cooled condenser type for floor mounting is shown in Fig. 1. Of the two fans shown, the lower one acts as condenser air fan, and in some units this fan is arranged with slingers for discharging condensate on the condenser coil while the upper fan discharges air into the

conditioned area. A feature of the design shown in Fig. 1 is that the condensate from the cooling coil is sprayed over the condenser surface and vaporized, thus eliminating the need for drain connections. A simple dampering arrangement is generally provided for exhausting some air from the room, in addition to introducing outside air and recirculating required amounts of air. It is possible to remove the equipment for winter storage or utilize the ventilating features for winter operation.

Water or evaporative-cooled units start at 1 hp in size and may be as large as 30 hp. Condensers requiring water use either city water, well water or recirculating water from a cooling tower. Evaporative condensers are seldom used on units under 5 hp.

The heat generated by the compression of refrigerant gases and that given off by the electric motor is removed from the compressor compartment in four ways: by the use of a water coil in the compressor compartment; by means of utilizing the cold suction gases; by drawing part of the return air through the compressor compartment and finally by

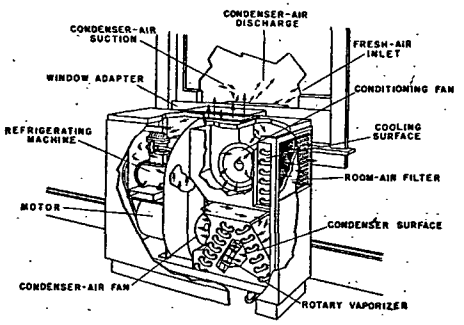


FIG. 1. SELF-CONTAINED AIR-COOLED UNIT AIR CONDITIONER

circulating room air through the compressor compartment by means of a fan attached to the motor shaft.

The smaller water-cooled units have somewhat the same application as the air-cooled units. Water and drain facilities must be available and, although window connections are not required for heat disposal, it is desirable to have an outside air connection for ventilation purposes. Electrical wiring is generally permanently connected.

Up to 7½ or 10 hp in size the units are usually styled for locating directly in the conditioned area. Above this size the tendency is to locate the equipment adjacent to the conditioned area, with supply and return duct connections. Compressor compartment heat is removed by the same methods described previously.

On water units of the vertical type of 5 hp and under, use is made of air distributor headers, equipped with directional louvers, on one or more sides, for discharging the air directly into the conditioned space in which the unit is located. Above 5 hp this method usually becomes impractical and a system of ducts is employed. Arrangements for outside air supply are similar to those in central plant design, although simpler, hand operated dampers are usually employed.

Accessory equipment for the self-contained unit conditioners includes heating coils, humidifiers, and controls for utilizing the unit for winter

heating and ventilating. Refinements of dust and odor control and constant temperature and humidity regulation are considered to be special application problems with this type of equipment.

A typical unit with water-cooled condenser is illustrated in Fig. 2. The header arrangement permits air distribution in several directions, and in such a way as not to fall on the room occupants. All side panels are removable for complete access to equipment. It is only necessary to bring water, drain and electrical service to the unit, and a source of heat if desired.

Remote type units cover a much broader range of size and application and generally are used in connection with or in lieu of a central plant

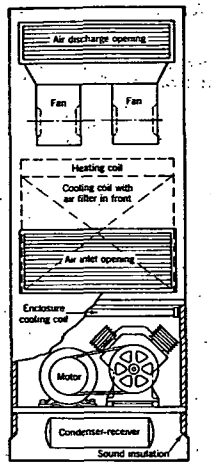


FIG. 2. SELF-CONTAINED WATER-COOLED AIR CONDITIONER

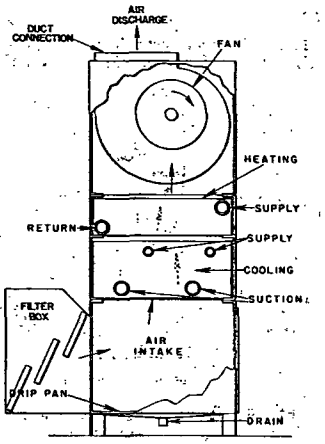


FIG. 3. VERTICAL REMOTE TYPE UNIT AIR CONDITIONER

system. They may be used individually or in groups in the place of self-contained equipment.

Without the weight of the refrigerating equipment, these units may be suspended from roofs or ceilings, or located wherever space permits on roofs and in basements, or in the conditioned area itself. All combinations of filtering, humidification, cooling and heating may be employed, with control as elaborate or as simple as is required. The remote unit is particularly adaptable where a variety of application conditions is to be met from a single source of refrigeration, such as the modern industrial plant which may have a laboratory, executive offices, a cafeteria, together with various processing departments.

The cooling and heating mediums, consisting of a refrigerant, chilled water or brine, and steam or hot water, are piped to the units, and transfer takes place by means of coils, or in the case of air washers, by means of water or brine sprays in the air stream, or a combination of the two.

The floor mounted or vertical type and the suspended or horizontal type of remote units are respectively shown in Figs. 3 and 4. The cabinets are generally of sheet steel, insulated to prevent heat transfer, finished to prevent corrosion and suitable for applying decoration if desired.

A spray type remote unit is illustrated by Fig. 5. Many designers prefer the air washing and coil wetting features. Air is circulated by means of attached or built-in fans, delivering the conditioned air through a system of ducts, which include outside air connections if desired.

Some types of units, mostly those suitable for suspension, employ a propeller fan such as in Fig. 6. These are located in the conditioned area, are without ducts and seldom do more than cool and dehumidify, since the propeller fan is not designed to move air against high pressure. Generally these units are provided with a lint screen instead of filters to limit resistance to air flow.

Individual floor mounted type remote units are available for use in offices or hotel rooms. Similar in appearance but somewhat smaller in size than the self-contained room unit, this form of equipment may be grouped:

1. Mechanical, year 'round type containing blowers, filters, humidifiers and coils, with outside air connection. Manual or automatic controls provided with each unit.

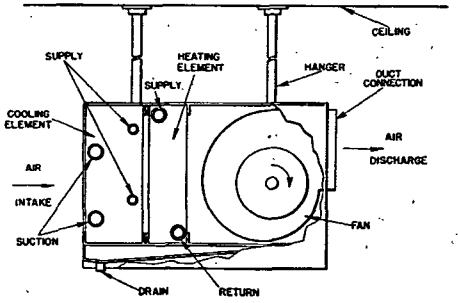


FIG. 4. HORIZONTAL REMOTE TYPE UNIT AIR CONDITIONER

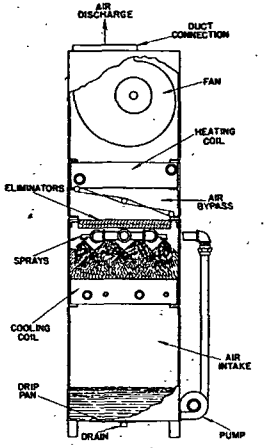


FIG. 5. (*right*) SPRAY TYPE REMOTE UNIT AIR CONDITIONER

2. Mechanical, semi-year 'round type with no outside air connection, containing blowers (filters optional), and coils. Usually used where there is an existing radiation or ventilating system. Controls provided with each unit.

3. Non-mechanical type containing coils, using air ejected under pressure from a remote source for inducing circulation over coils (see Chapter 43). Manual or automatic control of air temperature only is provided. Both summer and winter air conditioning functions may be performed by the one unit.

One coil for cooling and heating may be provided or a single coil used, through which hot water in winter and cold water in summer is circulated. All three types of systems require remote sources of refrigeration, with the first group obtaining outside air from the window, the second group having no outside air unless used in connection with a central plant or ventilating system and in the case of the third group, all air delivered under pressure is outside air.

Various combinations or alterations of these room units are available. Different filtering, humidifying and air delivery methods are employed to achieve the desired conditions. A typical remote floor type room unit air conditioner is shown in Fig. 7.

Application

In the application of unit air conditioners it is important to consider several factors:

1. Location of equipment.
2. Air distribution.
3. Multiple units versus central station system.
4. Multiple remote unit system versus a self-contained unit system.
5. Methods of control.
6. Methods of conserving water.
7. Code limitations.

In choosing locations for air conditioning units, consideration must be given to the characteristics and use of the conditioned space; type of system contemplated; duct locations; sources of power, water, refrigera-

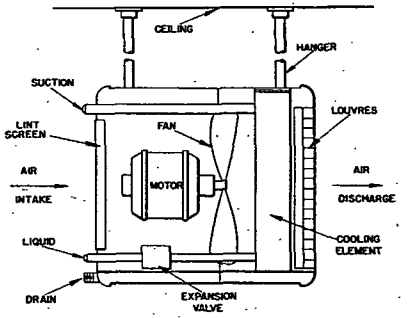


FIG. 6. SUSPENDED PROPELLER FAN TYPE UNIT AIR CONDITIONER

tion, heating and drainage; and accessibility of equipment and system for maintenance.

Where units are placed within the conditioned area, particular attention must be given to air distribution, sources of outside air and convenience to service sources and facilities previously noted. Skill and ingenuity are required to produce a neat appearing, inconspicuous job without sacrificing quality from an engineering point of view. This may be accomplished by building the units into the woodwork or walls, and refinishing them to match existing furniture and fixtures. Where location of equipment outside of conditioned space is possible, use may be made of storage rooms, halls, basements or any space less valuable than that to be conditioned. Less emphasis need be placed on the appearance of equipment.

The choice of equipment location is frequently influenced by the relative cost of long runs of insulated ducts required in order to locate units near refrigeration sources, as compared with short duct runs but long extensions of service facilities. Care must be taken that equipment will not be damaged by climatic conditions.

Air distribution from units located within the conditioned area is by means of grilles, either fixed or adjustable and mounted in air distribution headers, either furnished with the unit or constructed at the installation to meet the needs of the application under consideration. A system of ducts and distribution grilles may also be used, *similar* to the arrangement

used when equipment is outside of the conditioned area. The problem of securing a supply of outside air is often a difficult one. The proper design of ducts is of major importance for good air distribution. Chapters 40 and 41 are devoted to these subjects.

In the analysis of any large structure to be air conditioned which is divided into small spaces, such as an office building or hotel, a comparison should be drawn between the use of units and a central plant system. This study is one of economics, and should include such factors as first cost, installation costs, obsolescence, depreciation, maintenance costs, return on investment, flexibility, time of installation and possible loss of useful space during installation.

This same analysis should be extended to include a comparison of remote units with self-contained units. Self-contained units are possible where short term leases are involved, or where wiring can readily be brought to location or where existing water and drain facilities are adequate to handle the increased demand of water-cooled models. On

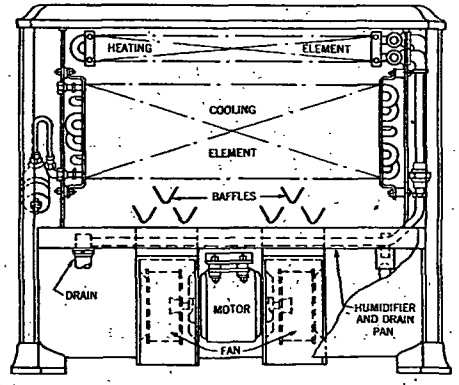


FIG. 7. REMOTE FLOOR TYPE ROOM UNIT AIR CONDITIONER

the other hand, remote type room units contain less mechanical machinery, avoid the operating cost penalty of an air-cooled condenser, as in the case of the self-contained, air-cooled room units, and seldom require heavy wiring to handle the fan load. In general, the smaller remote units are more suitable for use with new construction, where the units will probably remain during their useful life. This condition may change where larger, ceiling mounted units are compared with the floor mounted self-contained units. Controls are essential in unit application and range from the simple snap switch of a room unit to elaborate means of controlling temperature, humidity and air movement in laboratory and testing room application. The criterion of a well controlled installation is one which has neither too few nor too many controls. (See Chapter 34.)

With the expansion of cities during the past decade, the problem of water supply has become a costly and ever present problem[5]. Consequently, laws are appearing designed to restrict the usage of water wherever possible, particularly when substitute means are available. Evaporative condensers and cooling towers (described in Chapters 37 and 39) are means devised to save large quantities of water in this connection.

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.
Cooling	Water-Cooled, Air-Cooled and Evaporatively Cooled Condensers	b	Unit Ambient and Air Entering Room—Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 80 F 67 F
		c	Ventilation Air	See Note
	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 Outside Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 95 F 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 (2) Water In (3) Water Out	 16.7 lb per sq in. abs. 180 F 160 F
Humidifying	All Types Provided with Humidifying Function	i	Unit Ambient	70 F
		j	Total Air Entering Unit (1) Dry-Bulb (2) Wet-Bulb	 70 F 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).

Most manufacturers furnish equipment designed for use in combination with these water savers.

Municipal plumbing, heating, electrical and refrigeration codes, as well as fire underwriter restrictions, are likewise having their effects on the application of unit air conditioners. Meeting municipal and national code requirements should be an important item in connection with the installation of any unit equipment.

Ratings

There are two codes governing the rating and testing of unit air conditioners. The first code, Standard Method of Rating and Testing Air Conditioning Equipment[1], covers all types of air conditioning units except the self-contained type. The latter is covered by the second code, The Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling[2]. The two codes are necessary because of the basic difference caused by the heat given up by the self-contained units.

The standard rating conditions for self-contained unit air conditioners, as given in the code, are set forth in Table 1.

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 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 to those in Table 1 except the entering wet-bulb temperature for cooling is expressed as 50 per cent 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

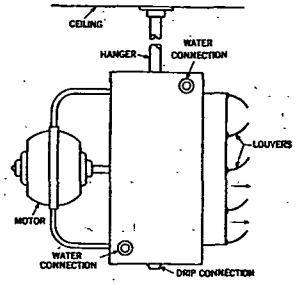


FIG. 8. CEILING TYPE UNIT AIR COOLER

self-contained units as immaterial in the rating of a unit that includes the evaporator and condensing unit.

UNIT AIR COOLERS

This type of unit is primarily intended to perform the main function of cooling air, with humidity control a secondary function within the limitations of the design. The main application of this equipment is in process and product refrigeration such as cold storage warehousing, fruit and vegetable packing, in breweries and in wholesale and retail food markets; although some comfort cooling may be obtained by the use of a unit similar in design to this type of unit as previously explained and as illustrated by Fig. 6.

Application of the unit method of air cooling with mechanical circulation is comparatively recent, being an improvement over the pipe or finned coil, which depended on gravity for circulation. Bunkers were sometimes constructed around the coils to direct the air flow and sometimes fans were used for forcing air over the coils. The location of the unit air cooler is usually within the refrigerated area, but the larger, blower type models may be remotely located.

Design and Performance

Greater application and use of commercial refrigeration have resulted from the development of the unit air cooler. Flexibility of design has permitted almost any condition to be met. By varying such physical features as the method of introducing the refrigerant into the coils, the depth of coil rows and area of their surfaces, and the air volume over the

coils, the designer is able to produce a wide range of performances and to offer many desirable features not obtainable with the coil and bunker method. More uniform temperatures, high relative humidity, moderate first cost and a minimum of installation expense are likewise factors in their development.

New uses have appeared for unit air cooler application in industrial and commercial processes involving both the raw materials and finished product, where the maintenance of low temperatures is a necessary part of these processes. Of particular interest is the new field of extreme low

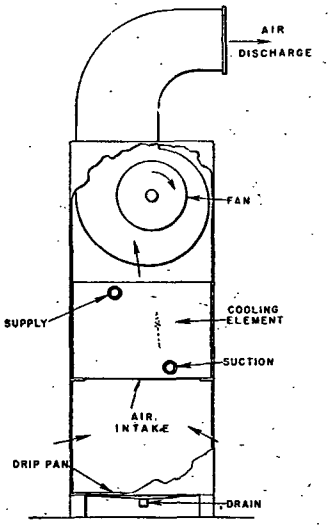


FIG. 9. SURFACE TYPE COOLING UNIT

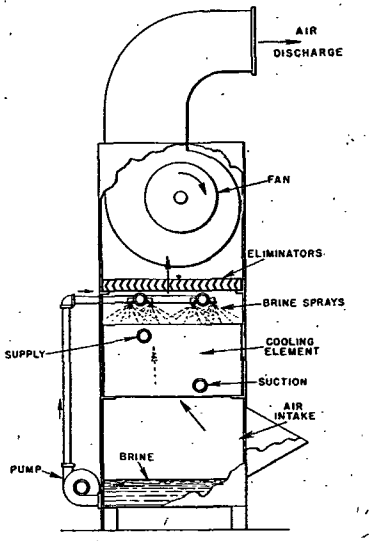


FIG. 10. BRINE SPRAY TYPE COOLING UNIT

temperature application where many new uses for refrigeration are being found.

Types of Units

The two standard types are the suspended or ceiling type, and the vertical or floor mounted type. There are variations of this such as the panel type which is wall mounted and arranged to take in air from the lower section and discharge it from the upper section.

The ceiling type has the appearance of a unit heater, with its propeller type fan blowing air through a bank of coils as shown in Fig. 8. Singly or in combination, they are easily installed and occupy little or no useful space. Alterations may be accomplished with little cost by relocating units or adding additional ones for increased capacity.

The floor mounted types employ blower type fans, as their air deliveries are higher and their locations may be remote from the space to be refrigerated. This type of unit is illustrated by Fig. 9. Air velocities and volumes must be designed for the individual application. This type of unit may employ a pump to spray a eutectic solution over the coils for the purpose of avoiding frosting as shown in Fig. 10.

Ratings

In order to rate and test equipment of this kind which normally operates below the frost line, a proposed code, Standard Methods of Rating and Testing Forced-Circulation Air Coolers for Commercial and Industrial Refrigeration [3], has been issued. This proposed standard covers only the modifications of the Standard Method of Rating and Testing Air Conditioning Equipment [1], as it is related to the different applications of unit air coolers. In this standard, the gross cooling effects are taken since the motor power input equivalent is to be computed as part of the load.

From this code Table 2 is abstracted to show standard rating conditions for forced-circulation air coolers. Other modifications take into consideration the effect of frost formation on the coils and the change in length of test runs required to meet such conditions.

TABLE 2. STANDARD RATING CONDITIONS FOR AIR COOLERS

GROUP NO.	ENTERING DRY-BULB TEMPERATURE, F DEG	ENTERING RELATIVE HUMIDITY, PER CENT	EVAPORATING TEMPERATURE, F DEG
I	45	85	30
II	35	85	25
III	35	85	15
IV	0	85	−10
V	0	85	−20

Defrosting

Unit air coolers are often required to operate in rooms where air and refrigerant temperatures are below the freezing point. This results in the freezing of the condensation on the coils and this accumulation of frost builds up to such an extent that there is a loss of capacity. This deposit is removed from the coils by the process known as defrosting which may be accomplished by several methods:

1. Where the room temperature is above the freezing point, the flow of refrigeration to the coils is halted and the fan continued in operation until the coils are defrosted.

2. The hot gas defrosting method is accomplished by a valving arrangement whereby hot compressed gases from the compressor are pumped directly into the evaporator. This operation is continued until defrosting is complete, when the system is returned to normal operation.

3. Where brine is used as a refrigerant, hot brine may be circulated through the coils.

Where the frosting is particularly heavy it is sometimes more advisable to apply the source of heat externally. This principle of defrosting is accomplished by:

4. Defrosting by warm air is accomplished by dampering arrangements which permit the cooler fan to draw warm air from a source outside of the refrigerated space, pass it over the coils and discharge it outside of the refrigerated space.

5. Constant wetting of the coils with a brine or eutectic solution prevents the formation of frost. This is accomplished on a vertical unit air cooler as shown in Fig. 10.

6. Electric heating elements, placed in such a manner that the fan forces the heated air over the coils.

7. Water sprays, placed so that the coil is thoroughly wetted during the defrosting process, are a simple means of defrosting. City water at ordinary pressures is used. The fan is not running during this operation.

Those systems using method 5 have the problem of removing condensation from the eutectic solution. One way is to waste the entire charge when dilution has rendered it ineffective. Another method employs a device which boils off the water and returns the eutectic solution to the system for further use.

ECONOMICS

In the planning and designing of a unit system or in comparing units with central plant application, a systematic approach to the problem should be made from the economic viewpoint.

First Costs. The question of first cost is but one factor in the economic approach of a skillful designer or an intelligent buyer.

1. *Equipment.* The use of a large percentage of factory fabricated equipment to maintain installation costs at a minimum, as represented by self-contained units, should be contrasted with the use of systems with a large percentage of installation labor and material, such as a central station system. The remote unit offers a compromise between these two.

2. *Installation costs.* The influence of existing codes and ordinances, and installation conditions, involving new construction or treating an existing structure deserve careful consideration. Methods which are suitable for new construction often may not be applied on buildings erected in the past. Sprinkler system rearrangements, fire doors and dampers, restrictions on multiple direct expansion units, rewiring or increasing service, cutting or reinforcing of ceilings and roofs are some items which should receive attention in this respect.

Operation and Maintenance. Tenants or occupants usually operate the self-contained equipment, which also lends itself readily to contract service for maintenance. Central plants require operating engineers in attendance who also frequently service the equipment. The remote units require a combination of these two methods of operation with maintenance by building personnel.

Water and power costs per season can be tabulated and compared for the various systems and an estimate of the costs of replacement material such as filters, belts, oil, refrigerant and wearing parts should be included on an annual basis.

Such questions as obsolescence, depreciation and return on investment are subjects for special study and investigation. Due consideration should be given to the value of resale of equipment and its portability in the event of removal to new locations.

ATTIC FANS

Attic fans are used during the warm months of the year to draw large volumes of outside air through a house and offer a means of using the comparative coolness of outside evening and night air to lower the inside temperature.

Because the low static pressures involved are usually less than ⅛ in. of water, disc or propeller fans are generally used instead of the blower types. The fans should have quiet operating characteristics, and they should be capable of giving about 20 to 30 air changes per hour in northern areas. In the South the usual specification requires one air change per minute which provides appreciable air movement in addition to lowering the inside air temperatures [6].

Types

Open attic fans are units in which the fan is installed in a gable or dormer of the attic and one or more grilles are provided in the floor of the

attic, permitting air to flow from the hall below. Outdoor air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged outside by the fan. An attic stairway may be used in place of the grilles. It is essential that the roof and the attic walls be free from air leaks.

Boxed-in fans are units in which the fan is installed within the attic in a box or housing directly over a central ceiling grille, or in a bulkhead enclosing an attic stair. The fan may be connected by a duct system to the grilles in individual rooms. Outdoor air entering through the windows of the rooms below is discharged into the attic space and escapes to the outside through louvers, dormer windows, or screened openings under the eaves.

Another version of the attic fan is the *window fan* for use when attic application is not feasible or no attic is available. Supplied with a perforated or expanded metal enclosure and mounted in either the upper or lower window section, this fan is easy to install or move to another location.

The locations of the fan, the outlet openings, and grilles should be selected after consideration of the room and attic arrangements in order to give uniform air distribution in the individual rooms served. If the outlet for the air is not on the side away from the direction of the prevailing wind as in the case of the boxed-in fan, openings should be provided on all sides. Kitchens should be separately ventilated because of the fire hazard, and to prevent the spread of cooking odors.

The window fan may be located in a hall or an unused bedroom. Noise of operation is more of a problem with the window fan than with the attic type, although care should be taken to locate either type of fan so that occupants are not disturbed.

These fans range in capacity from 3000 to 30,000 cfm. The window type usually does not exceed 8000 cfm, while the most generally used attic type ranges from 8000 to 16,000 cfm. Power consumption is under 50 watts an hour per 1000 cfm of rated output for the 8000 cfm fan and larger while the watts input for smaller fans is greater than this figure. Improved results can be secured with the window fan by closing off parts of the house where ventilation is not desired.

REFERENCES

1—Prepared by a 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* (*A.S.R.E. Circular* No. 13-42).

2—Standard Method of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling prepared by a 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* (*A.S.R.E. Circular* No. 16).

3—Defined in Proposed *A.S.R.E.* Standard Methods of Rating and Testing Forced-Circulation Air Coolers for Commercial and Industrial Refrigeration (*A.S.R.E. Circular* No. 25-43).

4—Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27). Reprints of this code are available at $0.10 a copy.

5—Concerning Conservation of Underground Water with Suggestions for Control, by Noel E. Porter (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 309).

6—Comfort Cooling with Attic Ventilating Fans, by G. B. Helmrich and G. H. Tuttle (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 155). A.S.H.V.E. RESEARCH REPORT No. 979—Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 167). A.S.H.V.E. RESEARCH REPORT No. 1198—The Effect of Attic Fan Operation on the Cooling of a Structure, by W. A. Hinton and A. F. Poor (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 145). The Installation and Use of Attic Fans, by W. H. Badgett (Agricultural and Mechanical College of Texas, *Bulletin* No. 52, 1940).

CHAPTER 37

Spray Apparatus

Air Washers, Humidification with Air Washer, Apparatus for Direct Humidification, Air Dehumidification with Washers, Well and Water Main Temperatures, Atmospheric Water Cooling Equipment, Selection of Water Cooling Equipment, Design Wet-bulb Temperatures for Water Cooling, Cooling Ponds, Spray Towers, Cooling Tower Design, Cooling Tower Performance, Winter Freezing

AIR humidification is effected by the vaporization of water and always requires heat from some source. This heat may be added to the water prior to the time vaporization occurs or it may be secured by a transformation of sensible heat of the air being humidified to latent heat as the vapor is added to the air. The thermodynamics of the process are discussed in Chapter 3. The removal of moisture from air may or may not involve the removal of heat from the air-vapor mixture. With spray equipment dehumidification of air always necessitates the removal of heat.

AIR WASHERS

Air washers may be used as either humidifiers or dehumidifiers depending upon the method of operation and the temperature of the spray water. The functions of an air washer are to regulate the moisture and heat content of air passing through it and, although not so effective as air filters, to remove dust and dirt from the air.

The construction of commercial air washers is indicated in Figs. 1 and 2. Any air washer consists essentially of a chamber through which the air passes in intimate contact with water. The lower portion of the washer chamber serves as a sump for the spray water.

Contact between the air and the washer water is secured: (1) by breaking the water into a very fine mist, (2) by passing the air over surfaces which are continuously wetted by water, or (3) by a combination of water sprays and wetted plates. Scrubber-plate types of washers are used largely to wash heavy reclaimable products from the air, and are generally composed of one to three eliminator-type baffle scrubber plates across the air stream. Water is supplied at the tops of the scrubber plates by flooding nozzles placed across the top of the washer. Spray washers have one or more banks of water atomizing nozzles placed in the air stream above the level of the water in the sump. The direction of the water sprays may be against the air stream, with the air stream, or with one bank spraying with the air stream and one against it. The number of nozzles required depends upon their design, the quantity of air handled, and the arrangement of the nozzles.

Scrubbers generally consist of eliminator-type baffle plates placed in the air stream to cause several reversals of the direction of air flow. The scrubber plates are more effective as air cleaners than as humidifiers. All washer chambers should have inlet diffuser plates to aid in producing more uniform air flow through the washer spray chamber. These inlet vanes also aid in preventing spray water from being thrown into the air duct ahead of the washer. However, if the water spray opposes the air flow, the ordinary perforated diffuser plate is not sufficient, and specially designed eliminator baffles must be used to prevent spray from passing into the air inlet duct. At the outlet end of the washer suitable flooded

eliminator plates, which will cause from 4 to 6 reversals of the direction of air flow, should be installed for the purpose of removing drops of unvaporized water from the leaving air. If the air contains certain substances which may cause the spray water to become acidulated, corrosion resistant materials must be used in the washer construction.

Essential items in air washer operation are: uniform distribution of the air across the chamber section above the level of the water in the sump; moderate velocities of air flow, 300 to 600 fpm in the spray chamber; an adequate amount of spray water broken up into a fine mist throughout the air stream; sufficient length of air travel through the water spray and over thoroughly wetted surfaces; and the elimination of free moisture from the air as it leaves the unit.

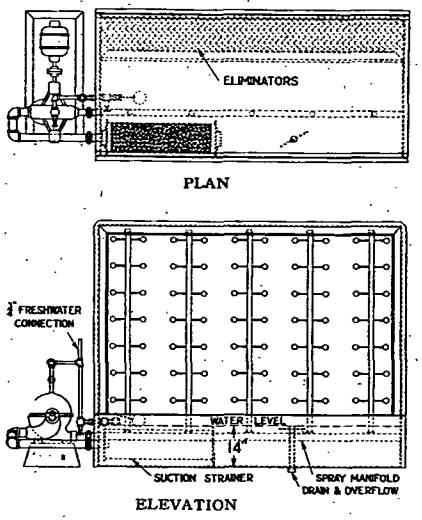


FIG. 1. TYPICAL SINGLE-BANK AIR WASHER

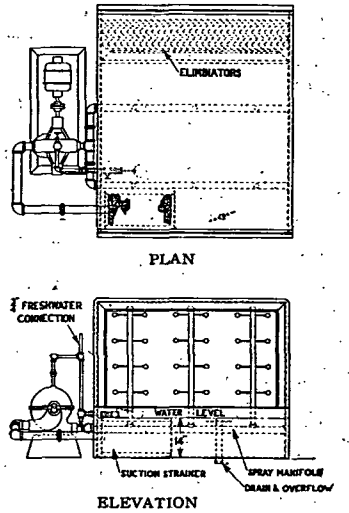


FIG. 2. TYPICAL TWO-BANK AIR WASHER

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the over-all efficiency of heat transfer between the air and the heating or cooling medium. A multi-stage washer is equivalent to a number of washers in a series arrangement.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 fpm through the gross inlet area of the unit. At this rating spray type washers handle about 2½ gpm of water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed.

For a single-stage air washer, a 15 deg drop in dry-bulb temperature of the air passing through the washer is about the maximum that should be anticipated. For greater decrease in dry-bulb temperature, multi-stage washers should be utilized. A rise of 6 deg should be the calculated maximum for the spray water.

The width and height of a washer may be dictated by space limitations outside the washer, such as headroom, or by the inside space requirements, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils. Roughly, a spray space of about 2 ft 6 in. in length is required for each bank of sprays; i.e., 1 ft for entering and 1 ft 6 in. for leaving.

The resistance to air flow through an air washer varies with the type of eliminators, number of banks of sprays, direction of spray, air velocity, type of scrubber plates, and size and type of cooling coils if located in the washer. Manufacturers should be consulted to obtain the resistance for a particular installation.

HUMIDIFICATION WITH AIR WASHER

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 dry-bulb temperature less than 35 F in order to eliminate 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 leak 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 final 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:

$$e_h = \frac{t_1 - t_2}{t_1 - t'} \qquad (1)$$

where

e_h = humidifying effectiveness, per cent.
t_1 = dry-bulb temperature of the entering air, Fahrenheit degrees.
t_2 = dry-bulb temperature of the leaving air, Fahrenheit degrees.
t' = thermodynamic wet-bulb temperature of the entering air, Fahrenheit degrees.

The following may be taken as representative humidifying or saturating effectiveness of an air washer for the conditions stated:

1 bank—downstream	60–70 per cent
1 bank—upstream	65–75 per cent
2 banks—downstream	85–90 per cent
2 banks—1 upstream and 1 downstream	90–95 per cent
2 banks—upstream	90–95 per cent

The humidifying or saturating effectiveness of a washer is dependent upon the number of spray banks and nozzles, the effectiveness of the nozzles in breaking an adequate quantity of water into a fine spray, the

velocity of air flow through the water sprays, and the time of the contact of the air with the spray water. Other conditions being the same, low velocity of air flow is more conducive to higher humidifying effectiveness.

The air leaving the washer may require reheating to produce the required dry-bulb temperature and relative humidity.

Method 2. The preheating of the air increases both the dry- and wet-bulb temperatures, lowers the relative humidity, but does not alter the humidity ratio (pound 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 entrance to the washer and then reheating the outlet air when necessary.

Method 3. Even if 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. By elevating the water temperature it should be possible to saturate the air and even raise its temperature above the dry-bulb temperature of the entering air.

APPARATUS FOR DIRECT HUMIDIFICATION

Humidifiers may be divided into the following general types, according to the method of operation: (1) indirect, such as the air washer, which introduces moistened air; and (2) direct, which sprays moisture into the room or introduces moisture by means of steam jets.

As in the cases of humidification by use of an air washer, the heat necessary for the vaporization of the moisture added to the air by direct humidification is secured either from heat stored in the spray water or by a transformation of sensible to latent heat in the air humidified. In the latter case the enthalpy of the air remains constant but the dry-bulb temperature of the air is reduced.

Direct humidification is usually preferable where high relative humidities must be maintained, but where there is little cooling or ventilation required. In comfort air conditioning, where both humidification and ventilation are required, the indirect humidifier is preferable. In industrial applications, where the cooling or ventilation load is large and where very high relative humidities must be maintained, a combined system employing both direct and indirect humidifiers is sometimes used.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves 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 immediately 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

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing, and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray. These cannot occur when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is evaporated and the resulting vapor diffused. This distribution of fine spray over the maximum possible area promotes complete and rapid vaporization.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray 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

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries. These may be used either singly or in groups. In large installations, where suitable connections are provided to permit the

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURES[a]

STATE	CITY	TEMP. F	STATE	CITY	TEMP. F	STATE	CITY	TEMP. F
Ala.	Birmingham	84	Mass.	Lowell	50	Pa.	Altoona	74
	Mobile	73		Lynn	68		Erie	75
Ariz.	Phoenix	81		New Bedford	70		Johnstown	74
	Tucson	80		Salem	68		McKeesport	82
Calif.	Anaheim	60		Worcester	76		Philadelphia	83
	Berkeley	69	Mich.	Detroit	77		Pittsburgh	81
	Fresno	72		Flint	70	R. I.	Providence	68
	Fullerton	75		Grand Rapids	84	S. C.	Charleston	80
	Glendale	68		Highland Park	77		Greenville	81
	Los Angeles	75		Jackson	56		Spartanburg	78
	Oakland	69		Kalamazoo	53	S. D.	Rapid City	55
	Ontario	70		Lansing	64	Tenn.	Chattanooga	84
	Pasadena	82		Saginaw	82		Knoxville	89
	Pomona	75	Minn.	Duluth	55		Memphis	70
	Riverside	78		Minneapolis	80		Nashville	90
	Sacramento	72		St. Paul	77	Texas	Amarillo	65
	San Bernardino	65	Mo.	Jefferson City	82		Austin	90
	San Diego	82		Kansas City	84		Beaumont	86
	San Francisco	62		St. Joseph	84		Dallas	86
	Whittier	75		St. Louis	85		Fort Worth	84
Colo.	Denver	75		Springfield	70		Galveston	90
Conn.	Bridgeport	66	Nebr.	Lincoln	63		Houston	84
	Hartford	73		Omaha	87		Port Arthur	83
	New Haven	76	Nev.	Reno	61		San Antonio	76
	Waterbury	72	N. H.	Manchester	76		Wichita Falls	85
D. C.	Washington	84	N. J.	Jersey City	63	Utah	Logan	44
Del.	Wilmington	83		Newark	74		Salt Lake City	60
Fla.	Jacksonville	80		Paterson	78	Va.	Fredericksburg	75
	Miami	80		Trenton	79		Lynchburg	73
	Tampa	77	N. Y.	Albany	68		Norfolk	80
Ga.	Atlanta	87		Buffalo	75	Wash.	Olympia	58
	Macon	80		Jamaica	56		Seattle	62
Ill.	Chicago	76		Mt. Vernon	74		Spokane	51
	Cicero	76		New Rochelle	75		Tacoma	57
	Evanston	73		New York	72	W. Va.	Charleston	85
	Peoria	67		Rochester	70		Huntington	78
	Rockford	59		Schenectady	60		Wheeling	78
	Springfield	82		Syracuse	74	Wis.	LaCrosse	54
Ind.	Evansville	86		Utica	69		Madison	58
	Gary	75		Yonkers	70		Milwaukee	70
	Indianapolis	80	N. C.	Asheville	74		Racine	68
	South Bend	61		Charlotte	85			
	Terre Haute	82		Winston-Salem	82			
Iowa	Cedar Rapids	78	N. M.	Albuquerque	65			
	Des Moines	77	Ohio	Akron	76			
	Sioux City	62		Canton	50			
Kans.	Concordia	57		Cincinnati	84	PROVINCE		
	Kansas City	86		Cleveland	74			
	Topeka	88		Columbus	82			
	Wichita	72		Dayton	60			
Ky.	Louisville	85		Lakewood	82	Alta.	Calgary	64
La.	Baton Rouge	85		Springfield	72	B. C.	Vancouver	60
	New Orleans	85		Toledo	83	Ont.	London	50
Me.	Augusta	60	Okla.	Oklahoma City	82		Toronto	63
Md.	Baltimore	75		Tulsa	85	P.E.I.	Charlottetown	48
Mass.	Boston	80	Ore.	Eugene	60	Que.	Montreal	78
	Cambridge	70		Portland	64		Quebec	68
	Fall River	76						

[a]These averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

AIR DEHUMIDIFICATION WITH WASHERS

Moisture removal from an air-vapor mixture can be accomplished by use of an air washer so long as the temperature of the spray medium is lower than the dew-point of the air passing through the unit. The final dry-bulb temperature and the relative humidity of the air leaving a dehumidifier washer are dependent upon: the air velocity, the length of air travel through the sprays, the dry- and wet-bulb temperatures of the entering air, the spray temperature, the number of spray banks and nozzles, the quantity of spray medium handled, and the effectiveness of the nozzles in breaking the spray into a fine mist.

Both sensible and latent heat are removed in the process of dehumidification by cooling. Abstraction 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 temperature the greater the amount of moisture removal per pound of dry air, all other conditions remaining the same. Washers with two or more banks of sprays are usually selected for comfort air conditioning installations. Such washers will cool the air to within 1 or 2 F of the leaving spray water temperature.

Where a limited supply of cold water is available multiple stage washers may be used to an advantage. The cool water is pumped through the multiple spray systems in series. By this arrangement the entering air is cooled first by the warmer water and finally by the cooler water which gives the maximum amount of cooling with the minimum amount of water. The approximate temperatures of water from wells at depths of 30 to 60 ft are given in Fig 3[1]. Frequently the temperature of the city water main supply is low enough during the summer to permit an appreciable cooling effect. Table 1 lists the maximum city water main temperatures for various localities in the United States and Canada.

Air washers using refrigerated spray generally have their own recirculating pumps. These pumps deliver to the sprays a mixture of water from the washer sump, which has not been re-cooled, and refrigerated water. The quantities of each are controlled by a three-way or mixing valve actuated by a dew-point thermostat located in the washer air outlet or by humidity controllers located in the conditioned space.

ATMOSPHERIC WATER COOLING EQUIPMENT

In the operation of a refrigerating plant or a condensing turbine, one of the main problems is the removal and dissipation of heat from the compressed refrigerant or the discharged steam. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger, from which water it may then be dissipated in a number of ways. If the plant is situated on the banks of a river or lake, an intake may be taken up-stream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of cooling water is a city supply or a well, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where purchased water is too expensive or where the water available contains dissolved salts which would form scale on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-

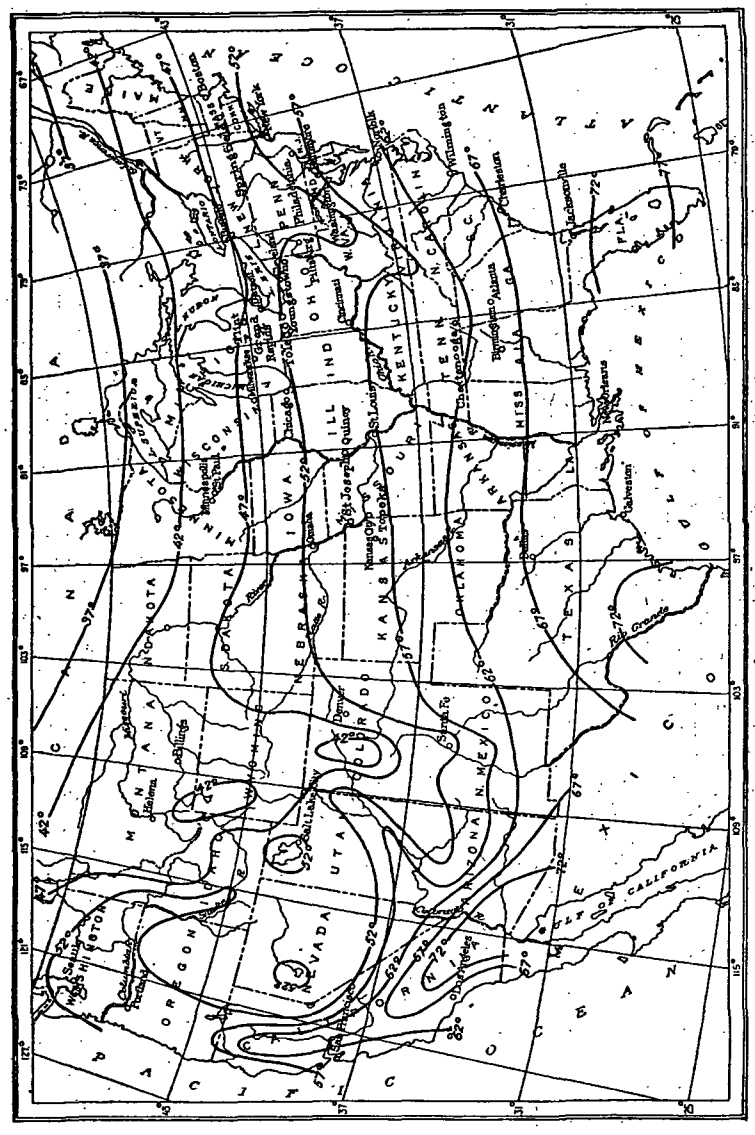


FIG. 3. APPROXIMATE WELL WATER TEMPERATURES AT DEPTHS OF 30 TO 60 FT

exchanger by exposure to air in an atmospheric water cooling apparatus.

Air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact. The rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. The rate of heat dissipation is influenced further by many small factors[2] prevailing upon these primary ones, and complicating the design. Selection of equipment for any specified service must ultimately rest in over-all economic considerations established from reliable performance data.

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 3.) Because it is impracticable to leave the air in contact with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least expenditure of power.

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design in that the size of the equipment is determined by the quantity of air to be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application. With water cooling apparatus, its size and the quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Selection of Water Cooling Equipment

The proper size and type of water cooling equipment for a given installation cannot be determined accurately without considering the characteristics of all types, together with all requirement factors and other matters which influence the selection. Very few installations are exactly alike in details of requirements, and conditions governing performance and operation of the several types of water cooling equipment vary widely with geographical location as well as other considerations. It necessarily holds, therefore, that requirement factors and equipment characteristics are closely interlocked and deserve careful study as a related whole.

Before the characteristics of a specific water cooling apparatus can be judged desirable or undesirable the survey must conclude definitely the importance of each of the following items: first cost including all necessary auxiliaries, area, height, weight, effect of wind velocity and direction, drift nuisance, make-up water requirements and cost of chemical treatment if needed, total power for pumping (plus fan operation in the case of mechanical draft), maintenance, available locations (with due thought to possible future expansion, wind restrictions, space cost, proximity and accessibility, etc.), appearance, the equipment's operating flexibility for most economical conformance to varying loads or seasonal changes, and other considerations occurring with regard to a specific application.

Sizes of Equipment

Assuming a definite quantity of water, the size and design of atmospheric cooling equipment are affected by the following primary conditions:

1. Temperature range through which the water must be cooled.
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced.
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air.)
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon: (1) type of service (refrigerating, internal combustion engine and steam condensing). (2) wet-bulb temperature at which the equipment must operate satisfactorily, (3) type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 psi head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 psi and power consumption increases. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 or 4 deg of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 deg difference.

Table 2 lists several gases with data as to the temperatures and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F for closed

TABLE 2. CONDENSER DESIGN DATA

Gas	Maximum Pressure Desired in Condenser	Gas Temperature in Condenser F Deg	Leaving Hot Water Temperature F Deg	
			Best Condenser Design	Average Condenser Design
Steam	28 in. vacuum	101.2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.9	120	114
Ammonia	185 psi[a]	96.0	92	88
Carbon dioxide	1030 psi[a]	86.0	83	81
Methyl chloride	102 psi[a]	100.0	96	92
Dichlorodifluoromethane	117 psi[a]	100.0	96	93

[a] Head pressure.

systems, and 110 F to 120 F for open systems, depending upon the quality of the cooling water. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

The temperature range, once the hot water temperature is approximately known, depends upon: (1) the maximum wet-bulb temperature at which the full quantity of heat must be dissipated, and (2) the efficiency of the atmospheric cooling equipment considered.

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1934, inclusive, there were but 8 hours per year when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September inclusive) when the wet-bulb temperature was 68 F or above. As these 975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for air conditioning design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City) with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on an air conditioning plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 2, Chapter 15, shows design wet-bulb temperatures which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the effectiveness range of the type of atmospheric water cooling apparatus to be used. This effectiveness is expressed as the percentage ratio of the actual cooling effect to the maximum possible cooling effect. Since the wet-bulb temperature of the entering air is the equilibrium temperature to which the water could be cooled, the effectiveness of water cooling apparatus can be indicated thus:

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}$$

Magnitudes of this effectiveness ratio will vary through wide limits in accordance with construction and conditions of operation. Values indicative of the commercial range of the effectiveness ratio are given in Table 3, although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be

calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various processes of the cooling equipment are:

Compressor Refrigeration: 220 to 270 Btu per (minute) (ton). Usual practice is to assume: 250 Btu per (minute) (ton) which is equivalent to 30 gal per (Fahrenheit degree) (minute) (ton).

Steam Turbine Condensers: 950 to 980 Btu per pound of steam. Usual practice is to assume 970 Btu per pound of steam.

Steam Jet Refrigerating Condensers: 1030 to 1150 Btu per pound of steam. Exact value depends upon initial steam conditions.

Diesel Engine Jackets: 2500 to 4000 Btu per bhp per hour. Usual practice is to assume 3500 Btu per bhp per hour.

Natural Gas or Gasoline Engines: 4500 to 6000 Btu per bhp per hour. Usual practice is to assume 5000 Btu per bhp per hour.

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line. Natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift away with the air movement. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased volume of air used, and (4) the vastly increased area of contact between air and water [3].

Spray pond effectiveness is increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the

TABLE 3. EFFECTIVENESS OF ATMOSPHERIC WATER COOLING EQUIPMENT

EQUIPMENT	COOLING EFFECTIVENESS—PER CENT		
	Minimum	Usual	Maximum
Spray Ponds	30	40 to 50	60
Spray Towers	40	45 to 55	60
Natural Draft Deck Towers	50	60 to 75	90
Mechanical Draft Towers	50	60 to 75	90

nozzles from 5 to 7 ft above the edge of the basin, to supply from 5 to 12 psi pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.5 to 0.7 gpm per square foot. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

Spray ponds may be located on the ground, or they may be placed on roofs. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae growths, during warm weather, in cooling towers and spray ponds may be eliminated while the plant is in operation by the use of potassium permanganate. This chemical can be dissolved at the rate of 1 lb in 1¼ to 1½ gal of hot water. About 10 parts of permanganate should be used per million parts of cooling water. Enough of the permanganate solution should be added periodically to cause the water to have a pink color for a period of from 15 to 20 min. The best results are obtained when sufficient quantities are added periodically at intervals of several weeks, the time intervals being dependent upon local operating conditions. The chemical is non-poisonous and non-corrosive when used as directed.

Natural Draft Spray Type Towers

Where not more than 30,000 Btu per minute are to be dissipated, the natural draft spray type tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The wet louvers add to the surface exposed to the air.

Natural Draft Deck Type Towers

In past years much of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 20 to 40 ft high and from 10 to 20 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to

the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks add to the area of water surface exposed to the air both by causing it to splash into fine droplets and to spread over the deck surfaces, but since some deck designs furnish a resistance to air flow, too many decks may be a detriment.

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. In some towers these boards or louvers extend outward and upward, usually with lower edge (which is attached to the frame of the tower) somewhat below the level of the outer and upper edge of the next louver board below it. Some louvering designs add *secondary* louvering to these *primary* louvers. In such designs the primary louvers are generally broader and do not slope so sharply nor overlap one another. Since their function is merely to return to the interior of the tower the water caught by the secondary louvering which extends upward from each primary louver's outer edge to the next primary louver above. The individual slats which comprise the secondary louvering may be arranged either vertically in an overlapping staggered manner with air passage between, or set horizontally with their flat surfaces sloping inward and spaced so their edges over-reach but permit free air passage. Secondary louvering reduces drift loss on this type of tower materially.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increase the area of water exposed to the wind, and the latter increases the time of contact of the air with the water. Performance is best when the wind blows crosswise to the tower and decreases with obliqueness of the angle until the effect of a lengthwise wind is virtually nil.

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling at times when the wind velocity is below average, when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 2, Chapter 15, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in no case for a wind velocity of more than 5 mph. Natural draft equipment must not be obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of box-like shells, constructed of wood, metal, or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood lattice work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles.

Air is drawn through the tower by induced draft fans or propelled through it by forced draft fans. Since the air flows at a constantly sustained and comparatively high velocity (by contrast with natural draft equipment which varies in performance with every change of the wind's direction or velocity) the air and water are brought into contact

under controlled conditions resulting in more efficient transfer of heat from a given quantity of water into a given volume of air. The least possible operating cost is achieved by that tower design which delivers the desired performance for the lowest total power input, both to fans for air movement and to pumps for raising and distributing the water.

The effectiveness of a mechanical draft tower is improved by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of lowered velocities it increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through mechanical draft towers vary from 250 to 450 fpm over the gross area of the structure.

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged through a duct to the outside.

Cooling Tower Design

The method of design of equipment for energy transfer from water to an air-water vapor mixture is similar to that used for absorption equipment. Details of this procedure are available[4, 5, 6] and its application to the problem of the cooling tower operating at atmospheric pressure is illustrated by the following development. The nomenclature used is as follows:

a = over-all average wetted area (surface of water drops plus wetted tower surface) square feet per cubic foot of tower volume.
c = specific heat of liquid water, Btu per (pound) (Fahrenheit degree).
ϵ = effectiveness.
ε = natural base.
G = weight rate of flow of air, pounds of dry air per hour.
h = enthalpy, Btu per pound of dry air.
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 = over-all energy unit conductance, Btu per (hour) (square foot over-all average wetted area) (Btu enthalpy difference per pound of dry air).
Ka = over-all rate coefficient, Btu per (hour) (cubic foot of tower volume) (Btu enthalpy difference per pound of dry air).
L = water rate, pounds per hour.
lm = logarithmic mean.
S = average cross-sectional area of cooling tower for air flow, square feet.
t = water-main body temperature, Fahrenheit degrees.
t_{wb} = wet-bulb temperature, Fahrenheit degrees.
V = tower volume, cubic feet.

Note: Subscripts 1 and 2 when used in equations refer to water entrance and exit sections respectively, for the counter-flow tower.

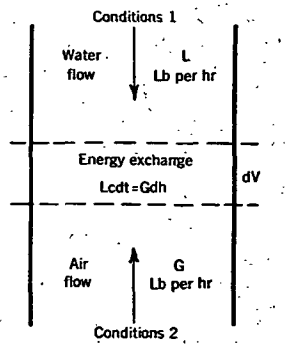


FIG. 4. SECTION OF TYPICAL COUNTER-FLOW TOWER

A section of a typical counter-flow tower is shown in Fig. 4. If the reduction in water rate due to evaporation within the volume is neglected, the energy balance for this differential section of the exchanger volume may be written as:

$$L\,c\,dt = G\,dh \tag{2}$$

The potential for net energy transfer due to heat and mass transfer from the water to the mixture in contact with it may be expressed with reasonable accuracy as the difference between the enthalpy of saturated air at the water temperature, h'', and the enthalpy of the main stream air vapor mixture[7], h_a. The rate of energy transfer is given by the expression:

$$Ka\,(h'' - h_a)\,dV \tag{3}$$

which equation defines the over-all rate coefficient, Ka; the latter being

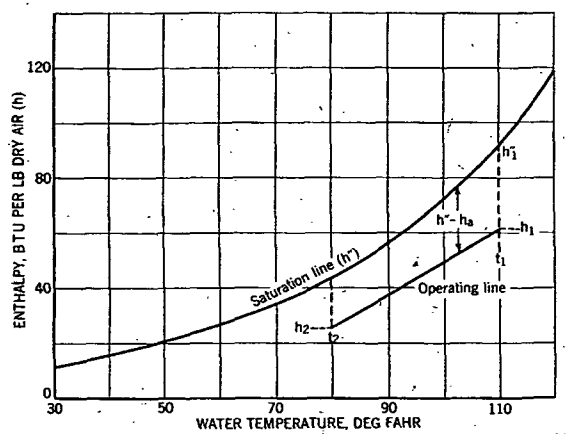


FIG. 5. TEMPERATURE ENTHALPY DIAGRAM FOR AIR WATER VAPOR MIXTURE SHOWING THE OPERATING LINE FOR EXAMPLE 1

the product of the over-all energy unit conductance, K, and the ratio of the transfer surface to the exchanger volume, a.

Equations 2 and 3 are conveniently illustrated by means of the temperature-enthalpy diagram of Fig. 5. Equation 2 indicates that the succession of air and water states existing in the exchanger sections must combine to form a straight line (for L = constant) on the temperature enthalpy diagram. The slope of this operating line is:

$$\frac{dh}{dt} = \frac{Lc}{G}$$

Since the heat capacity of water is approximately unity, this slope is the ratio of the water to the air rate. Equation 3 indicates that the potential for energy transfer at any section is the difference between the enthalpy of saturated air at the main-body water temperature at that section and the enthalpy of the air stream in contact with that water. This potential is the difference in the ordinates of the saturation and operating lines for the water temperature at the plane in the tower which is under consideration.

Combination of Equations 2 and 3 results in the expression:

$$G\,dh = Ka\,(h'' - h_a)\,dV \tag{4}$$

Integrating this equation over the length of the exchanger:

$$\int_2^1 \frac{G}{Ka}\,\frac{dh}{h'' - h_a} = V \tag{5}$$

The integration of the left side of Equation 5 determines the tower volume required to achieve the desired energy exchange. This summation is readily accomplished for counter and parallel flow arrangements. G and Ka are usually independent of the tower volume and Equation 5 then becomes:

$$NTU = \int_2^1 \frac{dh}{h'' - h_a} = \frac{KaV}{G} \tag{6}$$

Where NTU is defined as the *Number of Transfer Units* and is a measure of the difficulty of the cooling process.

The integration is made numerically or graphically. In the graphical integration $\frac{1}{h'' - h_a}$ is evaluated as a function of h_a. This determination involves the use of the energy balance equation integrated from one section to the section in question. The area under the curve between any two abscissae is the number of transfer units required to change the air state from h_2 to h_1.

An approximate value for the number of transfer units can also be determined by a simple graphical method of direct construction on the temperature enthalpy diagram[8]. This method cannot be applied very satisfactorily to cooling towers as the operating range is small and the value of the NTU is near unity.

When the relationship between the enthalpy of the saturated air and the temperature is linear over the range of water temperatures involved, it can be shown[9] that the logarithmic mean of the terminal potentials, h_{lm}, is the correct driving force. This is true to a good approximation when the water cooling does not exceed 15 deg. The approximation to linearity may be determined by inspection of Table 1, Chapter 3, or the temperature enthalpy diagram of Fig. 5. If the logarithmic mean is a valid potential, Equation 6 may be written:

$$\frac{h_1 - h_2}{\Delta h_{lm}} = \frac{KaV}{G} \tag{7}$$

and the need for the numerical integration for the determination of the tower volume is eliminated.

The over-all rate coefficient, Ka, must be known if the tower volume is to be determined. Experiments conducted on towers containing different packing construction have yielded some magnitudes of this coefficient, evaluated on an over-all basis. These data are presented in Fig. 6 as a function of the gas mass velocity through the packing, and apply only to the particular packing structure for which they were obtained. The over-all rate coefficient (Ka) may also be a function of the water rate, since a reduction of the water rate may reduce the wetted area within the

exchanger[5]. The results included in Fig. 6 probably represent magnitudes of Ka which were obtained for complete wetting. Within the cooling tower operating range the over-all rate coefficient for energy transfer is nearly the same numerically as the over-all rate coefficient for mass transfer. The conditions of test corresponding to the data presented in Fig. 6 are not well enough known in most cases to warrant recomputation of Ka. Therefore the magnitudes of the over-all rate coefficient for mass transfer presented in Fig. 6 may be used directly in Equations 5 and 6,

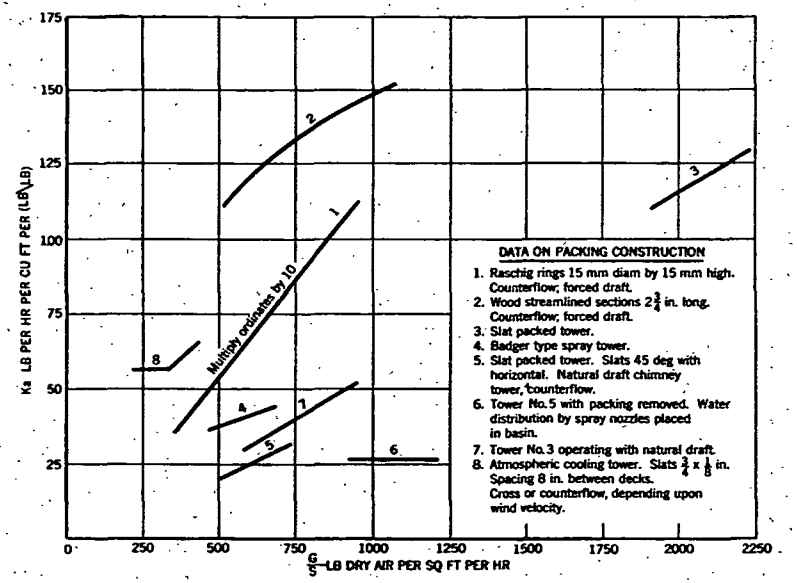


FIG. 6. UNIT CONDUCTANCES FOR VARIOUS TYPES OF PACKING CONSTRUCTION

the units of Ka in these equations being Btu per (hour) (cubic foot per pound of dry air).

Application of Design Procedure

A typical procedure which may be followed in designing a cooling tower is illustrated in the following Example 1.

Example 1. The rate of air flow, arbitrarily assumed in the data given, is related to the tower volume by economic considerations. A balance between air rate and tower volume rests on consideration of the costs of producing air flow and of the tower construction[10]. A counter-flow forced draft cooling tower is to cool 36,000 lb of water per hour from an initial temperature of 110 F to a final temperature of 80 F. Air having an initial condition of 65 F dry-bulb and 58 F wet-bulb temperature will be forced through the tower counter to the direction of water flow at the rate of 30,000 lb of dry air per hour.

The cross-section of the tower is to be 8 ft x 8 ft and the packing is to be of the type producing a rate coefficient as indicated in curve No. 2 of Fig. 6. For this type of packing the average cross-sectional area for air flow will be 36 sq ft.

Solution:

Initial air enthalpy = 25.1 Btu per pound of dry air.

Final air enthalpy:

$$(h_1 - h_2) = \frac{Lc}{G}(t_1 - t_2)$$

$$(h_1 - 25.1) = \frac{36{,}000 \times 1}{30{,}000}(110 - 80)$$

$h_1 = 61.1$ Btu per pound dry air.

A numerical integration (Table 4) is employed to determine the Number of Transfer Units (NTU) required. Temperature increments of 2 F are used between successive determinations of the quantity $\frac{1}{h'' - h_a}$. The energy balance indicates that the enthalpy

TABLE 4. NUMERICAL INTEGRATION FOR THE NUMBER OF TRANSFER UNITS

Water Temperature Interval F Deg	Mean Water Temperature F Deg	Mean Air Enthalpy, h_a Btu per Lb Dry Air	Saturated Air Enthalpy, h'' Btu per Lb, Dry Air	Enthalpy Potential $h'' - h_a$	$\frac{\Delta h}{h'' - h_a}$
80–82	81	26.3	44.6	18.3	0.131
82–84	83	28.7	46.9	18.2	0.132
84–86	85	31.1	49.2	18.1	0.133
86–88	87	33.5	51.7	18.2	0.132
88–90	89	35.9	54.4	18.5	0.130
90–92	91	38.3	57.1	18.8	0.128
92–94	93	40.7	60.0	19.3	0.124
94–96	95	43.1	63.0	19.9	0.121
96–98	97	45.5	66.2	20.7	0.116
98–100	99	47.9	69.6	21.7	0.111
100–102	101	50.3	73.2	22.9	0.105
102–104	103	52.7	77.0	24.3	0.099
104–106	105	55.1	80.9	25.8	0.093
106–108	107	57.5	85.1	27.6	0.087
108–110	109	59.9	89.5	29.6	0.081
					1.723

increments corresponding to these temperature increments are:

$$\Delta h = \frac{Lc}{G}\Delta t = \frac{36{,}000}{30{,}000} \times 2 = 2.4 \text{ Btu per pound of dry air.}$$

The result of the integration is that the Number of Transfer Units required (NTU) = 1.72.

Unit gas mass velocity, $\frac{G}{S} = \frac{30{,}000}{36} = 830$ lb per (hour) (square foot average cross-sectional air flow area).

From Curve 2, Fig. 6, Ka = 138 Btu.

The tower volume required is:

$$V = \frac{G}{Ka}(NTU) = \frac{30{,}000}{138} \times 1.72 = 217 \times 1.72 = 374 \text{ cu ft.}$$

Height of the packed section is:

$$\frac{374}{8 \times 8} = 5.9 \text{ ft.}$$

The graphical solution for the Number of Transfer Units required for the desired performance is plotted in Fig. 7. $\frac{1}{h'' - h_a}$ is plotted as a function of h, and the area

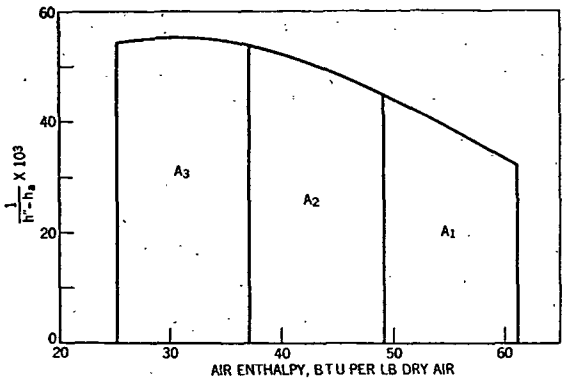


FIG. 7. GRAPHICAL INTEGRATION TO DETERMINE NUMBER OF TRANSFER UNITS REQUIRED FOR DESIRED OPERATING CONDITIONS OF EXAMPLE 1

under the curve from the initial enthalpy of the air, 25.1 Btu per pound, to the final enthalpy of the air, 61.1 Btu per pound is 1.72, the Number of Transfer Units required. The effect of the rapid decrease in potential due to the cooling of the water is indicated by comparison of area A_1, A_2, and A_3 of Fig. 7. Each represents the Number of Transfer Units required to achieve a water temperature reduction of about 10 F.

$$A_1 = 0.471\ NTU\ (110 \text{ to } 100 \text{ F})$$
$$A_2 = 0.595\ NTU\ (100 \text{ to } 90 \text{ F})$$
$$A_3 = 0.657\ NTU\ (90 \text{ to } 80 \text{ F})$$

The use of the logarithmic mean driving potential is illustrated by applying Equation 7 to Example 1:

$$NTU = \frac{(Ka)\ V}{G} = \frac{h_1 - h_2}{\Delta h_{lm}}$$

$$\Delta h_{lm} = \frac{(h_1'' - h_1) - (h_2'' - h_2)}{ln_e \frac{h_1'' - h_1}{h_2'' - h_2}} = \frac{30.7 - 18.4}{ln_e \frac{30.7}{18.4}} = 24$$

$$NTU = \frac{61.1 - 25.1}{24} = 1.5$$

The Number of Transfer Units required as determined by use of the logarithmic mean driving potential equals 1.5 which compares favorably with the correct magnitude, 1.72.

A variation of the design method according to Equation 6 is further possible [11] through the introduction of Equation 2 to yield

$$\int_{t_2}^{t_1} \frac{cdt}{h'' - h_a} = \frac{KaV}{L} \tag{8}$$

The advantage of this form is in the fact that the left side of the equation may be dealt with as a thermodynamic function only, while the right side depends upon the tower construction and operation. A series of performance curves may be determined from a graphic evaluation of the integral, which will serve either as a convenient means of analyzing experimental data to determine the conductance Ka, or as a rapid method

of performance prediction, once the tower characteristic, $\frac{KaV}{L}$, is known. These performance curves may be expressed in terms of the following variables; approach to wet-bulb, cooling range, L/G, and wet-bulb temperature of the entering air; and as such they will serve to accurately cover the tower performance in consistent terms.

The application of the foregoing design method to atmospheric towers is difficult because rate coefficients and flow conditions are not yet well defined for such equipment. If these are known, application of Equation 7 to sections of the tower small enough to justify use of the logarithmic mean potential will yield the tower volume required for each section. The sections must be taken perpendicular to the path of water flow. A correction to adjust the logarithmic mean potential, evaluated as for counter-flow, to the reduced effectiveness of cross-flow, has been derived for heat transfer and may be applied to this case [12].

For small size counterflow spray towers intended for a low cooling range only considerable data [13, 14] are available on the performance of the spray system in terms of the logarithmic mean enthalpy potential. These data show the importance of proper nozzle arrangement and spray distribution within the tower. The greatest cooling effect is shown to be obtained in a region close to the nozzles; and the beneficial effects of packing added in the lower section of the tower follow as increasing the cooling range.

Atmospheric towers operate with natural draft, produced in a vertical direction by the stack action of the tower structure, at zero velocity of the approach wind. Approach wind of sufficient magnitude (the magnitude depending on the baffle arrangement which is designed to reduce drift) will cause cross-flow augmenting the natural draft. An adequate design requires the consideration of both flow conditions.

Expression for Cooling Tower Performance

The performance of a cooling tower is described in terms of its effectiveness as an energy exchanger. The effectiveness is defined as the ratio of the energy actually exchanged to the energy available for exchange.

Effectiveness expressions:

Case 1. The slope of the operating line on the $t - h$ diagram exceeds the slope of the saturation line in the region of water temperatures considered.

$$e = \frac{h_1 - h_2}{h_1'' - h_2} \tag{9}$$

Case 2. The slope of the saturation line exceeds that of the operating line.

$$e = \frac{h_1 - h_2}{\frac{Lc}{G}(t_1 - t_{wb})} = \frac{t_1 - t_2}{t_1 - t_{wb}} \tag{10}$$

This equation represents the *approach to wet-bulb.*

Usual tower operating conditions conform to *Case 1.* Because of the curvature of the saturation line, operating conditions may present themselves to which neither *Case 1* nor *2* applies. Since a simple expression for the intermediate case is not available, the expression of *Case 1* may be utilized for the small number of operating conditions falling into the intermediate classification.

Make-Up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the remainder, there is a continual drain on the quantity of water in the system, and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 deg range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Chemical treatment of the make-up water may be necessary to avoid excessive deposits in the condensers.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled to freezing temperature so ice forms and collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

REFERENCES

1—Temperature of Water Available for Industrial Use in the United States, by W. D. Collins (*U. S. Geological Survey*, Water Supply Paper No. 520 F).

2—Cooling Tower Performance Studies, by L. M. K. Boelter (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939; p. 615).

3—A.S.H.V.E. RESEARCH PAPER—Design of Spray Cooling Ponds, by S. Hori, U. A. Patchett and L. M. K. Boelter (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1942, p. 624).

4—Principles of Chemical Engineering, by W. H. Walker, W. K. Lewis, W. H. McAdams and E. R. Gilliland (McGraw-Hill Co., 1937, p. 480).

5—Absorption and Extraction, by T. K. Sherwood (McGraw-Hill Co., 1937, p. 91).

6—Performance Characteristics of a Mechanically Induced Draft, Counterflow, Packed Cooling Tower, by A. L. London, W. E. Mason and L. M. K. Boelter (*A.S.M.E. Transactions*, January, 1940, Vol. 62, p. 41).

7—Determination of Unit Conductances for Heat and Mass Transfer by the Transient Method, by A. L. London, H. B. Nottage and L. M. K. Boelter (*Industrial and Engineering Chemistry*, April, 1941, Vol. 33, p. 467).

8—Graphical Method of Determining Number Transfer Units, by T. Baker (*Industrial and Engineering Chemistry*, August, 1935, Vol. 27, p. 977).

9—Loc. Cit. Note 4, p. 79.

10—Loc. Cit. Note 4, p. 142.

11—Performance and Selection of Mechanical-Draft Cooling Towers, by Joseph Lichtenstein (*A.S.M.E. Transactions*, October, 1943, Vol. 65, No. 7, p. 779).

12—Heat Transmission, by W. H. McAdams (McGraw-Hill Co., 1933, p. 157).

13—A.S.H.V.E. RESEARCH REPORT No. 1189—Performance Characteristics of a Forced Draft, Counterflow Spray Cooling Tower, by H. H. Niederman, E. D. Howe, J. P. Longwell, R. A. Seban and L. M. K. Boelter (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 413).

14—A.S.H.V.E. RESEARCH REPORT No. 1240—Spray Nozzle Performance in a Cooling Tower, by L. M. K. Boelter and S. Hori (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 309).

CHAPTER 38

Dehumidification by Sorbent Materials

Definitions and Principles of Dehumidification by Sorption, Adsorbents, Absorbents, Dehumidification with Solid and Liquid Sorbents, Equipment Auxiliaries and Controls, Equipment Performance, Comparison of Methods of Dehumidification

THE reduction of the density of water vapor existing in any form within a given space is defined as dehumidification. 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, this being one of the prime considerations in the choice of method. Dehumidification may be accomplished by sensible heat removal as described in Chapters 7, 37 and 43, or by utilizing sorbents.

Sorbents are materials which have the property of dehumidifying gases without the help of refrigeration. All known materials are sorbents to a certain degree, in that they will decrease or increase in moisture content if their vapor pressure is greater or less than that of the surrounding atmosphere. As generally used, however, the term refers to those materials whose capacity for moisture is large compared to their bulk and weight. Such materials are divided into two general classifications:

1. *Adsorbent*—A material which has the ability to hold water or other vapors on its internal surfaces without itself being permanently changed physically or chemically. Certain solid materials, such as silica gel, and activated alumina have this property. Adsorbents also show an affinity for vapors other than water and this action is frequently selective. Thus, in a mixture of water vapor and organic gases, silica gel will effectively remove the water vapor and charcoal will remove the organic gases. The selective property of adsorbents is used in the removal of some contaminating gases. (See Chapter 33.)

2. *Absorbent*—A material which has the ability to take up water vapor but which changes physically, chemically, or both, during the cycle. Calcium chloride is an example of a solid material while liquid materials include solutions of lithium chloride, calcium chloride, lithium bromide and the ethylene glycols.

The fundamental principle by which sorbents remove moisture from gases lies in their property (when properly active) of having a low vapor pressure. When brought in contact with humid gas having a higher vapor pressure, the moisture from the gas is adsorbed or absorbed into the sorbent. This process can continue until the vapor pressure of the sorbent has been raised by the increase in moisture content until it equals that of the humid air. The vapor pressure of the sorbent increases not only with moisture content but also with increased temperature. The process is reversible; hence to reactivate a sorbent, the temperature is raised until its vapor pressure exceeds that of the surrounding gases. The moisture will then be driven out of the sorbent by vapor pressure difference. Upon cooling down to normal working temperature, the vapor pressure of the dehydrated sorbent will have been reduced to the initial condition, thus completing the cycle.

Adsorbents

These substances contain a vast number of sub-microscopic pores which afford a tremendous internal surface to which water adheres or is

adsorbed. In spite of a porous structure these substances retain sufficient mechanical strength to resist the wear and handling to which they are subjected. To be suitable for dehumidification purposes such substances must fulfill the following requirements:

1. Possess suitable water vapor pressure characteristics.
2. Be available at a reasonable cost.
3. Adsorb sufficient moisture per pound of material to avoid excessive bed dimensions.
4. Be chemically stable, resisting contamination from impurities.
5. Be physically rugged to resist breakdown from handling, abrasion, etc.
6. Be able to withstand breakdown from indefinitely repeated reactivation cycles.
7. Be capable of reactivation at reasonable temperatures.

Activated Aluminum Oxide (Alumina) is a porous, amorphous form of aluminum oxide and is called by the common name Activated Alumina. Commercial activated alumina commonly used for dehumidification contains about 92 per cent of Al_2O_3 combined with hydrated aluminum oxide. It is available in granules ranging from a fine powder to pieces approximately 1.5 in. in diameter. It has high adsorptive capacity per unit of weight and is non-toxic. It may be repeatedly re-activated after adsorbing moisture without practical loss of its adsorptive ability. In the grade frequently used for air drying the reactivation may be accomplished at temperatures between 350 and 600 F. Specific gravity is 3.25 and the pores are reported to occupy 51 per cent of the volume of each particle. For most estimating purposes the volume-weight relation on a dry basis may be taken as 50 lb per cubic foot although in the smaller sizes the packed weight may be as much as 64 lb per cubic foot.

Silicon Dioxide (Silica), in a prepared form obtained by suitable mixing of sulphuric acid with sodium silicate, is another solid adsorbent and is commonly called silica gel. Its capillary structure is exceedingly small, so small that its exact structure has to be deduced as it cannot be observed. It has high adsorptive capacity per unit of weight; it is non-toxic, and may be repeatedly re-activated at temperatures up to 600 F without practical deterioration. Re-activation is generally accomplished with air or other gases at temperatures not over 350 F. Volume of the capillary pores is reported to be from 50 to 70 per cent of the total solid volume. Silica gel is available commercially in a wide variety of sizes of graded granules ranging from 3 to 8 mesh granules to impalpable powder (through 325 mesh). The 6 to 16 mesh granules are generally used in dehumidification applications and will vary in density from 40 to 45 lb per cubic foot.

There are other solid substances having marked adsorbent properties, but details concerning them are not available. Most adsorbents have the ability to absorb some gases and condensible vapors other than water vapor, a property sometimes useful in air conditioning applications, and one which accounts for the extensive use of adsorbent materials in the field of chemistry.

Air to be dehumidified is drawn or blown through a screened bed of dry, solid adsorbent and the water vapor in the air-vapor mixture is caught and retained in the pores of the medium. The exact nature of the process which goes on during adsorption is not known but it is stated that the action is brought about by surface condensation, and also by a difference between the vapor pressure of the water condensing inside the pores and the partial pressure of the water vapor in the air-vapor mixture. The adsorbing process in the bed can continue until the vapor pressures reach equilibrium. The amount of vapor adsorbed will depend on the

characteristic of the adsorbent being used, the temperature of the air and the bed, the vapor pressure of the water vapor in the air being passed through it, and its final vapor pressure or moisture content.

As the bed of material adsorbs moisture, its vapor pressure approaches that of the contacting air and the efficiency of adsorption gradually decreases. Equilibrium throughout the whole bed will not be reached for an extended period depending on bed thickness. Because of this diminishing efficiency of adsorption, commercially designed systems do not approach the state of equilibrium, but generally operate on a predetermined cycle or contact time and at partial saturation.

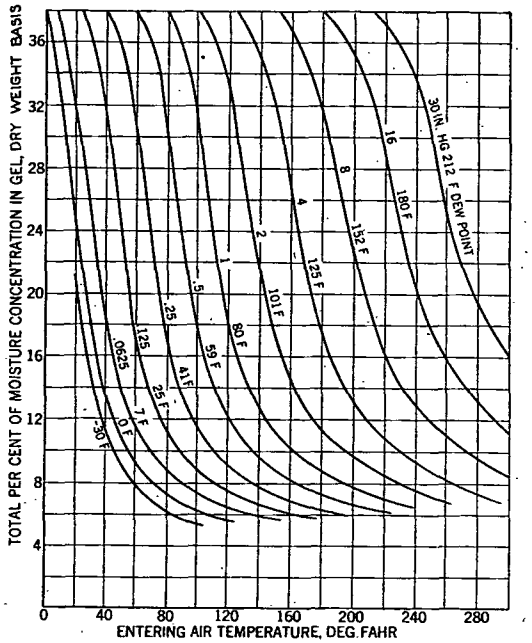


FIG. 1. TEMPERATURE—VAPOR PRESSURE—CONCENTRATION RELATION FOR A SILICA GEL BED AT CONSTANT TEMPERATURE

As the process of adsorption goes on heat is liberated in the bed. The total heat so liberated consists of the latent heat of the water vapor condensed together with the so-called heat of wetting of the adsorbent. The heat of wetting varies with the substance used as adsorbent while the latent heat of condensation depends only on the temperature and pressure of the water vapor.

Since the adsorptive ability of an adsorbent depends on the temperature of the bed and on the vapor pressure difference between the pores and the air-vapor mixture, it is important to know the pressures and temperatures at which pressure equilibrium is reached.

Evidently the equilibrium conditions represent the limits beyond which adsorption of vapor cannot continue. This relationship for one com-

mercial silica gel of 0.70 specific gravity is shown graphically in Fig. 1. These curves indicate the general manner in which adsorbents may be expected to perform, though the exact form and values of the curves will vary for each specific adsorbent and may vary even within the same generic group.

As an example in the interpretation of the chart consider the case when moist air at a temperature of 80 F and a vapor pressure of 0.5 in. of mercury flows through a bed of silica gel which is at a temperature of 80 F. The chart indicates that the equilibrium of pressure between the air-vapor mixture and the bed is reached when the dry bed has adsorbed moisture to the extent of 30 per cent of the weight when dry. When this happens the bed can adsorb no more moisture unless its temperature is decreased or the air vapor pressure increased.

The effect of temperature upon the adsorptive capacity of silica gel may be observed by following the 59 F dew-point line in Fig. 1. At 80 F gel temperature equilibrium is reached at 30 per cent moisture concentration while at 120 F and 200 F equilibrium is reached at 12 and 6.6 per cent moisture respectively.

Under normal reactivation temperatures the residual water content of silica gel is between 5 and 6 per cent.

In practice, the temperature rise in the dehumidified air caused by the adsorption heat is approximately 10 deg for each grain of moisture removed per cubic foot of air at atmospheric pressure. This temperature rise occurs progressively through the adsorbent bed and is an important consideration in predetermining the performance of a given design of apparatus. Data such as these together with information covering other characteristics, such as specific heat, resistance to air flow, etc., are of value in the basic design of adsorption apparatus. In the solution of air conditioning problems, however, reference must be made to performance data on established apparatus designs such as are presented later in this chapter.

Absorbents

Any absorbent substance may be used as a dehumidifying agent if it has a vapor pressure lower than the vapor pressure in the air-vapor mixture from which the moisture is to be removed.

Solid Absorbents. The substances used are in general the solid forms of the liquid absorbents. Calcium chloride is frequently used because of low cost. At present they are used principally in small dessicating chambers, and in small dryers of the cartridge type, through which air is forced under pressure.

Liquid Absorbents. These are characteristically water solutions of materials in which the vapor pressure is reduced to a suitable level by governing the concentration and temperature of the dehydrating solution. In addition to having suitable vapor pressure characteristics a practical absorbent must also be widely available at economical cost, be non-corrosive, odorless, non-toxic, non-inflammable, chemically inert against any impurities in the air stream, stable over the range of use and especially it must not precipitate out at the lowest temperature to which the apparatus is exposed. It must have low viscosity and be capable of being economically regenerated or concentrated after having been diluted by absorbing moisture.

Water solutions, or brines, of the chlorides or bromides of various inorganic elements such as lithium chloride and calcium chloride are the absorbents most frequently used in connection with air conditioning applications and detailed attention is confined to these two in this chapter.

The application consists of bringing the air-vapor stream into intimate

contact with the absorbent, permissibly by passing the air stream through a finely divided spray of the brine but more generally by passing the air over a contacting pack where the liquid absorbent presents a large surface to the air stream. The difference in vapor pressure causes some of the vapor in the air-vapor mixture to migrate into the brine. Here it condenses into liquid water and decreases the concentration of the absorbent.

As the water vapor is added to the absorbent and condenses, it gives up its latent heat of condensation. For every pound of water absorbed and condensed the heat released is obtainable from steam tables. For in-

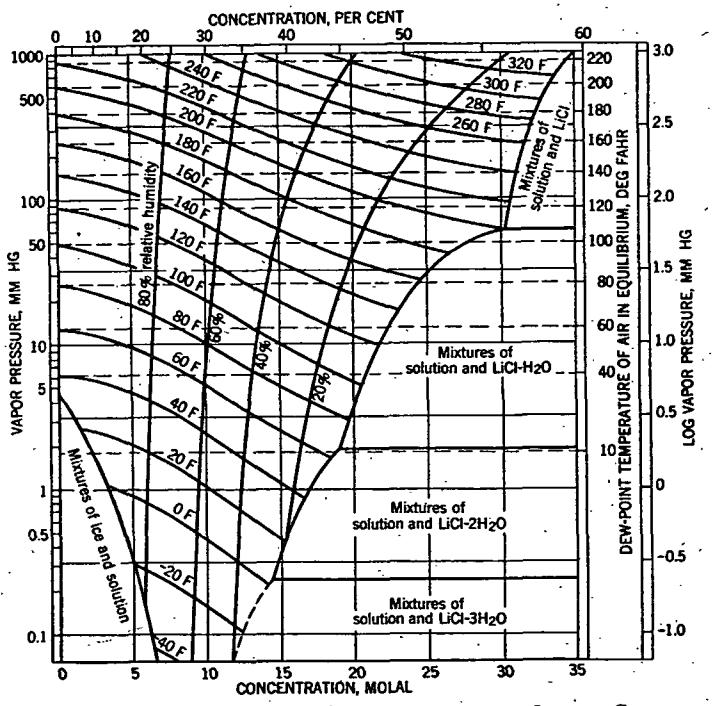


FIG. 2. TEMPERATURE—PRESSURE—CONCENTRATIONS FOR LITHIUM CHLORIDE

stance, at a dew-point temperature of 60 F the amount of this heat is about 1057 Btu. In addition to this heat there is involved also the so-called heat of mixing which is frequently considerable. The heat thus produced in the contacting packs of the average air conditioning installation is to a large extent transferred to the chemical solution itself and the balance to the air being dried.

A more complete cycle involves heat removal from the contacting medium, either within or external to the interchanger. Thus the temperature of the medium may be higher than, equal to, or lower than that of the air, depending on the agent used and the function to be performed. In such a cycle, the process may be accompanied by cooling or heating, or either, and such effect, if present, may be either a necessary by-product of the process, or for the specific purpose of obtaining both latent and

sensible heat removal simultaneously. In many industrial applications it is not necessary to aftercool the air. In applications requiring low dry-bulb temperatures excess sensible heat must be removed by an aftercooler.

Since the absorption process can continue only as long as there is a difference in vapor pressure between the absorbent and the air-vapor mixture and since at a given temperature of the absorbent the vapor pressure depends on the concentration of the solution, evidently there must be a relation between these quantities which, if known, would state

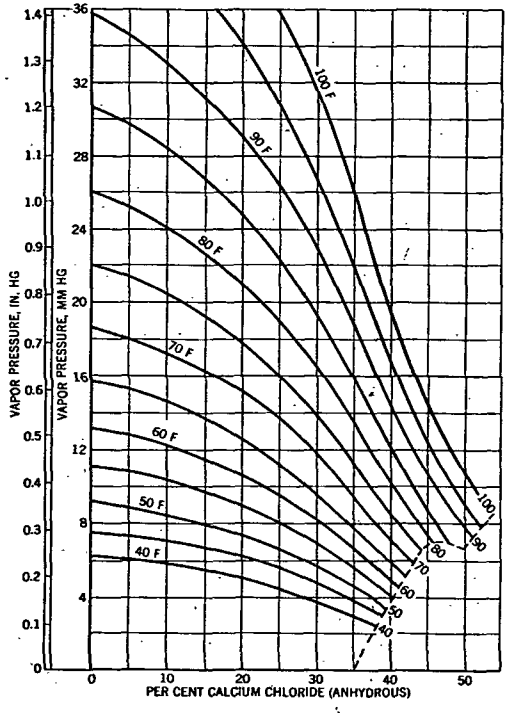


FIG. 3. TEMPERATURE—PRESSURE—CONCENTRATIONS FOR CALCIUM CHLORIDE

the limits of the process. The relationship would also depend on the absorbent being used, and would have to be determined for each substance used as an absorbent. This relationship is shown graphically in Fig. 2 for lithium chloride, and Fig. 3 presents similar data for calcium chloride. These charts are essentially similar to Fig. 1, and their direct usefulness is limited by much the same considerations. Other physical properties of lithium chloride are shown in Tables 1, 2 and 3.

In Fig. 2 and Table 1 the unit of concentration is the *mol*. An M molal solution is defined as a solution containing $M \times 42.37$ grains of anhydrous lithium chloride per 1000 grains of water. The formula connecting concentration in mols with weight in per cent is equivalent to: $(100 \times M \times 42.37) \div [1000 + (M \times 42.37)]$.

TABLE 1. PROPERTIES OF LITHIUM CHLORIDE SOLUTIONS

Concentration Pound Mols (42.4 Lb) *LiCl* per 1000 Lb Water	Concentration Per Cent by Weight	Specific Gravity at 100 F	Viscosity (Millipoise) 80 F	Viscosity (Millipoise) 180 F	Partial Heat of Mixing at 0 F, Btu per Lb	Temp. Coeff. of Partial Heat of Mixing Btu per Lb per F	Specific Heat at 70 F	Boiling Point F at (760 mm Hg)	Freezing Point, F
0	0.0	1.000	8.61	3.48	0.00	0.000	0.998	212.0	32.0
2	7.8	1.037	11.19	4.56	2.04	−0.014	0.901	215.8	16.3
4	14.5	1.076	14.42	6.01	7.24	−0.036	0.831	221.5	− 5.8
6	20.2	1.111	18.62	7.78	16.70	−0.069	0.778	228.9	−34.2
8	25.3	1.143	24.32	10.00	31.90	−0.109	0.739	238.1	−69.0
10	29.7	1.172	32.28	12.91	51.10	−0.143	0.710	248.4	−90.0
12	33.7	1.199	43.45	16.56	75.70	−0.160	0.687	258.8	−40.0
14	37.4	1.225	60.26	21.28	90.80	−0.167	0.666	268.9	1.0
16	40.4	1.248	82.04	27.10	124.80	−0.176	0.647	277.9	36.5
18	43.3	1.270	113.80	35.48	145.00	−0.186	0.631	285.8	58.1
20	46.0	1.291		46.45	162.00	−0.194		293.2	86.4
22	48.4			60.67	171.00	−0.200		300.2	133.0
24	50.3			84.33	177.00	−0.200		307.0	156.0
26	52.4				182.00	−0.210		313.0	180.0
28	54.3				191.00	−0.210		318.0	190.0
30	56.1				194.00	−0.210		323.0	195.0
32	57.5				198.00	−0.220		328.0	280.0

TABLE 2. DEW-POINT OF AIR IN EQUILIBRIUM WITH LITHIUM CHLORIDE SOLUTIONS
CONCENTRATION IN POUND MOLS (42.4 LB) LITHIUM CHLORIDE PER 1000 LB WATER

Dew-Point at Zero Conc.	Concentration of Lithium Chloride 2.0	4.0	6.0	8.0	10.0	12.0	14.0	16.0	18.0	20.0	22.0	24.0	26.0	28.0	30.0
320	315.2	308.7	299.9	290.2	279.7	269.4	259.6	251.5	244.1	236.5	230.0	223.8	218.6	214.5	210.3
300	295.4	289.1	280.5	270.9	260.6	250.5	240.8	232.6	225.4	218.0	211.8	205.8	200.8	196.9	192.8
280	275.6	269.5	261.1	251.7	241.5	231.6	222.2	214.0	206.7	199.7	193.5	187.8	183.2	179.3	175.2
260	255.8	250.0	241.9	232.6	222.7	212.8	203.5	195.5	188.4	181.7	175.4	170.0	165.6	162.0	158.4
240	236.0	230.4	222.5	213.5	203.8	194.2	185.0	177.1	170.0	163.6	157.5	152.2	148.3	144.6	140.5
220	216.2	210.8	203.2	194.4	184.9	175.5	166.4	158.6	151.6	145.3	139.6	134.6	130.7	127.3	124.2
200	196.4	191.2	183.9	175.4	166.1	156.7	148.0	140.3	133.5	127.3	121.9	117.0	113.3	110.1	
180	176.6	171.6	164.7	156.4	147.3	138.1	129.6	122.1	115.5	109.4	104.2	99.6	96.0		
160	156.8	152.1	145.4	137.4	128.6	119.7	111.3	103.9	97.4	91.6	86.6	82.2			
140	137.0	132.6	126.1	118.4	109.9	101.3	93.1	85.9	79.5	73.8	69.0				
120	117.2	113.0	106.8	99.4	91.1	82.7	74.7	67.8	61.5	56.0					
110	107.3	103.2	97.2	89.9	81.9	73.5	65.6	58.8	52.6	47.1					
100	97.4	93.4	87.5	80.5	72.7	64.4	56.6	49.8	43.7	38.2					
90	87.5	83.6	77.9	71.0	63.3	55.2	47.6	40.8	34.8	29.3					
80	77.6	73.8	68.4	61.6	54.0	46.1	38.5	31.8	25.9	20.6					
70	67.7	64.0	58.7	52.2	44.8	37.0	29.5	22.9	17.2	12.0					
60	57.8	54.3	49.1	42.7	35.5	27.9	20.5	14.0	8.3						
40	38.0	34.7	29.9	23.9	16.9	9.6	2.4	−3.9							
20		15.1	10.7	5.0	−1.7	−8.7	−15.4								
0		−4.5	−8.6	−13.9	−20.2	−27.0	−33.3								

TABLE 3. DENSITY OF LITHIUM CHLORIDE SOLUTIONS

CONCENTRATION, POUND MOLS (42.4 LB) *LiCl* PER 1000 LB WATER	TEMPERATURE F DEG						
	0	50	100	150	200	250	300
0							
2		1.045	1.037	1.026	1.012		
4	1.090	1.085	1.076	1.064	1.052		
6	1.124	1.119	1.111	1.100	1.087		
8	1.156	1.150	1.143	1.132	1.122		
10	1.188	1.181	1.172	1.162	1.152	1.142	
12	1.217	1.209	1.199	1.188	1.178	1.168	
14	1.242	1.235	1.225	1.214	1.203	1.192	
16		1.257	1.248	1.236	1.226	1.215	
18		1.279	1.270	1.259	1.248	1.237	
20			1.291	1.280	1.279	1.568	
22				1.310	1.289	1.278	1.267
24				1.317	1.307	1.296	1.286
26					1.313	1.312	1.302
28					1.338	1.327	1.318
30						1.34	1.33
32							1.35

Example 1. Determine the dew-point, wet-bulb, per cent saturation and humidity ratio of air in equilibrium at 100 F with pure lithium chloride solution of density 1.270.

Solution. From Table 1 the concentration of a solution of density 1.270 at 100 F is 18.0 *M*. From Fig. 2 the dew-point of 18 *M* lithium chloride at 100 F is 43.7 F. From Table 1, Chapter 3, the partial pressure of water over the solution is 0.2857 in. of Hg, and the humidity ratio is 0.00602 lb per pound dry air. From the Mollier Diagram the wet-bulb is 66.3 F, and the per cent saturation is 14 per cent.

Example 2. Determine the boiling point, and freezing point of 18 *M* lithium chloride solutions.

Solution. From Table 1, boiling point (standard) is 285.8 F, freezing point is 58.1 F.

Example 3. Calculate the heat of vaporization of 1 lb of water from a large amount of 18 *M* lithium chloride solution at the boiling point.

Solution. The heat of boiling is equal to the heat of mixing plus the heat of boiling pure water at the same temperature. The heat of mixing from Table 1 at 18 *M* and 285.8 F is 145 − (0.186 × 285.8) = 92 Btu per pound. The heat of vaporization of water from steam tables at 285.8 F is 920 Btu per pound. Therefore the heat of vaporization of water from the solution is 920 + 92 = 1012 Btu per pound.

Example 4. One thousand pounds of air per minute at 100 F dry-bulb with a dew-point of 70 F and 36.7 per cent saturation are passed over 18 *M* lithium chloride solution. The rate of flow of the solution is 200 gpm and the entering temperature is 80 F. The air leaves the absorber at 85 F dry-bulb and dew-point of 35 F. Calculate (*a*) the heat to be removed from the lithium chloride solution to maintain these conditions, and (*b*) the temperature rise of the solution in passing through the absorber.

Solution. (*a*) The enthalpy of the entering air at 100 F dry-bulb and 36.7 per cent saturation = $h_a + \mu h_{as}$ = 24.029 + (0.367 × 47.70) = 41.54 Btu per pound (Table 1, Chapter 3).

The per cent saturation of the air leaving at 85 F dry-bulb and 35 F dew-point is 16.2. Enthalpy of leaving air = 20.42 + 0.162 × 29.01 = 25.12 Btu per pound (Table 1, Chapter 3).

Heat to be extracted from air = 1000 (41.54 − 25.12) = 16,420 Btu per minute. Heat of mixing = 145 − (0.186 × 80) = 130 Btu per pound of moisture removed. From Table 1, Chapter 3, the moisture removal per pound of air = 0.01582 − 0.00428 = 0.01154 lb. Heat of mixing for 1000 lb of air = 1000 × 0.01154 × 130 = 1500 Btu. Total heat extraction = 16,420 + 1500 = 17,920 Btu per minute.

(*b*) The weight of solution circulated is 200 × 1.27 (Table 1) × 8.33 = 2116 lb per minute. Its heat capacity = 2116 × 0.631 (Table 1) = 1335 Btu per (minute) (Fahrenheit degree). The temperature rise = 17,920 ÷ 1335 = 13.4 F.

DEHUMIDIFICATION WITH SOLID SORBENTS

One type of equipment suitable for producing dehumidification with solid drying agents utilizes an apparatus with continuously rotating beds or dampers as illustrated in Fig. 4. The apparatus consists essentially of a cylinder or drum filled with a dehumidifying or drying agent. Air flow through the drum is directed by baffles which permit three independent air streams to flow through the adsorbing material. One air stream consists of the wet air which is to be dehumidified. The second is heated activation air used for drying that part of the dehumidifying material which has become saturated. The third air stream pre-cools the bed to permit an immediate pickup of moisture when that part of the bed returns to the dehydration cycle.

In the rotating bed apparatus, the baffle sheets are stationary and the screened bed rotates at a definite speed to permit the proper time of contact in the activation, cooling and dehumidifying cycles. In the

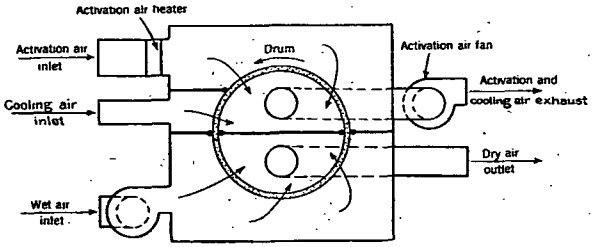


FIG. 4. SOLID ADSORBENT DEHUMIDIFIER—ROTATING BED TYPE

rotating damper apparatus, the bed remains stationary and a sectionalized damper rotates. This rotating damper produces the same general effect as if the stationary baffles previously mentioned rotated.

Clean air for activation is supplied at temperatures normally ranging from 300 to 350 F. Any source of clean heated air can be used such as air heated by electric heaters or steam coils, or air indirectly heated by coal or oil fired interchangers. Direct fired heaters are usually designed for gas since there must be no condensable, tarry, combustion products to contaminate the adsorbent. To avoid excessive contamination or clogging of the absorbent bed by dust it is frequently desirable to provide filters for the activation air.

Where activation air heating coils are supplied with steam at 80 lb per square inch gage pressure or higher, activation efficiency will remain normal and moisture removal capacity will not be reduced. When lower steam pressures are used, incomplete reactivation will result in reduced moisture removal.

Another type of solid adsorption equipment uses two complete sets of stationary adsorbing beds, arranged so that one set is dehumidifying the air while the other set is being activated. With the dampers in the position shown in Fig. 5, air to be dried flows through one set of beds and is dehumidified, while activation air is heated and circulated through the other set. After activation is complete, the beds are cooled by shutting off the activation air heaters and allowing unheated air to circulate through them.

After the beds have adsorbed moisture to a degree which begins to

impair performance, a timer-controller causes the dampers to rotate to the opposite side. Thus the beds which on the previous cycle were adsorbing have activation air circulated through them, and vice versa. Activation air is heated in the same manner as with continuous equipment.

DEHUMIDIFICATION WITH LIQUID SORBENTS

One type of system utilizing liquid sorbents includes an external interchanger having essential parts consisting of a liquid contactor, a solution concentrator, a solution heater and a cooling coil all as shown in Fig. 6.

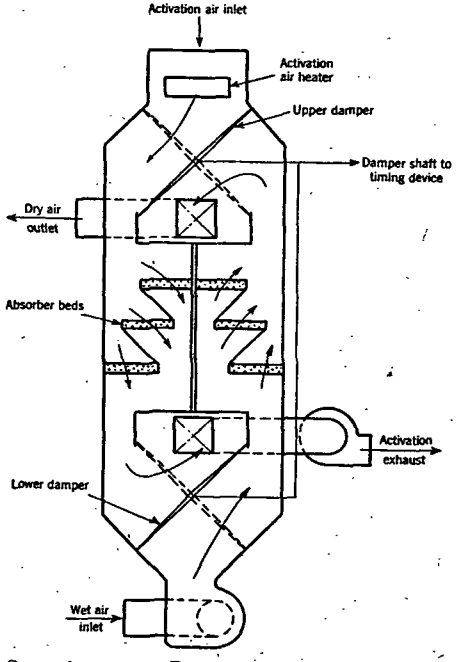


FIG. 5. SOLID ADSORBENT DEHUMIDIFIER—STATIONARY BED TYPE

The contactor and cooling coil are located in the wet air stream. The air to be conditioned is brought into contact with an aqueous brine solution having a vapor pressure below that of the entering air, resulting in a transfer of moisture (latent heat). As previously described, this results in a conversion of latent heat to sensible heat which raises the solution temperature and consequently the air temperature. The temperature change of the air being processed is determined by the cooling water temperature and the amount of moisture removed in the equipment. Control of leaving air temperature may be obtained by precooling the absorbent solution in a suitable surface cooler, by tap, well, or artificially chilled water.

The excess water of condensation, which dilutes the brine, is removed in the solution concentrator. This is a low pressure steam heat exchanger which over-concentrates a portion of the weak liquor and returns it to

the main brine reservoir for re-pumping. The concentrator operates in the manner of an evaporative condenser, whereby moisture is evaporated from the brine, by the heating coils, into a stream of regeneration air taken from and rejected to the outside atmosphere. Low pressure steam is normally used for heating the brine. When it is desirable or necessary to use gas or electricity, an auxiliary low pressure steam boiler is usually added to the equipment. Concentrators operating on a simple boiler principle have not as yet been commercially practical.

It should be noted that the solution concentration phase is the reverse of the absorption process. During concentration the aqueous vapor pressure of the solution is greater than that of the surrounding air, while during dehumidification, the reverse is the case. Utilization of this principle permits winter humidification, by heating (instead of cooling) the solution pumped to the contactor. Water is thereby evaporated

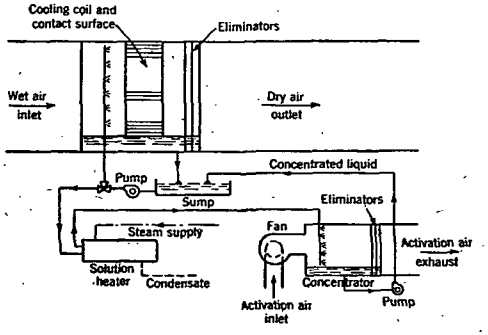


FIG. 6. LIQUID ABSORBENT EQUIPMENT IN WHICH SOLUTION COOLER AND CONTACTOR ARE COMBINED

into, instead of being condensed out of, the conditioned air stream. This requires dilution of the brine externally to the contactor, rather than concentration.

EQUIPMENT AUXILIARIES AND CONTROLS

Precoolers. When cold water is available, it is generally economical to use this water in a precooling coil in the warm, humid air stream. Any dehumidifying accomplished by this coil reduces the load; and moreover, lowering the temperature of the inlet air results in a higher moisture removal efficiency.

Dry Air Coolers. Particularly with the solid adsorbent process, and to a lesser extent with liquid absorbents, a dry air cooler may be employed to remove sensible heat from the dehumidified air whenever it leaves the apparatus at an elevated temperature. A cooling coil using city water is usual practice, and is considered economical whenever the difference between effluent air and entering water temperatures is greater than 15 F.

Controls. The use of direct or chemical dehumidifying equipment makes possible the use of a relatively simple control system with a humidistat or, alternatively, a wet-bulb controller, to regulate the operation of the machine, and a thermostat to control the sensible cooling apparatus. Functionally, the relative humidity control may consist of one of the following:

1. *Stop—Start*—Where the humidistat starts the dehumidifier on rising humidity and stops it on falling humidity.

2. *By-pass*—Where the humidistat modulates face and by-pass dampers located at the wet air inlet of the dehumidifier. Thus the quantity of air passing through the dehumidifier is proportioned in accordance with the change in latent heat load.

3. *Vapor Pressure Control* (used with liquid absorbents)—Where the humidistat directly controls the temperature or concentration of the contacting solution, thereby matching the heat or moisture removal to the load requirement.

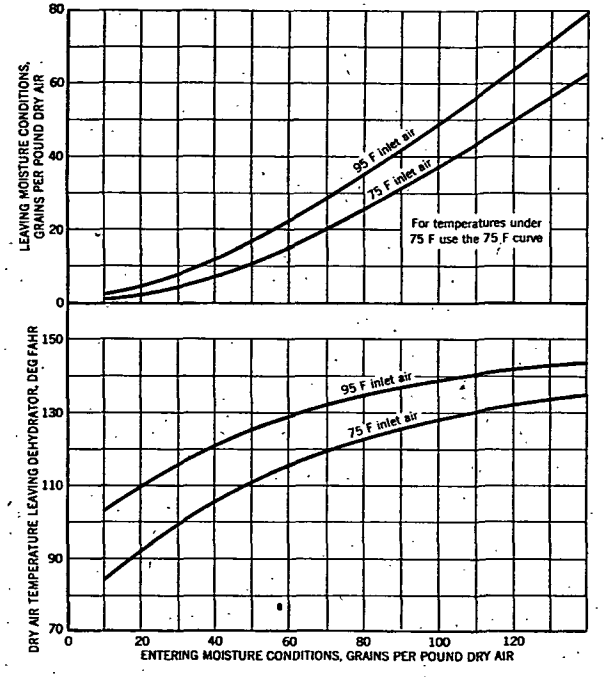


FIG. 7. PERFORMANCE DATA FOR ROTATING BED DIRECT GAS FIRED SILICA GEL DEHUMIDIFIER

EQUIPMENT PERFORMANCE

It is recognized that, whereas the curves relating temperature and vapor pressure of the several dehumidifying agents (Figs. 1, 2 and 3) accurately define the equilibrium limits for these materials, these curves cannot be used for predicting performance of available equipment. This occurs because (*a*) the materials themselves can only be utilized efficiently within certain ranges of moisture concentration, and (*b*) the degree to which the vapor pressure of the air being treated approaches that at the surface of the material depends upon the completeness of the contact. It is obvious, for example, that the thickness of a bed of solid adsorbent for a given air flow will have a profound effect upon the degree of moisture removal attained. However, it will have an inverse effect upon the

power required to drive air through the bed. It must also be borne in mind that the moisture removal efficiency of all solid adsorbents and absorbent solutions is related to the temperature of the medium as well as to its state of unsaturation. On the other hand, it will be noted that the lower limits of dehumidification, which can be obtained with absorbent solutions, are likely to be controlled by the equilibrium dew-point at the lowest temperature at which the solution will resist freezing or crystallization. For this reason, actual moisture removing capacity is determined from performance curves of the several materials under practical con-

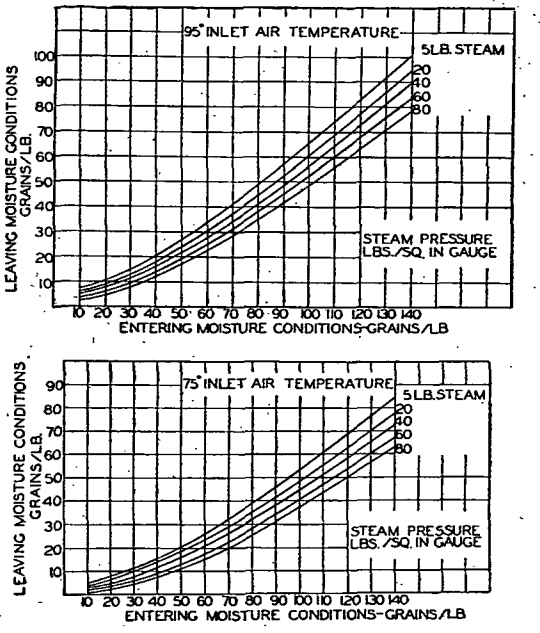


FIG. 8. SILICA GEL DEHUMIDIFIER PERFORMANCE DATA STEAM ACTIVATION TYPE

ditions of temperature, concentration and contacting efficiency as shown in Figs. 7, 8, 9 and 10. While these curves by no means explore all the performance possibilities, they may be considered to be representative of sound design and application practice.

Performance data for standard production designs of silica gel *rotating bed* dehumidifiers are shown in Fig. 7 for a *direct gas fired type*, and in Fig. 8 for *steam activation types* for steam pressures of 5 to 80 lb per square inch. They are designed for normal comfort cooling and moderate humidity industrial air conditioning applications. Performance data for a *heavy duty type activated alumina* dehumidifier are given in Fig. 9. This unit has extended surface coils, imbedded in the adsorber, which are supplied with high pressure steam for heating during activation and cool water for direct bed cooling during adsorption. Capacities shown in

Fig. 9 are based upon 75 F cooling water. Performance data for *standardized package lithium chloride* equipment are shown in Fig. 10. This is a single unit including both absorption contactor and regeneration section together with necessary heat exchangers.

COMPARISON OF METHODS OF DEHUMIDIFICATION

Almost all summer comfort air conditioning, as well as much industrial and commercial air conditioning, requires both of the functions of sensible and latent heat removal from air. Each of the methods of cooling and dehumidification has as its objective either the removal of sensible or latent heat, or both simultaneously. Choice of method and medium therefore depends solely on whether (*a*) method and medium are physically able to accomplish the desired result with practical equipment, and (*b*) method and medium are justifiable economically.

Referring particularly to the problem of moisture removal, it may be stated that dehumidification either by using chilled water, brine or direct

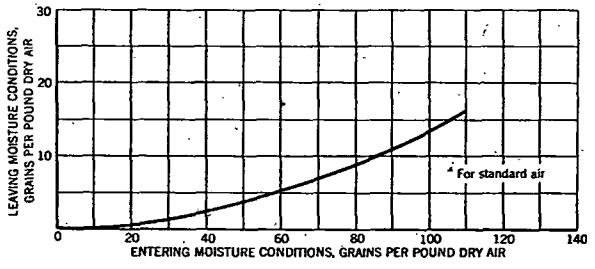


FIG. 9. ACTIVATED ALUMINA DEHUMIDIFIER PERFORMANCE DATA HEAVY DUTY TYPE

expansion refrigerant, or by using solid or liquid drying agents, is *equally practical* from the viewpoint of engineering performance for the vast majority of comfort and for a great many industrial applications requiring dew-point temperatures above 45 F.

For this reason, in the normal operating range, consideration is usually based upon practical factors which affect the over-all economics of the situation or upon the desire or necessity of attaining independent control of either temperature or humidity. In this connection, it should be remembered that though a satisfactory balance may exist between cooling and dehumidification requirements at maximum design conditions, there are many periods when practical operating conditions will result in an unsatisfactory balance of sensible and latent loads unless there is a simultaneous use of cooling and reheating.

In the normal range the choice between direct or chemical dehumidification, mechanical refrigeration, natural cold water, or ice, must be justified by the initial investment of available equipment; the availability and cost of prime energy sources; the charges to be allocated to space occupied, labor of operation and maintenance; and the degree of control required.

When required dew-point temperatures are below 45 F, direct or

chemical methods deserve special consideration, as they do in applications where the task is primarily one of moisture removal with no necessity for sensible temperature reduction.

It is evident that it is not possible to set forth definite rules governing the choice of the method of dehumidification. It is only possible to state certain general conditions.

Sorbent methods tend to be favorable where:

1. Low temperature water is available (below 65 F), to meet sensible cooling requirements.

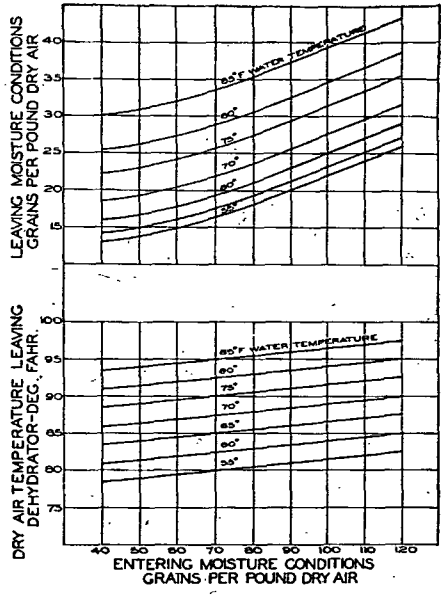


FIG. 10. LITHIUM CHLORIDE EQUIPMENT PERFORMANCE DATA

2. Gas or steam is available at costs competitive with electricity.

3. Abnormally high internal latent heat load is encountered.

4. Abnormally low dew-points are required (40 F or lower).

5. Required dry-bulb temperature is high or control of temperature is unimportant in comparison to maintenance of proper humidity.

6. In low temperature dryers where complementary heat exchange can be utilized. (In such cases the sensible heat of the dry air is reduced by the evaporation of moisture from the product being dried.)

Of the factors enumerated, Item 1, coupled with Item 2, tends to have the greatest influence since these factors affect the applicability to normal range requirements. However, each of the items listed has a direct influence upon economic considerations.

Sorbent methods tend to be unfavorable where:

1. Normal comfort dew-point temperatures are required with a predominantly sensible heat load and where mechanical refrigeration is required for sensible heat removal.

2. Water temperature is too high for practical sensible heat removal (above 65 F).

3. Cold water (below 55 F) is available in adequate quantities so that it can be directly used for both sensible and latent heat removal, or can be further chilled cheaply by mechanical refrigeration.

4. Electricity is low in cost compared with gas or steam.

No single unfavorable item listed will necessarily disqualify a method, but generally there will be several offsetting favorable factors required to make it the choice on a purely economic basis.

When analyzed with respect to the broadening scope of applications for air conditioning, it is now evident that sorbent dehumidification can be used, within its legitimate economic limits, for human comfort, commercial drying or storage of food products requiring low humidities, and for industrial processing and drying.

It provides an additional choice as to the type of equipment best suited to meet the requirements of special applications and conditions. Particular attention is called to those industrial and drying applications in which the dried air can be used at effluent temperature without further treatment.

BIBLIOGRAPHY

Direct Evaporative Cooling for Homes in the Southwest, by A. J. Rummel (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 381).

Chemical Dehumidification Agents, by F. R. Bichowsky (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1940, p. 627).

CHAPTER 39

Refrigeration

Theory and Cycle Calculations: Definitions, Simple Saturation Cycles, Complex Refrigeration Cycles, Pipe Sizes and Friction Losses, the Steam Jet System, the Absorption System, Ice Systems, Storage Systems, the Reverse Cycle; Equipment and Arrangements: Types of Compressors, Condensers, Expansion Valves, Evaporators and Coolers, Equipment 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 some knowledge on his part 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 two parts, the first covering the fundamental technical relationships which govern the selection and analysis of an operating cycle and the second presenting brief discussions of some of the equipment and equipment arrangements which are most commonly used.

THEORY AND CYCLE CALCULATIONS

Definitions

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.

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—which constitutes the working effectiveness of the refrigerant. Further, since the work required to compress a vapor increases rapidly with the pressure range, 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 pressure. 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 is that linear distances on the chart correspond to energy gains or losses and the two types of processes, constant-pressure and constant-enthalpy, which occur most frequently in refrigeration cycles, can both be represented by straight vertical or horizontal lines. Tabular arrangements of refrigerant properties have the disadvantage common to all point representations, in requiring interpolation between given values, but have the advantage of an accuracy greater than that obtainable from a chart. Tables 1, 2, 3, 4, 5, 6 and 7 give the thermodynamic properties of seven of the more common refrigerants: dichlorodifluoromethane (F-12), monochlorodifluoromethane (F-22), methyl chloride, ammonia, carbon dioxide, monofluorotrichloromethane (F-11), and water; the first five of these materials are commonly used in reciprocating compressors, the last two in centrifugal machines, while water also finds use in steam jet systems.

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 specific enthalpy and specific 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 F-12 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 sub-cooled liquids. With an accuracy sufficient for all practical purposes the specific volume and specific enthalpy of any

TABLE 1. PROPERTIES OF DICHLORODIFLUOROMETHANE (F-12)

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Specific Enthalpy and Entropy Taken From −40 F							
				Specific Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Sp. En.	Entropy	Sp. En.	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
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

NOTE: Sp. En. = Specific Enthalpy.

Table 2. Properties of Monochlorodifluoromethane (F-22)

Sat Temp F	Abs Press Lb per Sq In.	Volume		Heat Content and Entropy Taken from −40 F							
				Heat Content		Entropy		50 Deg Superheat		100 Deg Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Heat Content	Entropy	Heat Content	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	12.897	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
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

Table 3. Properties of Methyl Chloride

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Specific Enthalpy and Entropy Taken From −40 F							
				Specific Enthalpy		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Sp. En.	Entropy	Sp. En.	Entropy
0	18.73	0.0162	5.052	14.4	192.4	0.0328	0.4197	215.6	0.467	237.2	0.507
2	19.60	0.0162	4.856	15.1	193.1	0.0344	0.4196	216.2	0.466	237.7	0.505
4	20.47	0.0163	4.661	15.8	193.8	0.0360	0.4195	216.7	0.465	238.2	0.504
5	20.91	0.0163	4.563	16.2	194.1	0.0368	0.4195	217.0	0.464	238.5	0.503
6	21.39	0.0163	4.476	16.6	194.4	0.0376	0.4194	217.3	0.464	238.8	0.502
8	22.34	0.0164	4.303	17.3	195.1	0.0391	0.4193	217.9	0.463	239.4	0.501
10	23.30	0.0164	4.129	18.1	195.8	0.0407	0.4192	218.5	0.463	240.0	0.500
12	24.38	0.0164	3.984	18.8	196.3	0.0423	0.4184	219.0	0.462	240.5	0.499
14	25.46	0.0164	3.839	19.6	196.7	0.0439	0.4176	219.5	0.462	241.0	0.498
16	26.55	0.0165	3.693	20.3	197.2	0.0454	0.4168	220.0	0.461	241.5	0.498
18	27.63	0.0165	3.548	21.1	197.6	0.0472	0.4160	220.5	0.461	242.0	0.497
20	28.71	0.0166	3.403	21.8	198.1	0.0486	0.4152	221.0	0.460	242.5	0 496
22	29.98	0.0166	3.288	22.5	198.5	0.0501	0.4148	221.5	0.459	243.0	0.495
24	31.25	0.0166	3.172	23.3	198.9	0.0516	0.4143	222.0	0.459	243.6	0.495
26	32.53	0.0167	3.057	24.0	199.3	0.0532	0.4139	222.4	0.458	244.1	0.494
28	33.80	0.0167	2.941	24.8	199.7	0.0547	0.4134	222.9	0.458	244.7	0.494
30	35.07	0.0168	2.826	25.5	200.1	0.0562	0.4130	223.4	0.457	245.2	0.493
32	36.55	0.0168	2.734	26.2	200.5	0.0577	0.4124	223.9	0.456	245.7	0.492
34	38.03	0.0169	2.642	27.0	200.9	0.0592	0.4118	224.3	0.455	246.2	0.492
36	39.51	0.0169	2.549	27.7	201.4	0.0607	0.4111	224.8	0.455	246.7	0.491
38	40.99	0.0169	2.457	28.5	201.8	0.0622	0.4105	225.2	0.454	247.2	0.491
39	41.73	0.0170	2.411	28.8	202.0	0.0629	0.4102	225.5	0.453	247.4	0.490
40	42.47	0.0170	2.365	29.2	202.2	0.0637	0.4099	225.7	0.453	247.7	0.490
41	43.33	0.0170	2.328	29.6	202.4	0.0644	0.4096	225.9	0.453	248.0	0.490
42	44.18	0.0171	2.290	29.9	202.6	0.0651	0.4093	226.1	0.452	248.3	0.489
44	45.89	0.0171	2.216	30.7	203.0	0.0666	0.4087	226.6	0.451	248.8	0.489
46	47.61	0.0171	2.141	31.4	203.3	0.0680	0.4081	227.0	0.451	249.4	0.488
48	49.32	0.0172	2.067	32.2	203.7	0.0695	0.4075	227.5	0.450	249.9	0.488
50	51.03	0.0172	1.992	32.9	204.1	0.0709	0.4069	227.9	0.449	250.5	0.487
52	53.00	0.0172	1.931	33.7	204.4	0.0724	0.4063	228.2	0.448	251.0	0.486
54	54.97	0.0173	1.870	34.4	204.7	0.0739	0.4056	228.6	0.448	251.5	0.486
56	56.94	0.0173	1.810	35.2	205.1	0.0754	0.4050	228.9	0.447	252.0	0.485
58	58.91	0.0173	1.749	35.9	205.4	0.0769	0.4043	229.3	0.447	252.5	0.485
60	60.88	0.0174	1.688	36.7	205.7	0.0784	0.4037	229.6	0.446	253.0	0.484
62	63.13	0.0174	1.638	37.4	206.0	0.0798	0.4030	229.9	0.445	253.5	0.483
64	65.37	0.0174	1.588	38.2	206.3	0.0812	0.4024	230.3	0.444	254.0	0.483
66	67.62	0.0175	1.539	38.9	206.6	0.0827	0.4017	230.6	0.443	254.5	0.482
68	69.86	0.0175	1.489	39.7	206.9	0.0841	0.4011	231.0	0.442	255.0	0.482
70	72.11	0.0176	1.439	40.4	207.2	0.0855	0.4004	231.3	0.441	255.5	0.481
72	74.66	0.0176	1.398	41.1	207.5	0.0869	0.3998	231.6	0.440	256.0	0.480
74	77.21	0.0177	1.357	41.9	207.7	0.0883	0.3992	232.0	0.439	256.5	0.480
76	79.76	0.0177	1.315	42.6	208.0	0.0898	0.3985	232.3	0.439	256.9	0.479
78	82.31	0.0178	1.274	43.4	208.2	0.0912	0.3979	232.7	0.438	257.4	0.479
80	84.86	0.0178	1.233	44.1	208.5	0.0926	0.3973	233.0	0.437	257.9	0.478
82	87.74	0.0178	1.199	44.8	208.7	0.0940	0.3967	233.3	0.436	258.4	0.478
84	90.62	0.0179	1.165	45.6	209.0	0.0953	0.3960	233.6	0.435	258.9	0.477
86	93.50	0.0179	1.130	46.3	209.2	0.0967	0.3954	233.9	0.435	259.4	0.477
88	96.38	0.0180	1.096	47.1	209.5	0.0980	0.3947	234.2	0.434	259.9	0 476
90	99.26	0.0180	1.062	47.8	209.7	0.0994	0.3941	234.5	0.433	260.4	0.476
92	102.49	0.0180	1.033	48.6	209.9	0.1008	0.3935	234.8	0.433	260.8	0.476
94	105.72	0.0181	1.005	49.3	210.2	0.1022	0.3929	235.1	0.432	261.2	0.475
96	108.94	0.0181	0.9764	50.1	210.4	0.1035	0.3922	235.4	0.432	261.6	0.475
98	112.17	0.0182	0.9478	50.8	210.7	0.1049	0.3916	235.7	0.431	262.0	0.474
100	115.40	0.0182	0.9193	51.6	210.9	0.1063	0.3910	236.0	0.431	262.4	0.474
102	119.00	0.0183	0.8952	52.3	211.1	0.1076	0.3903	236.4	0.430	262.8	0.474
104	122.60	0.0183	0.8712	53.1	211.3	0.1090	0.3897	236.8	0.430	263.2	0.473
106	126.20	0.0184	0.8471	53.8	211.4	0.1103	0.3890	237.1	0.429	263.5	0.473
108	129.80	0.0184	0.8231	54.6	211.6	0.1117	0.3884	237.5	0.429	263.9	0.472
110	133.40	0.0185	0.7990	55.3	211.8	0.1130	0.3877	237.9	0.428	264.3	0.472
112	137.42	0.0185	0.7786	56.1	212.0	0.1144	0.3871	238.1	0.427	264.6	0.471
114	141.44	0.0185	0.7583	56.8	212.2	0.1157	0.3864	238.3	0.427	264.8	0.470
116	145.46	0.0186	0.7379	57.6	212.4	0.1171	0.3858	238.6	0.426	265.1	0.470
118	149.48	0.0186	0.7176	58.3	212.6	0.1184	0.3851	238.8	0.426	265.3	0.469
120	153.50	0.0187	0.6972	59.1	212.8	0.1198	0.3845	239.0	0.425	265.6	0.468

Note: Sp. En. = Specific Enthalpy.

TABLE 4. PROPERTIES OF AMMONIA

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Specific Enthalpy and Entropy Taken From −40 F							
				Specific Enthalpy		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Sp. En.	Entropy	Sp. En.	Entropy
0	30.42	0.02419	9.116	42.9	611.8	0.0975	1.3352	666.8	1.4439	720.3	1.5317
2	31.92	0.02424	8.714	45.1	612.4	0.1022	1.3312	667.6	1.4400	721.2	1.5277
4	33.47	0.02430	8.333	47.2	613.0	0.1069	1.3273	668.4	1.4360	722.2	1.5236
5	34.27	0.02432	8.150	48.3	613.3	0.1092	1.3253	668.8	1.4340	722.6	1.5216
6	35.09	0.02435	7.971	49.4	613.6	0.1115	1.3234	669.3	1.4321	723.1	1.5196
8	36.77	0.02440	7.629	51.6	614.3	0.1162	1.3195	670.1	1.4281	724.1	1.5155
10	38.51	0.02446	7.304	53.8	614.9	0.1208	1.3157	670.9	1.4242	725.0	1.5115
12	40.31	0.02451	6.996	56.0	615.5	0.1254	1.3118	671.7	1.4205	725.9	1.5077
14	42.18	0.02457	6.703	58.2	616.1	0.1300	1.3081	672.5	1.4168	726.8	1.5039
16	44.12	0.02462	6.425	60.3	616.6	0.1346	1.3043	673.4	1.4130	727.8	1.5001
18	46.13	0.02468	6.161	62.5	617.2	0.1392	1.3006	674.2	1.4093	728.7	1.4963
20	48.21	0.02474	5.910	64.7	617.8	0.1437	1.2969	675.0	1.4056	729.6	1.4925
22	50.36	0.02479	5.671	66.9	618.3	0.1483	1.2933	675.8	1.4021	730.5	1.4889
24	52.59	0.02485	5.443	69.1	618.9	0.1528	1.2897	676.6	1.3985	731.4	1.4853
26	54.90	0.02491	5.227	71.3	619.4	0.1573	1.2861	677.3	1.3950	732.4	1.4816
28	57.28	0.02497	5.021	73.5	619.9	0.1618	1.2825	678.1	1.3914	733.3	1.4780
30	59.74	0.02503	4.825	75.7	620.5	0.1663	1.2790	678.9	1.3879	734.2	1.4744
32	62.29	0.02508	4.637	77.9	621.0	0.1708	1.2755	679.7	1.3846	735.1	1.4710
34	64.91	0.02514	4.459	80.1	621.5	0.1753	1.2721	680.4	1.3812	736.0	1.4676
36	67.63	0.02521	4.289	82.3	622.0	0.1797	1.2686	681.2	1.3779	736.8	1.4643
38	70.43	0.02527	4.126	84.6	622.5	0.1841	1.2652	681.9	1.3745	737.7	1.4609
39	71.87	0.02530	4.048	85.7	622.7	0.1863	1.2635	682.3	1.3729	738.2	1.4592
40	73.32	0.02533	3.971	86.8	623.0	0.1885	1.2618	682.7	1.3712	738.6	1.4575
41	74.80	0.02536	3.897	87.9	623.2	0.1908	1.2602	683.1	1.3696	739.0	1.4559
42	76.31	0.02539	3.823	89.0	623.4	0.1930	1.2585	683.4	1.3680	739.5	1.4542
44	79.38	0.02545	3.682	91.2	623.9	0.1974	1.2552	684.2	1.3648	740.4	1.4510
46	82.55	0.02551	3.547	93.5	624.4	0.2018	1.2519	684.9	1.3616	741.3	1.4477
48	85.82	0.02558	3.418	95.7	624.8	0.2062	1.2486	685.6	1.3584	742.2	1.4445
50	89.19	0.02564	3.294	97.9	625.2	0.2105	1.2453	686.4	1.3552	743.1	1.4412
52	92.66	0.02571	3.176	100.2	625.7	0.2149	1.2421	687.1	1.3521	744.0	1.4382
54	96.23	0.02577	3.063	102.4	626.1	0.2192	1.2389	687.8	1.3491	744.8	1.4351
56	99.91	0.02584	2.954	104.7	626.5	0.2236	1.2357	688.5	1.3460	745.7	1.4321
58	103.7	0.02590	2.851	106.9	626.9	0.2279	1.2325	689.2	1.3430	746 5	1.4290
60	107.6	0.02597	2.751	109.2	627.3	0.2322	1.2294	689.9	1.3399	747.4	1.4260
62	111.6	0.02604	2.656	111.5	627.7	0.2365	1.2262	690.6	1.3370	748.2	1.4231
64	115.7	0.02611	2.565	113.7	628.0	0.2408	1.2231	691.3	1.3341	749.1	1.4202
66	120.0	0.02618	2.477	116.0	628.4	0.2451	1.2201	691.9	1.3312	749.9	1.4172
68	124.3	0.02625	2.393	118.3	628.8	0.2494	1.2170	692.6	1.3283	750.8	1.4143
70	128.8	0.02632	2.312	120.5	629.1	0.2537	1.2140	693.3	1.3254	751.6	1.4114
72	133.4	0.02639	2.235	122.8	629.4	0.2579	1.2110	694.0	1.3226	752.4	1.4086
74	138.1	0.02646	2.161	125.1	629.8	0.2622	1.2080	694.6	1.3199	753.3	1.4059
76	143.0	0.02653	2.089	127.4	630.1	0.2664	1.2050	695.3	1.3171	754.1	1.4031
78	147.9	0.02661	2.021	129.7	630.4	0.2706	1.2020	695.9	1.3144	755.0	1.4004
80	153.0	0.02668	1.955	132.0	630.7	0.2749	1.1991	696.6	1.3116	755.8	1.3976
82	158.3	0.02675	1.892	134.3	631.0	0.2791	1.1962	697.2	1.3089	756.6	1.3949
84	163.7	0.02684	1.831	136.6	631.3	0.2833	1.1933	697.8	1.3063	757.4	1.3923
86	169.2	0.02691	1.772	138.9	631.5	0.2875	1.1904	698.5	1.3040	758.3	1.3896
88	174.8	0.02699	1.716	141.2	631.8	0.2917	1.1875	699.1	1.3010	759.1	1.3870
90	180.6	0.02707	1.661	143.5	632.0	0.2958	1.1846	699.7	1.2983	759.9	1.3843
92	186.6	0.02715	1.609	145.8	632.2	0.3000	1.1818	700.3	1.2957	760.7	1.3818
94	192.7	0.02723	1.559	148.2	632.5	0.3041	1.1789	700.9	1.2932	761.5	1.3793
96	198.9	0.02731	1.510	150.5	632.6	0.3083	1.1761	701.5	1.2906	762.2	1.3768
98	205.3	0.02739	1.464	152.9	632.9	0.3125	1.1733	702.1	1.2881	763.0	1.3743
100	211.9	0.02747	1.419	155.2	633.0	0.3166	1.1705	702.7	1.2855	763.8	1.3718
102	218.6	0.02756	1.375	157.6	633.2	0.3207	1.1677	703.3	1.2830	764.6	1.3693
104	225.4	0.02764	1.334	159.9	633.4	0.3248	1.1649	703.8	1.2805	765.3	1.3668
106	232.5	0.02773	1.293	162.3	633.5	0.3289	1.1621	704.3	1.2780	766.1	1.3643
108	239.7	0.02782	1.254	164.6	633.6	0.3330	1.1593	705.0	1.2755	766.9	1.3619
110	247.0	0.02790	1.217	167.0	633.7	0.3372	1.1566	705.5	1.2731	767.6	1 3596
112	254.5	0.02799	1.180	169.4	633.8	0.3413	1.1538	706.1	1.2708	768.3	1.3573
114	262.2	0.02808	1.145	171.8	633.9	0.3453	1.1510	706.6	1.2684	769.1	1.3550
116	270.1	0.02817	1.112	174.2	634.0	0.3495	1.1483	707.2	1.2661	769.8	1.3527
118	278.2	0.02827	1.079	176.6	634.0	0.3535	1.1455	707.7	1.2636	770.5	1.3503
120	286.4	0.02836	1.047	179.0	634.0	0.3576	1.1427	708.2	1.2612	771.3	1.3479
122	294.8	0.02846	1.017	181.4	634.0	0.3618	1.1400	708.6	1.2587	772.0	1.3455
124	303.4	0.02855	0.987	183.9	634.0	0.3659	1.1372	709.1	1.2563	772.8	1.3431
126	312.2	0.02865	0.958	186.3	633.9	0.3700	1.1344	709.6	1.2538	773.5	1.3407
128	321.2	0.02875	0.931	188.8	633.9	0.3741	1.1316	710.0	1.2513	774.2	1.3383

NOTE: Sp. En. = Specific Enthalpy.

TABLE 5. PROPERTIES OF CARBON DIOXIDE

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Specific Enthalpy and Entropy Taken From −40 F							
				Specific Enthalpy		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Sp. En.	Entropy	Sp. En.	Entropy
0	305.5	0.01570	0.29040	18.8	138.7	0.0418	0.3024	153.7	0.3342	167.5	0.3612
2	315.9	0.01579	0.28030	19.8	138.8	0.0440	0.3014	153.7	0.3330	167.6	0.3600
4	326.5	0.01588	0.27070	20.8	138.8	0.0461	0.3005	153.7	0.3318	167.7	0.3588
5	332.0	0.01592	0.26610	21.3	138.8	0.0472	0.3000	153.7	0.3312	167.7	0.3582
6	337.4	0.01596	0.26140	21.8	138.7	0.0483	0.2994	153.7	0.3306	167.8	0.3576
8	348.7	0.01605	0.25260	22.9	138.7	0.0504	0.2982	153.7	0.3293	167.9	0.3563
10	360.2	0.01614	0.24370	24.0	138.7	0.0526	0.2970	153.7	0.3281	168.0	0.3550
12	371.9	0.01623	0.23540	25.0	138.6	0.0548	0.2958	153.7	0.3270	168.1	0.3538
14	383.9	0.01632	0.22740	26.1	138.6	0.0571	0.2946	153.7	0.3259	168.2	0.3526
16	396.2	0.01642	0.21970	27.2	138.5	0.0593	0.2933	153.7	0.3249	168.3	0.3513
18	408.9	0.01652	0.21210	28.3	138.4	0.0616	0.2921	153.7	0.3238	168.5	0.3501
20	421.8	0.01663	0.20490	29.4	138.3	0.0638	0.2909	153.7	0.3227	168.6	0.3489
22	434.0	0.01673	0.19790	30.5	138.2	0.0662	0.2897	153.7	0.3214	168.7	0.3479
24	448.4	0.01684	0.19120	31.7	138.1	0.0686	0.2885	153.7	0.3202	168.8	0.3470
26	462.2	0.01695	0.18460	32.9	138.0	0.0710	0.2873	153.7	0.3189	168.9	0.3460
28	476.3	0.01707	0.17830	34.1	137.9	0.0734	0.2861	153.7	0.3177	169.0	0.3451
30	490.8	0.01719	0.17220	35.4	137.8	0.0758	0.2849	153.7	0.3164	169.1	0.3441
32	505.5	0.01731	0.16630	36.7	137.7	0.0781	0.2834	153.7	0.3158	169.2	0.3431
34	522.6	0.01744	0.16030	37.9	137.4	0.0804	0.2820	153.7	0.3151	169.3	0.3421
36	536.0	0.01759	0.15500	39.1	137.2	0.0828	0.2805	153.7	0.3145	169.4	0.3411
38	551.7	0.01773	0.14960	40.4	136.9	0.0851	0.2791	153.7	0.3138	169.5	0.3401
39	559.7	0.01780	0.14700	41.0	136.8	0.0862	0.2783	153.7	0.3135	169.5	0.3396
40	567.8	0.01787	0.14440	41.7	136.7	0.0874	0.2776	153.7	0.3132	169.6	0.3391
41	576.0	0.01794	0.14185	42.3	136.5	0.0887	0.2768	153.7	0.3127	169.6	0.3386
42	584.3	0.01801	0.13930	42.9	136.3	0.0899	0.2761	153.7	0.3122	169.7	0.3381
44	601.1	0.01817	0.13440	44.3	136.1	0.0924	0.2745	153.7	0.3112	169.8	0.3371
46	618.2	0.01834	0.12970	45.6	135.7	0.0950	0.2730	153.7	0.3101	169.9	0.3362
48	635.7	0.01851	0.12500	47.0	135.4	0.0975	0.2714	153.7	0.3091	170.0	0.3352
50	653.6	0.01868	0.12050	48.4	135.0	0.1000	0.2699	153.7	0.3081	170.1	0.3342
52	671.9	0.01887	0.11610	49.8	134.5	0.1027	0.2681	153.7	0.3069	170.2	0.3333
54	690.6	0.01906	0.11170	51.2	133.9	0.1054	0.2663	153.7	0.3057	170.3	0.3324
56	709.5	0.01927	0.10750	52.6	133.4	0.1081	0.2644	153.7	0.3046	170.5	0.3315
58	728.8	0.01948	0.10340	54.0	132.7	0.1108	0.2626	153.7	0.3034	170.6	0.3306
60	748.6	0.01970	0.09940	55.5	132.1	0.1135	0.2608	153.7	0.3022	170.7	0.3297
62	769.0	0.01995	0.09545	57.0	131.3	0.1164	0.2584	153.7	0.3012	170.8	0.3289
64	789.4	0.02020	0.09180	58.6	130.6	0.1194	0.2560	153.7	0.3002	170.9	0.3281
66	810.3	0.02048	0.08800	60.2	129.7	0.1223	0.2535	153.7	0.2991	171.0	0.3273
68	831.6	0.02079	0.08422	61.9	128.7	0.1253	0.2511	153.7	0.2981	171.1	0.3265
70	853.4	0.02112	0.08040	63.7	127.5	0.1282	0.2487	153.7	0.2971	171.2	0.3257
72	875.8	0.02152	0.07654	65.5	126.0	0.1321	0.2450	153.7	0.2962	171.3	0.3250
74	898.2	0.02192	0.07269	67.3	124.5	0.1360	0.2414	153.7	0.2953	171.4	0.3242
76	921.3	0.02242	0.06875	69.4	122.8	0.1398	0.2377	153.7	0.2945	171.5	0.3235
78	944.8	0.02300	0.06473	71.6	120.9	0.1437	0.2341	153.7	0.2936	171.6	0.3227
80	968.7	0.02370	0.06064	73.9	118.7	0.1476	0.2304	153.7	0.2927	171.7	0.3220
82	993.0	0.02456	0.05648	76.4	116.6	0.1578	0.2195	153.7	0.2920	173.8	0.3215
84	1017.7	0.02553	0.05223	79.4	113.9	0.1679	0.2087	153.7	0.2914	176.0	0.3209
86	1043.0	0.02686	0.04789	83.3	110.4	0.1781	0.1978	153.7	0.2907	178.2	0.3204
87.8	1069.9	0.03454	0.03454	97.0	97.0	0.1880	0.1880	153.7	0.2901	180.1	0.3199

NOTE: Sp. En. = Specific Enthalpy.

sub-cooled refrigerant can be taken as equal to the values read from the tables for a saturated liquid at the same temperature. Thus if F-12 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 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 which is present in vapor form.

TABLE 6. PROPERTIES OF MONOFLUOROTRICHLOROMETHANE (F-11)

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Specific Enthalpy and Entropy Taken From −40 F							
				Specific Enthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Sp. En.	Entropy	Sp. En.	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	98.2	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

NOTE: Sp. En. = Specific Enthalpy.

TABLE 7. PROPERTIES OF WATER

Sat. Temp. F	Abs. Press. Lb per Sq In.	Volume		Specific Enthalpy and Entropy Taken From +32 F							
				Specific Enthalpy		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Sp. En.	Entropy	Sp. En.	Entropy
32	0.0887	0.01602	3296.0	0.00	1073.0	0.0000	2.1826	1096.9	2.2277	1120.8	2.2688
35	0.1000	0.01602	2941.0	3.02	1074.4	0.0062	2.1724	1098.3	2.2172	1122.2	2.2581
40	0.1217	0.01602	2441.0	8.05	1076.8	0.0163	2.1555	1100.6	2.2000	1124.5	2.2406
45	0.1475	0.01602	2034.0	13.07	1079.2	0.0262	2.1390	1102.9	2.1832	1126.7	2.2234
50	0.1780	0.01602	1702.0	18.08	1081.5	0.0361	2.1230	1105.2	2.1667	1129.0	2.2066
55	0.2140	0.01603	1430.0	23.08	1083.9	0.0459	2.1073	1107.5	2.1506	1131.3	2.1902
60	0.2561	0.01603	1206.0	28.08	1086.2	0.0556	2.0920	1109.8	2.1349	1133.5	2.1742
65	0.3054	0.01604	1021.0	33.08	1088.6	0.0652	2.0771	1112.2	2.1196	1135.8	2.1585
70	0.3628	0.01605	868.0	38.07	1090.9	0.0746	2.0625	1114.5	2.1046	1138.1	2.1432
75	0.4295	0.01606	740.0	43.06	1093.2	0.0840	2.0483	1116.7	2.0900	1140.3	2.1283
80	0.507	0.01607	632.9	48.05	1095.5	0.0933	2.0344	1119.0	2.0758	1142.5	2.1138
85	0.596	0.01609	543.3	53.04	1097.8	0.1025	2.0208	1121.2	2.0619	1144.7	2.0996
90	0.698	0.01610	467.9	58.03	1100.0	0.1116	2.0075	1123.4	2.0483	1146.8	2.0857
95	0.815	0.01612	404.2	63.01	1102.3	0.1206	1.9946	1125.6	2.0350	1148.9	2.0721
100	0.949	0.01613	350.3	68.00	1104.6	0.1296	1.9819	1127.9	2.0220	1151.1	2.0588
105	1.101	0.01615	304.4	72.98	1106.8	0.1384	1.9695	1130.2	2.0093	1153.2	2.0458

NOTE: Sp. En. = Specific Enthalpy.

Consider, for example, F-12 with a quality (the per cent in vapor form) of 30 per cent; the specific enthalpy of this material would be equal to:

$$h_m = h_l + 0.30\,(h_v - h_l) \qquad (2)$$

where

h_m = specific enthalpy of the mixture.
h_l = specific enthalpy of the liquid.
h_v = specific enthalpy of the saturated vapor.

Values of h_l and h_v are obtained from Table 1 for the actual pressure of the mixture.

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 completly 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_{ls}, and can therefore be read from the table.

Thus,

$$h_{ls} = h_m = h_{vd} - (1 - x)\,(h_{vd} - h_{ld}) \qquad (3)$$

or,

$$x = (h_m - h_{ld}) \div (h_{vd} - h_{ld}) \qquad (4)$$

where

h_{ls} = specific enthalpy of saturated liquid at entrance to expansion valve.
h_m = specific enthalpy of mixture.
h_{vd} = specific enthalpy of saturated vapor at discharge.
h_{ld} = specific enthalpy of liquid at discharge.
x = proportion of liquid in the mixture.

The refrigerant *cycle* is the series of state changes which occur in the conditioning processes needed 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 costs of an expanding engine so greatly exceed the advantage resulting from the work realized that such engines are not used and 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.

Simple Refrigeration Cycles

The most common and least complicated type of refrigeration cycle is shown in Fig. 1 and is called the *simple saturation cycle*. 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 irreversibly 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 desuperheating 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

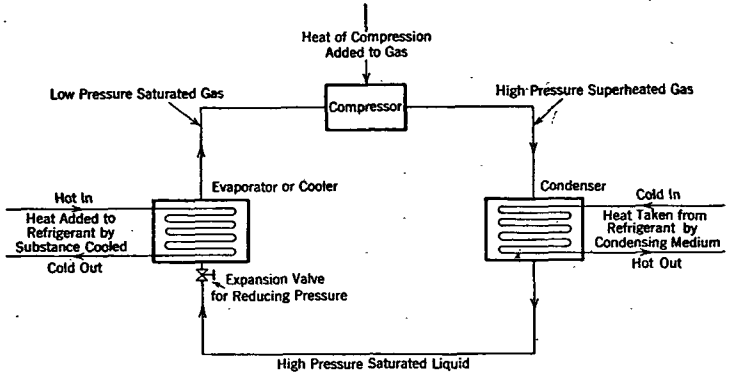


FIG. 1. MECHANICAL REFRIGERATION SYSTEM

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 pressures of 52.7 psia and 121 psia with F-12 as the refrigerant. Determine: (*a*) the cooling effect provided by each pound of refrigerant, (*b*) the refrigerant circulating rate, (*c*) the horse power required, (*d*) the quantity of heat to be dissipated from the condenser, (*e*) the required condenser cooling water, in gallons per minute, if 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.

Solution. (*a*) Saturated liquid F-12 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 pick-up per pound of refrigerant or,

$$W_r = (7 \text{ ton} \times 200) \div 53.14 = 26.3 \text{ lb per minute.}$$

(*c*) The horse power 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,

$$(\text{hp}) = W_r (h_d - h_{vs}) \div 42.42 \qquad (5)$$

where

hp = horse power.
W_r = refrigerant circulating rate in pounds per minute.
h_d = specific enthalpy of vapor at condition of discharge from compressor.
h_{vs} = specific enthalpy of saturated vapor entering compressor.

W_r is known from (*b*) and h_{vs} is the enthalpy of refrigerant as it enters the compressor in a saturated vapor state at 52.7 psia; thus $h_{vs} = 82.82$.

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 state 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 into Equation 5,

$$(\text{hp}) = 26.3 (89.34 - 82.82) \div 42.42 = 4.03$$

(*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 $26.3 \times 59.66 = 1569$ Btu per minute. This same figure can, of course, be determined more directly by subtraction of the enthalpy 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 per cent 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,

$$(\pi d^3) \div 4 = 39.4 \qquad d = 3.69 \text{ in.}$$

Coefficient of Performance

In order to permit evaluation of the effectiveness with which any given cycle operates, some term is desirable which would be comparable to the *efficiency* that is used for heat engines. In refrigeration the desired effect is heat extraction and the cost of achieving this extraction is the amount of energy which 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, thus,

$$(\text{cop}) = (h_{vs} - h_{lc}) \div (h_d - h_{vs}) \qquad (6)$$

where

cop = coefficient of performance.
h_{vs} = specific enthalpy of saturated vapor entering compressor.
h_{lc} = specific enthalpy of liquid at discharge from condenser.
h_d = specific enthalpy of vapor at discharge from compressor.

The subscripts vs, d, and lc represent state points at suction and discharge of the compressor and at discharge from the condenser. Thus for the conditions of the simple saturation cycle which was used in Example 1.

$$(\text{cop}) = (82.82 - 29.68) \div (89.34 - 82.82) = 8.17.$$

This coefficient can be compared with that which would exist if the system were to operate on an ideal Carnot cycle for which the coefficient of performance would be,

$$(\text{cop}) = \frac{T_s}{T_c - T_s}$$

where

T_s = evaporator temperature, Fahrenheit degrees, absolute.
T_c = condenser temperature, Fahrenheit degrees, absolute.

In problem 1, $T_s = 501$ F (which is 41 F + 460) and $T_c = 554$ F (which is 94 F + 460) and,

$$(\text{cop}) = \frac{501}{(554 - 501)} = 9.6$$

The actual cycle is therefore 8.17 ÷ 9.6 or 85 per cent 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 performance.

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.

Influence of Discharge Pressure

In contrast to the 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 which 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, the enthalpy of saturated liquid leaving the condenser increased with pressure so 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.

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.

Exact measurement of the heat carried away in the jacket cooling water requires facilities which frequently are not available on field installations, but a reasonable close approximation to both the heat loss and the work requirement can be obtained from theory, providing the temperature of the gas leaving the compressor is experimentally determined. Knowing the temperature and pressure at both suction and discharge, the actual state points can be readily determined from the tables of refrigerant properties and the entropies and enthalpies thereby evaluated. Then the energy dissipation to cooling water can be calculated approximately from the equation,

$$Q_j = W_r\,(\Delta s)\,T_{avg} \qquad (7)$$

where

Q_j = energy dissipated to cooling water, Btu per minute.
W_r = weight of refrigerant, pounds per minute.
Δs = entropy change between suction and discharge.
T_{avg} = average temperature of gas passing through compressor, Fahrenheit degrees, absolute.

The work of compression is then given by the equation:

$$(\text{hp}) = [W_r (h_d - h_{vs}) + Q_j] \div 42.42 \qquad (8)$$

where

hp = horsepower.
h_d = specific enthalpy of vapor at compressor discharge.
h_{vs} = specific enthalpy of saturated vapor entering compressor.

From the form of Equation 8 it would appear that an increase in the energy loss to the jacket would result in a greater power requirement since the Q_j term is additive; actually, however, computation will readily show that the effect of an increase in the Q_j term is to *decrease* the horse power since the rate of decrease of enthalpy difference will, in every case, exceed the increase in heat loss. Thus 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 since 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 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 simple saturation than when operating with maximum sub-cooling. What is even more surprizing is that the condenser pressure is lower for the sub-cooling cycle than for the saturation cycle; this condition results from the fact that, for the same load, the refrigerant flow rate is less when there is sub-cooling.

Because of the advantages attendant upon the use of sub-cooling, many methods are in use for obtaining some sub-cooling 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. In this case subcooling is realized at the expense of superheating the suction vapor and thermodynamic analysis will show at once that a definite loss will accrue. The only conditions under which such a system can be justified are those for which the vapor would, in any event, pick up a comparable degree of superheat while en route to the compressor; in this event the deliberate superheating, with useful subcooling as a result, is preferable to the useless superheating which would otherwise occur. In small compressors having large surface area in relation to volume, a definite gain is usually accomplished by such interchange, because for such units there would be considerable heating in any event. On very large compressors the loss in volumetric efficiency would more than offset any gain, and the capacity would be reduced. Somewhere between these two extremes, the exact point depending on the type of compressor, location of valves, etc., the two factors would offset one another.

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. In such a case the *expansion* valve becomes a simple pressure-reducing valve since there is no vapor formation—hence no expansion—during passage of the liquid refrigerant through it. This system is theoretically identical, from the standpoint of thermodynamic effectiveness, with the simple saturation cycle.

Clearance and Conventional Volumetric Efficiency

Clearance, like displacement, is a characteristic—usually fixed—of a given compressor. In some cases clearance pockets are provided which place within the operator's control the ability to alter the clearance of the machine, but most moderate size compressors are built with fixed clearance. By definition the clearance is the percentage of the volume swept by the piston which is represented by spaces in the end of the cylinder (including valve spaces, etc.) when the piston is at the end of its stroke.

Because of the trapping of high pressure vapor in the clearance space, and its subsequent re-expansion, the suction valves of the compressor do not open until the piston has completed part of its stroke. Hence the volume of fresh vapor introduced into the compressor per stroke is less than the volume swept by the piston. The ratio of actual volume of fresh gas to swept volume is, by definition, the conventional volumetric efficiency, CVE. In equation form,

$$(\text{CVE}) = 100 - V_c \left[\frac{v_s}{v_d} - 1 \right] \qquad (9)$$

where

CVE = conventional volumetric efficiency.
V_c = clearance, per cent of volume swept by piston, which is contained in spaces at end of cylinder when piston is at end of stroke (clearance includes valve spaces, etc.)
v_s = specific volume of gas at compressor inlet.
v_d = specific volume of gas at compressor discharge.

Values of v_s and v_d can be obtained directly or by calculation from the tables of properties of refrigerants.

Influence of Wiredrawing and of Piping Losses

The pressure drop which occurs during passage of the refrigerant through suction and discharge valves of the compressor is known as wiredrawing. Its effect is equivalent to that of a reduction in the evaporator pressure (but without the disadvantage of reduced refrigerating effect) and an increase in condenser pressure, hence is undesirable. In analyzing a cycle, wiredrawing can be taken into account since it amounts to the interposition of a small *expansion valve* between the state point of vapor leaving the evaporator and that of vapor at start of the compression process; a similar equivalent valve can be considered as interposed at the end of the compression process and before the state point corresponding to the condition of the superheated vapor in the discharge line.

In exactly the same way representation on the ideal cycle can be given to the pressure losses that occur in the connecting piping. In each case the entire loss of a given line can be treated as though it occurred during passage through an equivalent expansion (or pressure reducing) valve located at any convenient place in the line. Data to permit evaluation of line losses are given in a subsequent section of this chapter.

Complex Refrigeration Cycles

The preceding sections have dealt only with refrigeration systems in which there is but one evaporator. When two or more evaporators are required, and the pressures differ in each, much greater opportunity is afforded the engineer for obtaining large economies through selection of one of the more complex cycles. Consider, for example, the refrigeration requirements of an air conditioning system which is to cool a very large volume of 90 F outside air down to a conditioned temperature of 40 F. A simple saturation system, operating with evaporator temperature less than 40 F, would accomplish the desired purpose, but at the expense of excess power requirements since 50 per cent of the sensible cooling load could instead be handled by an evaporator operating at a temperature somewhat less than 65 F. The more effective procedure in this case would be to place two direct expansion coils in series, the first operating at a temperature less than 65 F and the second, at a temperature below 40 F. In this way approximately one half of the total load would be picked up at the higher evaporator pressure and therefore need be raised through a much smaller thermal height. The theoretical advantage of such operation can be visualized from the increase in Carnot effectiveness; assuming that the condenser temperature is 100 F, the coefficient of performance (cop) of the low temperature evaporator is,

$$(\text{cop}) = (35 + 460) \div (100 - 35) = 7.63$$

while that of the higher temperature evaporator is,

$$(\text{cop}) = (60 + 460) \div (100 - 60) = 12.97.$$

Since the load is assumed to be equally distributed between the two evaporators the cop of the series system is the arithmetical average of the values for the two evaporators or, $(7.63 + 12.97) \div 2 = 10.30$. Thus use of the series cycle would afford a theoretical saving in power of approximately one third.

One common fallacy, with respect to complex systems, is the misconception that a high evaporator pressure necessarily means a low power requirement. In many instances operating conditions will require use of series evaporators, but in a cycle for which the vapor leaving the higher pressure evaporator must be throttled to the pressure of the low pressure evaporator before entering the compressor, there obviously is no advantage resulting from the higher operating pressure of the one evaporator since the refrigerant which it handles must be compressed through the same *lift* as though both evaporator pressures were the same. Consideration of this case brings out the fact that the effectiveness of a complete cycle depends upon the possibility of operating the system with suction vapor at different pressures. This can be accomplished through use of more than one compressor, or by means of special individual compressor arrangements which permit use of different suction pressures in the opposite ends of a double-acting machine, or introduction of vapor at two different pressures into the cylinder of a dual-effect compressor.

Compound Compression Cycles

In large systems the compression process can be carried out in steps as the refrigerant passes through a number of cylinder ends arranged for operation in series. Thermodynamically the advantage of 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 over-all compression process to more closely approach the ideal condition of isothermal compression. Essentially, such intercoolers—whether of the water or the flash refrigerant type—serve the same purpose as a cooling jacket, but with greater effectiveness because of the more satisfactory heat transfer conditions.

Multiple Expansion Valves

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. Unfortunately, however, there is no effective means of extracting vapor continuously during the expansion and re-compressing it. Thus in the simple cycle the flash vapor must necessarily be allowed to drop to evaporator pressure.

When a compound compression cycle is used there is at least one intermediate pressure at which flash vapor can be extracted. In such cases all refrigerant from the condenser can be dropped through a first expansion valve to the higher suction pressure and the flash vapor then extracted and returned to the compressor. Some of the resultant liquid refrigerant then passes through the high-pressure evaporator while the remainder proceeds through a second expansion valve in which its pressure is dropped to the valve corresponding to the low-pressure evaporator.

Pipe Sizes and Friction Losses

The effect on performance of pressure losses in the piping of a refrigerating system has already been discussed. In all cases frictional losses should be kept to a minimum and piping should be selected which will give the smallest loss consistent with over-all economy of the system. Actual losses vary, of course, with the physical characteristics of the particular refrigerants, but, by way of example, data will be given for one of the refrigerants, dichlorodifluoromethane (F-12) which finds wide use in air conditioning applications.

Tables 8, 9, and 10 give the pressure loss per 100 ft of piping (including an *average* number of fittings) for oil-free dichlorodifluoromethane; tabular values should be increased if oil is flowing with the refrigerant. For copper pipe the tables are for type L tubing and are based on the outside diameter.

Common practice fixes suction line velocities between 1,500 and 3,000 fpm while 2,000 to 3,500 fpm is the accepted range for discharged lines. Velocities higher than those indicated will result in noisy operation while lower velocity in the suction line may result in the loss of entrained oil. Refrigeration for air conditioning usually involves wide variation in load,

TABLE 8. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE DISCHARGE OR HOT GAS LINES[a]

CAPACITY BTU PER HOUR	PRESSURE DROP IN POUNDS PER SQUARE INCH PER 100 FT[b] — LINE SIZES, INCHES									
	⅝	¾	⅞	1⅛	1⅜	1⅝	2⅛	2⅝	3⅛	3⅝
10,000	2.3	1.0	0.6							
15,000	4.9	2.0	1.0							
20,000	8.5	3.4	1.7	0.6						
25,000		5.3	2.6	0.9						
30,000		7.5	3.6	1.2	0.5					
40,000			6.4	2.1	0.7					
50,000			9.8	3.1	1.0	0.5				
60,000				4.4	1.3	0.7				
70,000				6.0	1.9	0.9				
80,000				8.0	2.5	1.1				
90,000				10.2	3.1	1.4				
100,000					3.8	1.7	0.5			
125,000					6.0	2.6	0.7			
150,000					8.5	3.8	1.0			
175,000					11.6	5.1	1.3			
200,000						6.7	1.7	0.6		
250,000						10.4	2.6	0.9		
300,000							3.7	1.2	0.5	
400,000							6.7	2.2	0.9	
500,000							10.5	3.5	1.5	0.7
600,000								5.0	2.1	1.0
800,000								9.0	3.8	1.8
1,000,000									5.8	2.9
1,250,000									9.5	4.4
1,500,000										6.4
2,000,000										11.3

[a]Soft annealed copper tubing up to and including ¾ in. outside diameter. Hard copper pipe ⅞ in. outside diameter and larger.

[b]Length of tubing includes the average number of fittings.

requiring double suction risers, one large and one small. The large riser should be trapped to insure oil return while the small riser should be sized for a velocity of 3,000 fpm at minimum load; and the large riser then sized to handle the balance of the load at a velocity not less than 2,000 fpm. The total pressure loss in the suction piping, including valve losses, is usually between 2 and 3 psi while hot gas line losses are held to approximately 4 psi and liquid line losses are not greater than 5 psi.

As protection against re-condensation on the cylinder head of the compressor, a surge drum is sometimes installed in the discharge line close to the compressor; this protection is especially desirable on systems using evaporative condensers where sun effect during shut-down may be serious.

The Steam Jet System

The steam jet system under certain circumstances is desirable for use in air conditioning[1]. 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.

TABLE 9. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE LIQUID REFRIGERANT LINES

CAPACITY BTU PER HOUR	PRESSURE DROP IN POUNDS PER SQUARE INCH PER 100 FT[a] — PIPE SIZES, INCHES			
	⅞	1⅛	1⅜	1⅝
100,000	0.6			
125,000	0.9			
150,000	1.3			
175,000	1.8			
200,000	2.3	0.6		
225,000	2.9	0.8		
250,000	3.6	1.0		
275,000	4.3	1.2		
300,000	5.1	1.4		
325,000	5.9	1.6		
350,000	6.9	1.8		
375,000	7.9	2.1		
400,000	9.0	2.3	0.8	
450,000		2.9	1.0	
500,000		3.5	1.3	
550,000		4.3	1.5	0.7
600,000		5.0	1.8	0.8
700,000		6.7	2.4	1.1
800,000		8.7	3.1	1.4
900,000			3.9	1.7
1,000,000			4.7	2.1
1,200,000			6.7	3.0
1,400,000			9.0	4.0
1,600,000				5.1
1,800,000				6.3
2,000,000				7.9
2,200,000				9.2

[a]Length of tubing includes the average number of fittings.

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. 2. 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 hour per 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 and the vapor and air mix with the impel-

TABLE 10. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES

Copper Pipe Actual O.D. Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature F Deg						
		−10	0	10	20	30	40	50
⅜	2,000	0.3	0.3	0.2	0.2	0.2	0.1	0.1
	4,000	1.3	1.0	0.8	0.7	0.6	0.5	0.4
	6,000	2.8	2.2	1.8	1.5	1.2	1.0	0.9
	8,000	4.8	3.8	3.1	2.6	2.1	1.8	1.5
	10,000	7.4	5.8	4.8	3.9	3.3	2.8	2.3
	12,000	10.5	8.4	6.8	5.6	4.7	4.0	3.3
	14,000	14.0	11.0	9.1	7.6	6.4	5.4	4.5
	16,000		14.5	12.0	9.8	8.3	7.0	5.8
	18,000			15.0	12.3	10.4	8.7	7.2
	20,000				15.0	12.7	10.7	8.9
1⅛	7,000	0.4	0.3	0.3	0.2	0.2	0.2	0.1
	10,000	1.0	0.7	0.5	0.5	0.4	0.3	0.3
	15,000	1.9	1.5	1.2	1.0	0.8	0.7	0.6
	20,000	3.3	2.6	2.1	1.7	1.4	1.2	1.0
	25,000	5.0	4.0	3.2	2.7	2.2	1.9	1.6
	35,000	9.7	7.7	6.2	5.1	4.3	3.6	3.0
	45,000	15.8	12.6	10.0	8.4	7.0	5.9	4.9
	60,000				14.8	12.2	10.2	8.6
	70,000						14.0	11.7
1⅜	10,000	0.3	0.2	0.2	0.2	0.1	0.1	0.1
	15,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	20,000	1.2	0.9	0.7	0.6	0.5	0.4	0.4
	30,000	2.6	2.1	1.6	1.3	1.1	0.9	0.8
	40,000	4.6	3.6	2.8	2.3	1.9	1.6	1.4
	50,000	7.0	5.5	4.4	3.5	2.9	2.5	2.1
	60,000	10.0	7.8	6.2	5.0	4.2	3.5	3.0
	80,000		14.0	11.0	8.7	7.3	6.2	5.2
	100,000				13.5	11.3	9.5	8.2
1⅝	30,000	1.6	1.3	1.0	0.8	0.7	0.6	0.5
	40,000	2.7	2.1	1.7	1.4	1.1	0.9	0.8
	50,000	4.2	3.2	2.5	2.1	1.7	1.4	1.2
	60,000	6.1	4.5	3.6	2.9	2.4	2.0	1.7
	70,000	8.7	6.3	4.8	3.8	3.1	2.6	2.2
	80,000		8.4	6.3	4.9	4.0	3.3	2.8
	90,000			8.0	6.2	4.9	4.1	3.5
	100,000			10.0	7.6	6.1	5.0	4.2
	120,000					8.6	7.0	5.9
	140,000						9.5	7.9
2⅛	50,000	0.7	0.5	0.4	0.3	0.3	0.2	0.2
	100,000	2.6	1.8	1.4	1.1	0.9	0.8	0.7
	150,000	5.6	3.9	3.0	2.4	2.0	1.6	1.4
	200,000	9.8	6.7	5.2	4.1	3.4	2.8	2.4
	250,000	14.8	10.3	8.0	6.3	5.1	4.2	3.6
	300,000		14.5	11.3	9.0	7.2	6.0	5.0
	350,000		19.5	15.3	12.0	9.7	7.8	6.7
	400,000			19.6	15.3	12.5	10.0	8.5

[a]Length of tubing includes the average number of fittings.

ling steam on the discharge side of the jet. 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 which 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 lb per square inch and condenser water temperatures as high as

TABLE 10. PRESSURE LOSSES IN DICHLORODIFLUOROMETHANE SUCTION REFRIGERANT LINES (CONCLUDED)

Copper Pipe Actual O.D. Inches	Capacity Btu per Hour	Pressure Drop in Pounds per Square Inch per 100 Ft[a]						
		Refrigerant Temperature Deg F						
		−10	0	10	20	30	40	50
2⅝	50,000	0.2	0.2	0.1	0.1	0.1	0.1	0.1
	100,000	0.7	0.6	0.5	0.4	0.3	0.2	0.2
	150,000	1.6	1.2	1.0	0.8	0.6	0.5	0.4
	200,000	2.8	2.1	1.7	1.4	1.1	0.9	0.7
	250,000	4.3	3.4	2.6	2.1	1.7	1.3	1.1
	300,000	6.1	4.5	3.7	3.0	2.4	1.9	1.5
	350,000	8.2	6.0	5.0	4.0	3.2	2.5	2.0
	400,000		7.8	6.5	5.1	4.2	3.3	2.7
	450,000			7.7	6.4	5.3	4.0	3.5
	500,000				7.8	6.4	5.0	4.2
	550,000					7.7	6.2	5.1
	600,000						7.4	6.2
3⅛	200,000	1.2	1.0	0.8	0.6	0.5	0.4	0.4
	300,000	2.6	2.0	1.6	1.3	1.0	0.8	0.7
	400,000	4.5	3.4	2.6	2.1	1.7	1.4	1.3
	500,000	7.3	5.4	4.1	3.3	2.7	2.2	1.9
	600,000		8.1	6.0	4.7	3.8	3.1	2.7
	700,000			8.4	6.5	5.2	4.2	3.5
	800,000				8.6	6.8	5.5	4.6
	900,000					8.7	7.0	5.9
	1,000,000						8.9	7.3
3⅝	300,000	1.2	0.9	0.7	0.6	0.5	0.4	0.3
	400,000	2.0	1.6	1.3	1.0	0.8	0.7	0.6
	500,000	3.2	2.5	1.9	1.6	1.3	1.0	0.9
	600,000	4.6	3.6	2.8	2.2	1.8	1.5	1.3
	700,000	6.4	4.9	3.8	3.0	2.5	2.0	1.7
	800,000	8.7	6.4	4.9	3.9	3.2	2.5	2.2
	900,000		8.2	6.2	4.9	3.9	3.2	2.7
	1,000,000			7.7	6.1	4.9	4.0	3.3
	1,100,000			9.4	7.3	5.8	4.8	4.0
	1,200,000				8.7	6.9	5.6	4.8
	1,300,000					8.0	6.6	5.6
	1,400,000					9.3	7.6	6.4
4⅛	400,000	1.0	0.8	0.6	0.4	0.4	0.3	0.3
	600,000	2.4	1.8	1.4	1.1	0.9	0.7	0.6
	800,000	4.1	3.1	2.4	2.0	1.6	1.3	1.1
	1,000,000	6.6	4.8	3.7	3.0	2.5	2.0	1.6
	1,200,000	10.0	7.1	5.4	4.4	3.5	2.9	2.4
	1,400,000		10.0	7.5	5.9	4.8	3.9	3.3
	1,600,000			10.0	7.7	6.2	5.1	4.2
	1,800,000				10.0	7.9	6.4	5.3
	2,000,000					9.7	7.9	6.6
	2,200,000						9.5	7.9

[a]Length of tubing includes the average number of fittings.

90 F. The 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 lb per square inch results in an increase in steam consumption of approximately 5 per cent, whereas a further decrease in booster steam pressure to 10 lb per square inch increases the steam consumption by approximately 72 per cent over that required at 200 lb per square inch.

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 per cent 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 fundamental rule governing the absorption (in a closed system) of a gas by a liquid is Raoult's Law, which states that at any given temperature the ratio of the partial pressure of a volatile component in a solution to the vapor pressure of the pure component at the same tem-

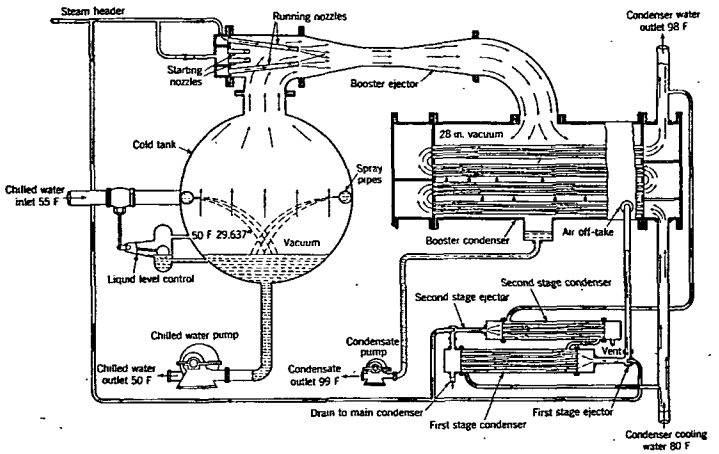


FIG. 2. DIAGRAMMATIC ARRANGEMENT OF STEAM JET VACUUM COOLING UNIT

perature is equal to its mol fraction in the solution. The mol fraction in turn is equal to the number of mols of substance divided by the total number of mols present. The number of mols in a given weight of a compound is equal to the weight divided by the molecular weight.

This law applies strictly only to what is known as an ideal solution, one in which the inter-molecular forces between the substances present in the solution are equal. Actually, no such solutions exist, so that deviations from Raoult's Law are always found in practice. The deviation is called positive when the observed pressure is greater than that calculated from Raoult's Law, while the term negative deviation refers to the opposite case. Negative deviations are found wherever chemical attraction exists between the solvent and the solute. Positive deviation occurs when there is a difference in the internal pressure of the components, chemical attraction between them being absent.

In order to make an effective absorption machine, large negative deviations from Raoult's Law must be shown by solutions of the refrigerant in the liquid absorbent, because the larger the negative deviation, the greater is the amount of refrigerant that can be cycled, using a given weight of absorbent. Cycling a large amount of refrigerant for a given

weight of absorbent is important because of the heat required to raise the temperature of the mixture and disassociate the refrigerant and the absorbent. Only the latent heat of the refrigerant can be recovered for useful work.

Many refrigerant-absorbent combinations have been proposed and quite a number have been tested. A diagrammatic representation of a typical closed absorption system is outlined in Fig. 3. In this system a mixture of refrigerant and absorbent is evaporated in the generator, passes to an analyzer and rectifier where it is purified, and then to a condenser where the refrigerant and remaining absorbent are condensed. It then passes through an expansion valve to an evaporator, where heat is absorbed from a cooling load. From the evaporator the vapor and resi-

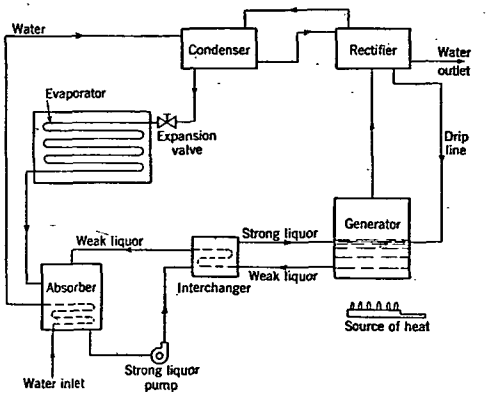


FIG. 3. CLOSED ABSORPTION SYSTEM

dual absorbent passes to an absorber where it meets absorbent which is initially low (weak) in refrigerant concentration. The absorbent absorbs the vapor and the strong absorbent liquor is transferred to the generator through an interchanger with the weak liquor returning from the generator.

A cooling medium, ordinarily water, is used in the absorber to remove the heat of absorption and maintain the absorptive power of the absorber at a maximum.

Like the steam jet system, the absorption system compares most favorably when a cheap source of cooling water and steam or other heat is available. Unlike the steam jet system, the comparative performance is usually best with a wide range of temperature between the evaporator and absorber, since with a good refrigerant-absorbent combination, the amount of heat and water required for a given refrigerating effect increases slowly with an increase of evaporator-condenser temperature range.

At the present time the most used refrigerant-absorbent combinations are: (1) water and ammonia, and (2) dichloromonofluoromethane and dimethyl ether of tetraethylene glycol. With the latter combination the boiling points of the refrigerant and absorbent are sufficiently wide apart that almost pure refrigerant is obtained without the use of a rectifier.

Ice Systems

Cold water systems using ice as the cooling agent have been installed in many theaters, restaurants, funeral homes, churches and other places where short hours of operation and high peaks of cooling demand make this type of system desirable. A comparatively small quantity of ice in the water cooling tank of such a system can release refrigeration at a relatively rapid rate. For instance, neighborhood theaters having a peak demand of 1,200,000 Btu per hour (100 tons refrigeration) have found 8 ton capacity ice bunkers satisfactory.

In operation, the water in the air conditioning system is circulated over ice placed in an insulated box and is cooled to the 38 or 40 F range or higher if desired. This cold water is pumped from the ice bunker to air cooling coils or spray type air washers. The blowers, coils, air washer or air handling sections are the same as those parts in any system employing cold water as a refrigerant.

The ice water cooler or ice bunker is usually built at the installation in

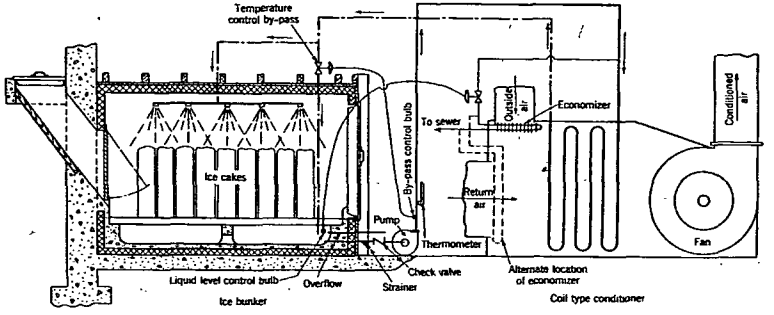


FIG. 4. TYPICAL ICE SYSTEM SHOWING BUNKER DETAILS

a location where it can easily be iced. It can be constructed of any desired material such as concrete, steel, or wood with an adequate amount of insulation to save the ice from one period of use to the next. The basic requirement is that the tank be durable and water tight. A typical bunker with connections to a coil type air conditioning system is shown in Fig. 4. About 60 cu ft of gross bunker volume are allowed per ton of ice capacity.

The shape of the bunker usually conforms to the available space. The one illustrated has overhead sprays, but if head-room is lacking the ice is placed on the floor of the bunker with the water returned around the lower part of the blocks from a perforated distribution pipe run along one side of the bunker. To secure good circulation the supply water is extracted from a similar perforated pipe on the opposite side of the bunker.

The temperature of the water is controlled at a predetermined point by a thermostat in the supply line. If the temperature drops too low, a part of the return water is by-passed directly to the sump and is not cooled over the ice. In the larger systems it is customary to install an overflow control which, as the ice melts, discards the excess water through an economizer coil. The surface of the economizer is large in relation to

the flow so that the water is warmed to 60 F or more as it is discharged from the system.

Storage Systems

In an attempt to lower initial equipment cost and operating expense, or increase the refrigeration capacity of an existing air conditioning system, storage refrigeration has been utilized in a few applications. Some of the methods which have been adopted include the storage of refrigeration in the form of chilled water, chilled brine, ice on evaporator coils[2] and the accumulation of thin sheets of ice on copper plates in a steel tank[3]. If the peak load factor is low as compared with a long period of operation, such as in a restaurant, or if the hours of operation are short but the usage factor high as in a church, then it is possible to consider storage refrigeration. This method of accumulating refrigeration frequently makes it possible to use low cost off-peak electric power. Power costs may also be reduced by installing a smaller refrigeration plant, augmented by a storage system, and by operating it for longer periods.

The Reverse Cycle

The reverse cycle—frequently referred to as a *heat pump*—is identical in theory with the ordinary refrigeration cycle and differs only in the sense that the desired effect is a heat source rather than a heat sink. Since a condenser delivers more heat than is picked up by the evaporator (it discharges the heat equivalent of work supplied during compression) it follows that the effectiveness of a heat pump, its coefficient of performance, is greater than that of a refrigeration cycle operating between the same temperature limits.

In heating by the reverse refrigeration cycle energy is absorbed in an evaporator from some available source of heat, pumped to a higher temperature and delivered to a condenser[4]. The heat from the condenser is used for heating purposes. The compressor acts as a heat pump whose fundamental function is to raise the potential of the heat. The theoretical ratio of the heat delivered to the work of compression, the coefficient of performance, is given in Equation 10.

$$(\text{cop}) = \frac{T_c}{T_c - T_s} \qquad (10)$$

where

cop = coefficient of performance.
T_s = absolute temperature of evaporator.
T_c = absolute temperature of condenser.

Thus, with a small spread of temperature between the evaporator and the condenser, 6 or 8 times as much heat may be obtained theoretically, and 3 to 5 times practically, as the work introduced. There are a number of limitations, however, the most serious of which is the lack of ready availability of a practical source of heat.

1. Well water is the most desirable since its temperature is higher than other sources even in the winter, and thus a large amount of heat may be removed in relation to the weight of water handled.

2. Air may be used but its specific heat is low and its temperature uncertain. When the most heat is needed, the temperature of the air is lowest, thus resulting in the least favorable temperature combination.

3. It has been proposed to obtain heat by freezing water but this is still in the experimental stage.

Some of the other factors which act as limitations are: the large temperature spread when using air as a source of heat and when attempting to cool with even moderately low outside temperatures, the frequent disparity between the size of the cooling load and heating load requiring extra equipment for a complete heating load, and the relatively high initial cost of equipment as compared to that at present available for heating by conventional means.

Because of these limitations, the present application of the system is largely limited to temperate climates, such as Florida and Southern California, or to heating only for intermediate seasons, or to other localities which have peculiar advantages as, for instance, the ready availability

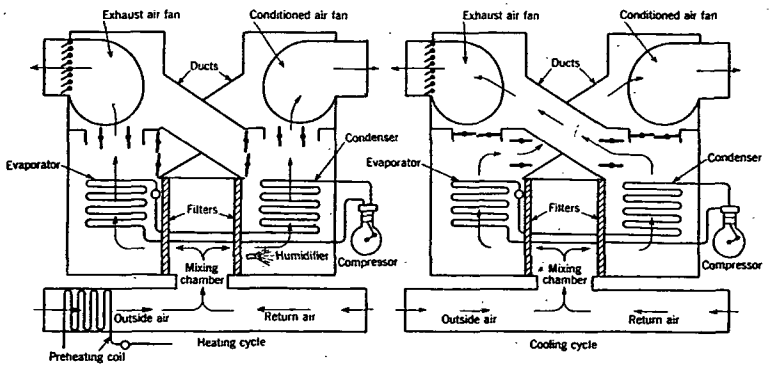


FIG. 5. SCHEMATIC OPERATION OF REVERSED CYCLE CONDITIONING SYSTEM

of well water. In these locations it is frequently possible to do all of the heating necessary with the refrigeration equipment so that the extra cost is only that of reversing the functions of the condenser and evaporator.

There are a number of reversed systems now in operation, particularly among utility companies, using well water as the source of heat. These systems range in size up to 320 hp. In the case of the largest system in operation at present, the cost of the electrical energy would have to be approximately 0.7 cents per kilowatthour in order to compete with oil at 6 cents per gallon.

A typical arrangement of a reversed cycle conditioning system where air is used as a source of heat is shown in Fig. 5. If the air seldom drops below freezing, heat is often required in the morning and cooling during the afternoon in order to maintain comfortable conditions in such a system. The arrangement as shown lends itself to automatically changing over as required.

REFRIGERATION EQUIPMENT AND ARRANGEMENTS

Types of Compressors

There are many different types of compressors, using various refrigerants. Each type has its advantages for its particular application,

and those generally used for air conditioning are of the following types:

1. Reciprocating compressors (commonly referred to as piston type).
2. Centrifugal compressors.

Reciprocating compressors are available in a wide range of sizes and types. Any of a number of refrigerants, including dichlorodifluoromethane (F-12), methyl chloride, ammonia, carbon dioxide, and sulphur dioxide may be used in reciprocating machines. The first of these is used extensively in direct expansion systems of comfort air conditioning.

Compressors may be classified into two general types, (*a*) open type, (*b*) enclosed type. If the driving mechanism is external to the compressor, then the shaft must be brought out through the crankcase and a shaft seal or stuffing box must be used to prevent escape of the refrigerant. This type of compressor is known as an open-type compressor. When the driving mechanism is located within the crankcase of the compressor in such a way as to avoid the necessity of a shaft seal, the compressor is known as the completely enclosed or hermetically sealed type.

Open-type compressors may be further classified as belt driven and directly connected. A great number of direct-driven units are now being used which generally operate at higher rotational speeds than the belt-driven type.

The present tendency is toward forced lubrication of the bearings of compressors by means of an oil pump driven from the crankshaft, although there are many splash lubricated compressors on the market. The chief advantages of the forced lubricated compressor are that the lubrication system requires less energy for its operation than the splash type, the oil can be easily filtered before it enters the bearings, and less oil is usually required.

The compressor capacity must be selected for and matched to the maximum load for the installation on which it is to be used. Air-conditioning loads, however, vary over a wide range, and a wide fluctuation in air conditions may result during periods of light load if on-and-off control of full compressor capacity is used. To prevent such undesirable fluctuation, several methods are employed to vary the capacity of reciprocating compressors, such as:

1. By-passing one or more cylinders, of a multi-cylinder compressor, from discharge to suction.
2. Rendering the suction valves of one or more cylinders of a multi-cylinder compressor inoperative. This is usually accomplished by depressing the suction valves.
3. Varying the speed of the compressor, usually by using variable speed or two-speed electric motors.
4. Using clearance pockets to control the quantity of refrigerant pumped.
5. Restricting the suction inlet to one or more of the cylinders of a multi-cylinder compressor either by an automatic modulating valve or by an *on* and *off* valve.

All of these methods, with the exception of variable speed, result in a slightly lowered over-all compressor efficiency when in use, since the mechanical losses remain constant whereas the quantity of refrigerant pumped is lowered.

Centrifugal compressors are used with very low pressure refrigerants; usually both evaporator and condenser work below atmospheric pressure. Water and monofluorotrichloromethane (F-11) are the refrigerants commonly used in centrifugal machines.

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 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. 6.

The compressor capacity can be varied by controlling the condensing pressure. This is accomplished by regulating the quantity and temperature of the condenser cooling water. The capacity falls off with increasing condensing pressure. Centrifugal compressors are seldom built for less than 50 tons capacity, since it is not practical to make impellers which pump much less than the volume of refrigerant required for this tonnage.

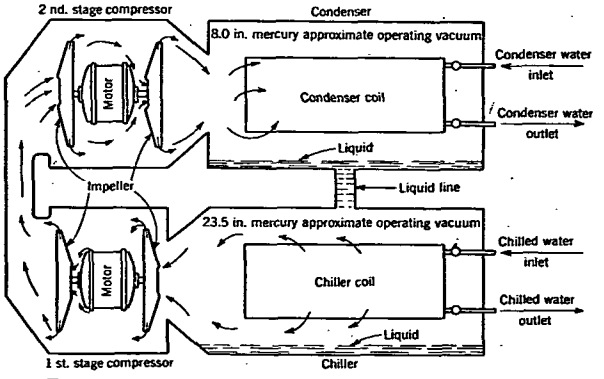


FIG. 6. ENCLOSED TYPE CENTRIFUGAL CONDENSING UNIT

Characteristics of Compression Systems

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. 7 and 8.

From Fig. 7 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. 7 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. 8. It may be noted that the power required by the 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

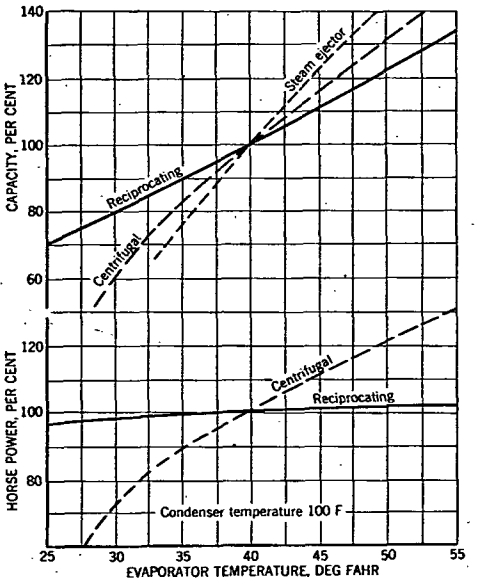


FIG. 7. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

systems are well suited to those applications where condensing water is cheap, or where condensing water is rather high in temperature.

Condensers

Condensers used for liquifying the refrigerant are of three general designs: (1) *air cooled*, (2) *water cooled*, and (3) *evaporative* (combination air and water).

1. *Air cooled condensers* are seldom used for capacities above 3 tons of refrigeration, unless an adequate water supply is extremely difficult to obtain, as, for instance, in railway air conditioning. Even on fractional tonnage installations, air is used as the condensing medium only where water is expensive or where simplicity of installation warrants the higher condensing pressure, and consequent higher power costs than would be obtained using water as the condensing medium.

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 cooled air.

The principal disadvantages of air cooled condensers are the power required to move the air and the reduction of capacity on hot days. This loss of capacity due to high condensing pressures on hot days requires that equipment of increased capacity be selected to meet the peak load. Thus at normal loads the equipment is oversized.

2. *Water cooled condensers* are of the double pipe type, the shell and tube type, or the shell and coil type. Double pipe condensers are arranged so

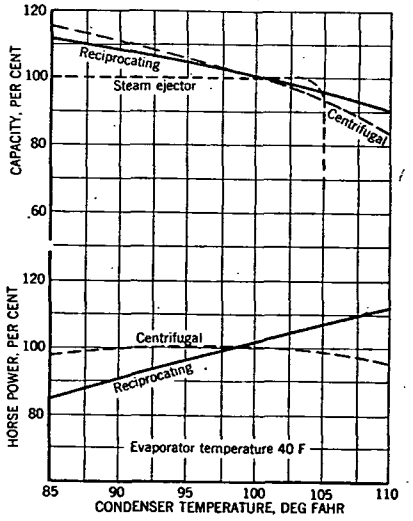


FIG. 8. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

that water passes through the inner of two concentric pipes and refrigerant circulates through the annular space between the pipes. Where possible, there should be counter-flow of the refrigerant and the condensing water to obtain maximum temperature differences. This type is usually used only with small condensing units.

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.

Because there is a decided tendency to conserve the water in city mains and because most large cities are restricting the use of water for air conditioning and refrigeration equipment, it is often necessary to install cooling towers or evaporative condensers. Cooling towers, unfortunately, produce the warmest condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet the maximum load at abnormal condensing water temperatures.

If properly designed, this makes little difference in the efficiency of operation throughout the year except at those times when the condensing water temperature is highest. As this occurs only for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains. Economies are possible when a cooling tower is used, which cannot be achieved by the use of condensing water from city mains. In certain localities, the lowest city water temperature during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of the outdoor temperature. With a cooling tower, the temperature of the condensing water may rise to 80 or 85 F under maximum conditions, but under less than maximum conditions the temperature of the water leaving the cooling tower drops considerably. It has been established that in these localities during 50 per cent of the time, the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60 F, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is small as the only water required is that used to make up the loss by evaporation in the cooling tower itself. Refer to the section on Cooling Towers in Chapter 37.

Shell and coil condensers are in general use for medium sized condensing units, and consist of a coil of tubing mounted inside a shell. The cooling water passes through the coil.

3. *Evaporative condensers.* Due to the high cost of city water for condenser purposes, and due to ordinances in some localities prohibiting the discharge of large quantities of such water into the sewage systems, there has been developed a condenser which uses a minimum amount of water on a finned 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. 9. The fan draws the air over a finned tube condenser which is kept wet by a water spray. The discharge refrigerant gas from the compressor enters the top of the condenser coil and 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 the level is maintained in the sump 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. The air leaving the unit is almost completely saturated, so that care must be taken in locating discharge ducts to prevent condensation.

Evaporative condensers are available in sizes up to 100 tons or more. These units use only a small portion of the water required for a water cooled condenser. The water is vaporized by the heat of the refrigerant so that each pound of water used extracts approximately 1000 Btu from the refrigerant, whereas under standard rating conditions where the water temperature rise is 20 F, each pound of water extracts only 20 Btu from the refrigerant. Including the water lost by entrainment in the discharge air, by overflow and stand-by evaporation, the water used is about 3 to 5 per cent of the amount that would be required for a water cooled 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 combination uses about the same quantity of water, the evaporative condenser has the advantage of lower cost and smaller space requirements.

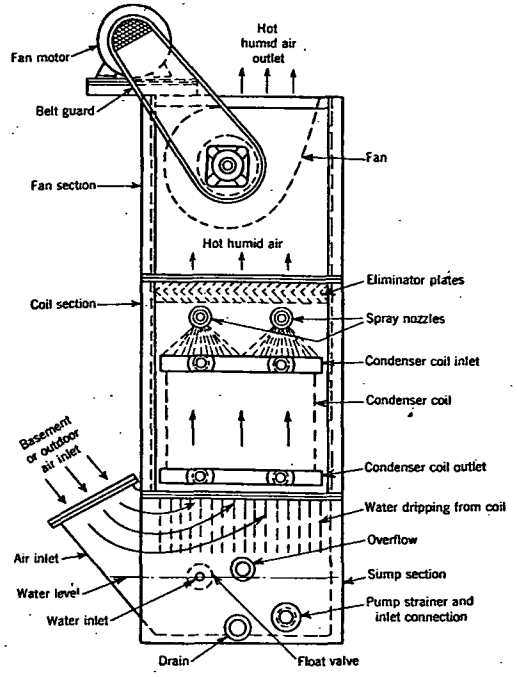


FIG. 9. SCHEMATIC VIEW OF AN EVAPORATIVE CONDENSER

Expansion Valves

The thermostatic expansion valve is a device to regulate the flow of liquid refrigerant so that the evaporator will always be used to best advantage. The evaporator coil must be kept as full as possible without any chance of liquid refrigerant entering the suction line. The expansion valve accomplishes this by regulating the supply of refrigerant, so that the temperature of the gas leaving the evaporator is always slightly higher than the temperature of the boiling refrigerant inside of it. This difference in temperature between the outgoing (suction) gas and the liquid refrigerant in the evaporator is called the superheat of the gas.

The operation of the thermostatic expansion valve can best be explained by means of a diagram, Fig. 10. A small refrigerant charge in the control bulb exerts a pressure through the tube to the upper side of the diaphragm, which tends to open the valve.

The magnitude of this pressure is determined by the temperature of the suction gas leaving the evaporator, as the control bulb is attached to the suction line at this point and is at approximately the same temperature. The suction pressure in the evaporator is transmitted through the equalizer tube and exerts an opposing force on the other side of the diaphragm in the direction to close the valve. This pressure corresponds to the temperature of the boiling refrigerant. The resulting force on the diaphragm is determined by the differential between the temperature of the suction gas and the boiling point of the refrigerant, which is the amount of superheat in the gas. If this temperature differential becomes greater (superheat increases), the resultant force on the diaphragm opens the valve and admits more refrigerant. The reverse is true if the superheat decreases, and the valve partly closes, thus admitting less refrigerant. The spring keeps the valve closed until the resultant force on the diaphragm cor-

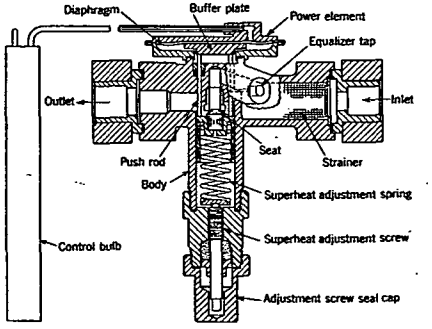


FIG. 10. TYPICAL THERMOSTATIC EXPANSION VALVE

responds to the desired superheat. The adjustment of the spring will change the amount of superheat to be maintained in the suction gas.

The selection of the expansion valve is, of course, determined by the capacity of the valve. The capacity of a valve with a given orifice is determined by the refrigerant used, the differential of pressure across the valve, and the amount the liquid is sub-cooled as it enters the valve. The expansion valves are usually rated at zero sub-cooling of the liquid, or 100 per cent liquid. Frequently special devices are used to properly distribute the refrigerant among the parallel paths of the evaporator. These distributing devices usually have considerable pressure drop. Where they are used, the pressure drop across the expansion valve is not the difference between suction and discharge pressures, as allowance must be made for the pressure drop across the distributing device. An equalizer connection from the evaporator suction line must be made to the underside of the diaphragm (see Fig. 10) whenever the valve outlet is not at the evaporator pressure so as to insure suction pressure at this point. When distributing devices are used, this equalizer connection is essential for proper operation of the valve. Another pressure drop allowance must be made for the liquid line, particularly when the liquid line has an appreciable vertical rise.

Whenever possible, expansion valves should be installed in a manner

which will permit placing thermometer wells in the suction connection and allowing sufficient room for free adjustment of the valve.

Evaporators and Coolers

The types of coolers used in connection with air conditioning work fall into three general groups. The *first*, is the direct cooling of water; the *second*, direct cooling of air; and the *third*, cooling of brine for circulation in a closed system, which can cool either water or air. 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

TABLE 11. BASIS OF EQUIPMENT SELECTION

CAPACITY TONS	MAJORITY USED	SOME USED	FEW USED
0 to 5	Unit systems in conditioned space.	Unit central systems using duct distribution.	Built up central systems.
5 to 25	Built up central systems using reciprocating compressors.	Unit central systems using duct distribution.	Unit systems in conditioned space. Built up systems using absorption and adsorption systems.
25 to 50	Built up central systems using reciprocating compressors.	Built up central systems using centrifugal compressors.	Central systems using adsorption systems.
50 to 400	Built up central systems using reciprocating compressors.	Built up central systems using steam jet and centrifugal compressors.	
400 and Over	Built up central systems using centrifugal compressors.	Built up central systems using steam jet.	

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.

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.

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

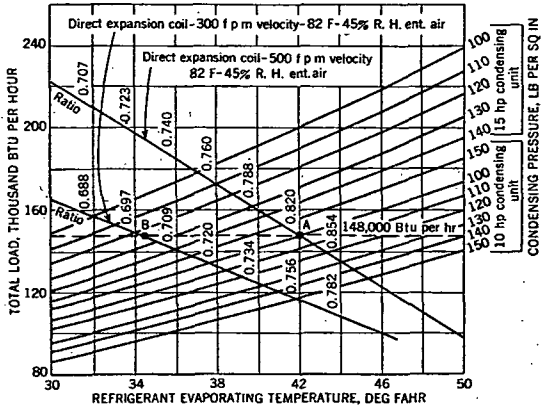


FIG. 11. COMPRESSOR AND COIL PERFORMANCE

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.

Equipment Selection

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 11.

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 36).

The reciprocating compressor in the built-up central system (see Chapter 43) 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.

Centrifugal compressors are used for large installations, and usually where the indirect system is required. The driving mechanism can be

TABLE 12. TYPICAL OPERATING CONDITIONS FOR TWO TYPES OF LOAD

Type of Enclosure	Load, Btu per Hour			Ratio Sensible to Total	Air Entering Coil		Operating Balance Point		
	Sensible	Latent	Total		F Deg	Per Cent R.H.	Evaporator Temp F Deg	Condenser Pressure Lb per Sq In.	Per Cent Sensible Heat
Restaurant	103,000	45,000	148,000	0.695	82	45	34.4	123	69.9
Office	121,000	27,000	148,000	0.820	82	45	42.2	100	82.1

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. 7 that all systems using compressors have a common characteristic and that is, 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. 11. 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 12 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 per cent sensible to total heat, the operating point *A* in Fig. 11 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 per cent 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. 11. 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.

ABBREVIATIONS AND SYMBOLS IN CHAPTER

- cop = coefficient of performance, ratio of refrigerating effect to the heat equivalent of the compressor work.
- CVE = conventional volumetric efficiency.
- d = internal diameter in inches.
- H_t = cooling load in tons.
- h_d = specific enthalpy of vapor at condition of discharge from compressor.
- h_{lc} = specific enthalpy of liquid at discharge from compressor.
- h_{ld} = specific enthalpy of liquid at discharge of expansion valve.
- h_{ls} = specific enthalpy of liquid at entrance to expansion valve.
- h_m = specific enthalpy of mixture.
- h_v = specific enthalpy of saturated vapor.
- h_{vd} = specific enthalpy of saturated vapor at discharge of valve or compressor.
- h_{vs} = specific enthalpy of saturated vapor at state s entering compressor.
- hp = horsepower.
- p_d = pressure of saturated liquid and vapor at discharge of compressor.
- p_s = pressure of saturated liquid.
- psig = pressure pounds per square inch, gage.
- psia = pressure pounds per square inch, absolute.
- Q = quantity of heat, Btu.
- Q_c = heat loss from condenser, Btu per pound refrigerant.
- Q_j = heat dissipated in cooling water, Btu per hour.
- s = entropy.
- Δs = entropy change between suction and discharge.
- T = absolute temperature, Fahrenheit degrees.
- T_{avg} = average temperature, Fahrenheit degrees, absolute of gas passing through compressor.
- T_c = condenser temperature, Fahrenheit degrees, absolute.
- T_s = evaporator temperature, Fahrenheit degrees, absolute.
- t_{sd} = degrees superheat at discharge condition of vapor leaving compressor.
- t_d = discharge temperature, Fahrenheit degrees.
- V_c = clearance, percentage of volume, swept by piston, which is contained in spaces at end of cylinder when piston is at end of stroke (clearance includes valve spaces, etc.)
- v_s = specific volume of gas at suction, cubic feet per pound.
- v_d = specific volume of gas at discharge, cubic feet per pound.
- W_r = refrigerant rate, pounds per minute.
- x = proportion of liquid in mixture of vapor and liquid.

REFERENCES

1—Application and Economy of Steam Jet Refrigeration to Air Conditioning, by A. R. Mumford and A. A. Markson (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 33).

2—The Application of Storage Refrigeration to Air Conditioning, by C. F. Boester (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 675).

3—Use of Cold Accumulators in the Air Conditioning Field, by R. W. Evans and C. J. Otterholm (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 123).

4—Cooling Homes, A Field for Refrigeration, by A. R. Stevenson, presented at the symposium of the Refrigeration with Gas Committee of the *American Gas Association*, April 20, 1926. The Heat Pump, An Economical Method of Producing Low-grade Heat from Electricity, by T. G. N. Haldane (*Electric Review*,

Vol. 105, p. 1161-1162, December 27, 1929, and *I. E. E. Journal*, Vol. 68, p. 666-675, June, 1930). Edison Building Heated and Cooled by Electricity, by H. L. Doolittle (*Power*, Vol. 74, p. 384, September 8, 1931). House Heating by Pump with 5 to 1 Pick-up Ratio, by Gilbert Wilkes and R. E. Marbury (*Electrical World*, Vol. 100, p. 828, December 17, 1932). An All Electric Heating, Cooling and Air Conditioning System, by Philip Sporn and D. W. McLenegan (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 307). Using the Reversed Cycle Refrigerating Principle for a Self-Contained Heating and Cooling Unit, by Henry L. Galson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1935, p. 497). Heating by Reversed Refrigeration, by A. J. Lawless (*Heating, Piping and Air Conditioning*, August, p. 473, September, p. 519, 1940).

BIBLIOGRAPHY

Refrigerating Data Book, Vol. 1 (*American Society of Refrigerating Engineers*).

Refrigeration Engineering, by H. J. Macintire (John Wiley & Sons).

Theory of Mechanical Refrigeration, by N. R. Sparks (McGraw-Hill Book Co., Inc.).

Refrigeration and Air Conditioning Engineering, by B. F. Raber and F. W. Hutchinson (John Wiley & Sons).

Refrigeration, by J. A. Moyer and R. U. Fittz (McGraw-Hill Book Co., Inc.).

CHAPTER 40

Air Distribution

Standards for Satisfactory Conditions, Definitions, Mechanics of Air Distribution, Types of Supply and Return Openings, Outlet Locations, Return and Exhaust Intakes, Specific Applications, Balancing the System

CORRECT air distribution contributes as much or more to the success of a forced air heating, ventilating, cooling or air conditioning system as does any other single factor. An air conditioning system may deliver the required quantity of conditioned air and still fail to give satisfactory room conditions because of poor air distribution. The scope of the chapter is limited to the air distribution within the conditioned space. Reference is made to the distributing duct system only insofar as it affects the performance of the air distribution outlet. See Chapter 41 for information on air duct design.

STANDARDS FOR SATISFACTORY CONDITIONS

Air distribution within a space to be conditioned consists in producing within the space the proper effective temperature whether for cooling, heating or ventilating. This means that a reasonable uniformity of temperature, humidity and velocity must be obtained within the zone of occupancy. This zone is usually the room space between the floor and a plane 6 ft above the floor. Air must have a proper combination of quantity, temperature, humidity and velocity to satisfy the over-all demands, whether for cooling, heating or ventilating. Any or all of these factors may need to be controlled to produce proper comfort conditions. Also the quantities delivered to various locations must be proportioned, so that the sections of the area having the highest or lowest load requirements are satisfied. Otherwise some areas will be too hot, cold, stuffy or drafty, since the effective temperature will be too high or low in these areas.

Other important considerations are dirt and noise. The use of air cleaning devices is recommended to prevent or reduce streaking and smudging of walls and ceilings. Outlets should not increase the noise above the acceptable level.

Reference should be made to Chapter 12, Physiological Principles, for the accepted standards on room temperature, humidity, air motion and direction. Material in Chapter 42, Sound Control, covers acceptable room noise levels and noise generated by air outlets.

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, or cooled, or humidified, or dehumidified, or ventilated.
2. *Exhaust Opening or Return Intake:* Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified, or ventilated.
3. *Outside Air Opening:* Any opening used as an entry for air from outdoors.
4. *Grille:* A covering for any opening and through which air passes.
5. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.
6. *Multiple Louver Damper:* A damper having a number of adjustable blades.

7. *Single Louver Damper:* A damper having one adjustable blade.
8. *Face:* A grille with provision for attaching a damper.
9. *Register:* A face with a damper attached.
10. *Flange:* The portion (either integral or separate) of a grille, face, or register extending into the duct opening for the purpose of mounting.
11. *Frame:* The portion (either integral or separate) of a grille, face, or register extending around the duct opening for the purpose of mounting.
12. *Margin:* The margin of a grille, face, or register is one-half of the difference between the duct dimension and over-all dimension measured either horizontally or vertically.
13. *Fret:* The member separating the openings of a grille, face, or register.
14. *Free Area:* The total minimum area of the openings in the grille, face, or register through which air can pass.
15. *Core Area:* The total plane area of the portion of a grille, face, or register bounded by a line tangent to the outer edges of the outer openings through which air can pass.
16. *Mean Area:* The total of the core and free areas divided by two.
17. *Duct Area:* The area of a cross-section of the duct based on the inside dimensions at a point where the grille, face or register is mounted.
18. *Percentage Free Area:* The ratio of the free area to the core area expressed in percentage.
19. *Aspect Ratio:* The ratio of length of the core of a grille, face or register to the width.
20. *Throw (Blow):* The distance an air stream travels on leaving the outlet (grille or register) to a position at which air motion reduces to a maximum velocity of 50 fpm.
21. *Envelope:* The outer boundary of an air stream moving at a perceptible velocity.
22. *Drop:* The vertical distance, h, in feet, the lower edge of the air stream drops between the outlet and the end of its throw.
23. *Rise:* The converse of drop.
24. *Induction:* The entrainment of room air by the air discharged from the outlet.
25. *Primary Air:* The air leaving the duct and entering the outlet (Q_1 in cfm).
26. *Secondary Air:* The room air picked up by the primary air through induction (Q_2 in cfm).
27. *Total Air:* The mixture of primary and secondary air (Q_3 in cfm).
28. *Induction Ratio:* The total air divided by the primary air equals r, or Q_3/Q_1.
29. *Outlet Velocity:* The average air velocity emerging from the outlet (V_1 in fpm) measured in the plane of the opening.
30. *Terminal Velocity:* The average air stream velocity at the end of the throw (V_T in fpm).
31. *Horizontal Spread:* The divergence of the air stream in the horizontal plane after it leaves the outlet (Degrees).
32. *Vertical Spread:* The divergence in the vertical plane (Degrees).
33. *Temperature Differential:* Temperature difference between primary and room air ($t_r - t_{as}$).
34. *Vane Ratio:* The ratio of depth of vane to shortest opening width between two adjacent grille bars.
35. *Radius of Diffusion:* The horizontally measured distance served by a ceiling outlet within which air motion is reduced to 50 fpm (maximum).

MECHANICS OF AIR DISTRIBUTION

In the mechanics of air distribution, two major problems are involved: (1) complete mixing of the primary air and secondary air outside of the zone of occupancy in order to reduce the temperature difference and air motion to acceptable limits before the air enters the occupied zone; and (2) counteraction of the natural convection and radiation effects within the room.

Momentum Theory

When air is discharged from an outlet into a free open space, the primary air stream entrains room air as it traverses the space. This

entraining effect increases the cross-sectional area and reduces the velocity of the resulting air stream. Induction takes place with the conservation of linear momentum; this has been confirmed by tests which indicate that the momentum remains almost constant throughout the entire measureable length of the air stream. This relationship may be expressed by the Equation 1:

$$M_1 V_1 + M_2 V_2 = (M_1 + M_2) V_3 \qquad (1)$$

where

M_1 = mass of primary air.
M_2 = mass of secondary air.
V_1 = velocity of primary air.
V_2 = velocity of secondary air (normally, $V_2 = 0$).
V_3 = velocity of the mixture.

Since the secondary air velocity is zero,

$$M_1 V_1 = (M_1 + M_2) V_3$$

or,

$$\frac{V_1}{V_3} = \frac{M_1 + M_2}{M_1} = r \qquad (2)$$

In Equation 2, r is called the induction ratio.

Since in many applications the densities of primary and room air are about equal, air volumes may be substituted for mass in Equation 2.

$$r = \frac{Q_1 + Q_2}{Q_1} \qquad (3)$$

where

Q_1 = volume of primary air, cubic feet.
Q_2 = volume of secondary air, cubic feet.

Throw

Mathematical statements for throw have been evolved from research[1] that have proved the following to be *approximately true* for outlets having aspect ratios below 50.

1. Residual velocity of an unrestricted air stream is directly proportional to the outlet velocity.

2. Residual velocity is directly proportional to the effective diameter of the outlet. Effective diameter is defined as a diameter of a circular area (effective area) which would give the same throw. It is the free area times a constant.

3. Residual velocity is inversely proportional to the distance from the outlet.

From these statements the following equation can be established [1].

$$V_r = \frac{C V_1 D_1}{X} = \frac{K V_1 \sqrt{A_1}}{X} \qquad (4)$$

where

V_r = residual maximum velocity in air stream, *i.e.*, the highest maintained velocity at the given cross section in the room.
V_1 = average velocity across the effective area of the outlet.
X = throw.
C and K = constants of proportionality.
$K = \sqrt{\frac{4}{\pi}}\, C = 1.13\, C$
D_1 = diameter of effective area of outlet.
A_1 = effective area of outlet.

Substituting $\frac{Q_1}{V_1}$ for A_1 in Equation 4

$$X = \frac{K V_1 \sqrt{\frac{Q_1}{V_1}}}{V_r} = \frac{K\sqrt{Q_1 V_1}}{V_r} \quad (5)$$

The throw of an unrestrained jet is therefore proportional to the square root of the supply air quantity and initial velocity, and inversely proportional to residual velocity.

The proportionality factor K is a function of residual velocity and initial velocity.

Another variation of the formula (Equation 4) has been suggested [2] in which K_1 is suitable for all conditions;

$$X = K_1 \frac{V_1^{1.16}}{V_r^{0.72}} \sqrt{A} \quad (6)$$

The average velocity at the effective area can be obtained from the following equation.

$$V_1 = \frac{Q_1}{A_1} \quad (7)$$

where A_1 = effective area.

Jet Pattern From Round or Rectangular Openings in a Large Room

The relation between the shape of the discharge of a jet and the shape of the outlet has long been the subject of research. It has been proved to be incorrect to assume that the jet retains the outlet shape when it discharges into a free open space [1]. Air streams from rectangular outlets having low aspect ratios develop a symmetrical or cone shape appearance within a few diameters from the outlet face. From there on, the jet continues to expand at fairly constant rate. Beyond 20 diameters there is very little difference between round and rectangular jets. The assumption can be made that the apex of the cone is in the same position for any jet having small aspect ratio. For the more usual problems of the conventional room with outlets near the ceiling, there are insufficient experimental data to justify a definite statement on the effect of aspect ratio.

If the round or rectangular opening is divided into a number of orifices having straight sides, the performance of the air stream will be similar to that of a plain opening.

Velocity Across Jets

Results of many tests [1] indicate that the ratio of centerline velocity to average velocity is about 3, irrespective of outlet size, shape or initial velocity. This statement is true for stream cross-sections located beyond 10 diameters from the outlet, and is fairly accurate for distances up to 50 diameters. Experimental data are lacking for distances beyond 50 diameters.

Guide Vanes

The average jet angle (included angle in both planes, see Fig. 1) for an air stream as it emerges from a rectangular outlet of any shape without spreading vanes is about 19 deg, plus or minus 5 deg, depending on the type of approach, type of outlet and velocity. The spread increases slightly with velocity. A vaned outlet discharging air uniformly forward

will result in a spread of about 14 deg. This is equivalent to a spread in any direction of about one foot in every 8 ft of blow.

Vanes should have a depth of one to two times the spacing between the vanes. If the ratio of vane depth to spacing is less than one, effective control by means of the vanes cannot be obtained. Little improvement is obtained by increasing the ratio beyond two. The effect of various types of vanes is as follows.

Straight Vanes

As mentioned previously, the included angle between both planes will be in the neighborhood of 14 deg, for a straight setting of the vanes as shown in Fig. 1.

Diverging Vanes

Such vanes set for an angular spread will have a marked effect on the direction and distance of travel of an air stream. An outlet having vertical vanes set straight forward in the center, with uniformly increasing

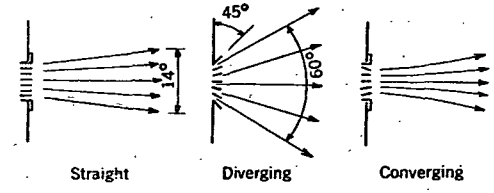


FIG. 1. SPREAD OF AIR STREAM WITH VARIOUS VANES

angular deflection to a maximum at each end of 45 deg, will produce an air stream with a horizontal included angle of approximately 60 deg as shown in Fig. 1. The throw will be reduced one-half for such a vane setting. Increasing the divergence of the vanes reduces the air quantity handled by an outlet for a given duct static pressure. The primary function of the vanes is to spread the air horizontally. Little is gained by spreading the air vertically.

Converging Vanes

The blow of an outlet may be somewhat increased by converging the vanes of an outlet as illustrated in Fig. 1. Even with converging vanes, the resultant angle of spread of an air stream will not be less than 14 deg. The air converges for a few feet in front of the outlet, and then diverges more than if the vanes had been set straight.

Both the horizontal and vertical vanes of an outlet are important. After an installation has been made, many conditions of draftiness or stuffiness can be alleviated by some vane adjustment, provided an independent means for regulation of static pressure behind the vanes is included.

Vertical Drop and Rise

The vertical distance the lower edge of an air stream moves between the outlet and the end of the blow is termed the drop or rise (H). This drop or rise is influenced by the difference in density between the air stream and the room air, resulting from the temperature difference and

the spread of the air stream. For air emerging at room temperature, the drop or rise will be a function of the spread only and will be equal to:

$$H = L \times \tan\left(\frac{\text{Spread Angle}}{2}\right) \quad (8)$$

where

H = drop due to spread, feet.
L = throw, feet.

When there is a temperature difference between the air stream and the room, the additional drop or rise is approximately given by Equation 9:

$$H = \frac{n_1 (t_r - t_{as})\, L\, n_2}{V_1} \quad (9)$$

where

n_1 and n_2 = constants (tentative suggested values $n_1 = 5$, $n_2 = 1.2$).
t_r = room temperature, degrees Fahrenheit.
t_{as} = supply air temperature, degrees Fahrenheit.
V_1 = jet velocity, feet per minute.

For cooling application H is subtracted from the outlet height, for heating H is added.

Effect of Aspect Ratio on Entrainment

In slotted outlets, the air entrainment of the primary jet is a function of aspect ratio [1]. This effect is most pronounced when large changes in the ratio are made. A comparison between a slot of aspect ratio 24 and a square opening of the same area is given in curves *A* and *B* of Fig. 2. At a distance of 8 ft from the outlet, the entrainment of the slot is 8.1 as compared with 6.9 for the square, or an increase of about 17 per cent. Curve *C* shows the further increase in entrainment obtained by using an aspect ratio of 48. An increase of 40 per cent is obtained over the 24 in. x 1 in. slot. This indicates that long narrow slots produce air streams that give high induction of secondary air.

Parallel Slots

The use of several slots in parallel to vary the rate of air entrainment depends mainly on the distance between the slots. If close together, the air pattern is about the same as a single opening of equal area. Spacing the openings farther apart gives an increase in entrainment as shown on curves *C*, *D* and *E* of Fig. 2. It will be noted that 2 openings 24 in. x ¼ in. located very close together will obtain an entrainment which is

TABLE 1. VALUES OF ROOM CIRCULATION FACTOR (F) IN EQUATION 11

Outlet Velocity Fpm V_1	Average Room Velocity Fpm, V_r					Outlet Velocity Fpm V_1	Average Room Velocity Fpm, V_r				
	10	20	30	40	50		10	20	30	40	50
200	6.0	12.0	18.0	24.0	30.0	900	1.3	2.7	4.0	5.3	6.7
300	4.0	8.0	12.0	16.0	20.0	1000	1.2	2.4	3.6	4.8	6.0
400	3.0	6.0	9.0	12.0	15.0	1200	1.0	2.0	3.0	4.0	5.0
500	2.4	4.8	7.2	9.6	12.0	1400	0.9	1.7	2.6	3.4	4.3
600	2.0	4.0	6.0	8.0	10.0	1600	0.8	1.5	2.3	3.0	3.8
700	1.7	3.4	5.1	6.8	8.5	1800	0.7	1.4	2.0	2.7	3.4
800	1.5	3.0	4.5	6.0	7.5	2000	0.6	1.2	1.8	2.4	3.0

about the same as obtained with one 24 in. x ½ in. opening. However, if the slots are spaced 6½ in. apart there is a marked increase in entrainment.

Room Air Motion

The air motion in the occupied zone is usually such that the air travels across the room in reverse direction to the blow of the outlet. The cross-

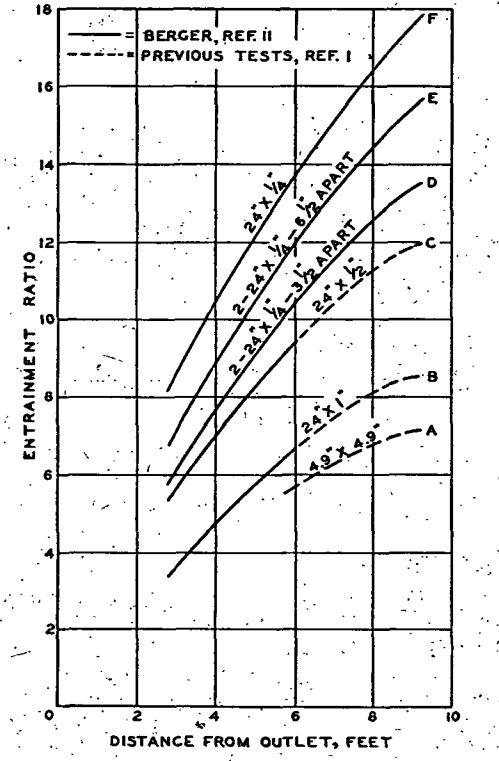


FIG. 2. TYPICAL RELATION OF ENTRAINMENT RATIO TO DISTANCE FROM OUTLET FOR SLOTTED OUTLETS. (BASED ON 800 FPM OUTLET VELOCITY.)

sectional area of this stream is equal to the outlet wall area less the stream area and the area obstructed by furnishings. Equation 10 gives the average room velocity in the occupied zone as a function of the air volume, Q_3, supplied per square foot of outlet wall area and the outlet velocity.

$$V_r = \frac{Q_3}{AZ} \quad (10)$$

where

$Q_3 = Q_1 + Q_2$ = mixture of primary and secondary air, cfm.
V_r = average room velocity, feet per minute.

A = outlet wall area, square feet.

Z = 0.6 (reduction factor to allow for supply air stream 20 per cent and furniture obstruction 20 per cent, at point where supply air stream occupies 20 per cent of the room cross-section).

Since $Q_s = Q_1 r$ by definition, and $r = \frac{V_1}{V_s}$ from Equation 2, and V_s is assumed to be about 200 fpm for total induction in actual practice, then from Equation 10:

$$V_r = \frac{Q_1}{A} \times \frac{V_1}{V_s} \times \frac{1}{Z} = \frac{FV_1}{120}$$

or

$$F = \frac{120 V_r}{V_1} \qquad (11)$$

where F is the room circulation factor expressed in cubic feet per minute per square foot of outlet wall area. Thus room air motion is directly a function of outlet velocity and air volume per square foot of outlet wall area. Hence Equation 11 and Table 1 can be used to determine the probable acceptability of a particular installation from the standpoint of proposed air volume, outlet wall area, and grille velocity.

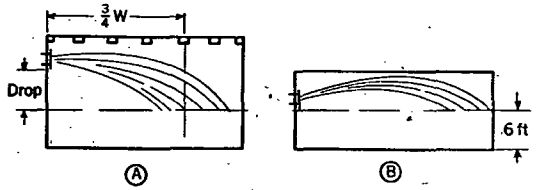


FIG. 3. THROW OF WALL OUTLETS

Outlet Performance

The factors of outlet performance, throw, drop, capacity, noise, dirt and room air motion place considerable limitations on the design of a satisfactory distribution system.

1. *Throw.* The throw of wall or ceiling outlets should be selected to cover three-quarters of the distance toward an exposed wall or window as shown in *A* of Fig. 3. The throw must be sufficient to produce satisfactory conditions over the area to be conditioned. In some ceiling outlets, plaques or similar units, the throw is along the ceiling. The air tends to hug the ceiling and the throw is increased approximately 20 per cent above the throw of an unrestrained air stream. Overblowing is considerably more serious than underblowing, as an overblow will create objectionable down drafts from any surface it strikes; although underblowing in the case of heated air may be serious in that the warm air may rise too rapidly and thus cause stratification in the occupied zone. 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 producing objectionable drafts. If an outlet is adjusted downward to avoid a beam, cold air may enter the zone of occupancy long

before the desired induction has taken place, thus causing serious discomfort.

In ceiling applications, since there is a component downward, the ceiling height and throw are both very important factors. If cold air is used, it must be brought to the proper temperature before entering the zone of occupancy. If perforated plates are used in the ceiling in cooling, the jet velocity must be low. Air slightly above room temperature will usually be properly distributed by outlets selected for cooling. When delivering warm air the stream may be projected downward and the amount of dispersal of the jet varied to get proper mixing and control.

2. *Drop.* The outlets should be located so that the air stream at the termination of the blow is not less than 5 or 6 ft above the floor level. As illustrated in *B* of Fig. 3 the maximum permissible blow 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 ceiling. The air, as it traverses the room, will adhere to the ceiling. The objection to this method is the possible streaking of the ceiling with dirt.

3. *Room Air Motion.* The factors leading to high air motion are excessive velocity, high air volume per square foot of floor area, overblow, striking of beams causing a spilling of the air into the zone of occupancy, and heating in severe climates by means of ceiling outlets which are directed downward.

4. *Dirt.* Although the primary air may be carefully filtered, dirt from the conditioned space may be deposited on the walls or ceiling wherever there is considerable secondary air motion. With ceiling outlets, dirt streaking may be minimized by carefully controlling the discharge of the outlets. With wall outlets, dirt streaking may be minimized by preventing direct impingement of the air on any ceiling or room surface. Floor outlets may offer objection as dirt collectors.

5. *Noise.* The noise of an outlet is primarily the function of the outlet velocity and size, and secondarily of the outlet construction. The maximum acceptable noise level in a space may completely dictate the permissible outlet velocities that may be employed. (See Chapter 42 for discussion of permissible room noise levels and noise generated by outlets.)

6. *Capacity.* The quantity of air to be handled is determined by the heating, cooling, or ventilating requirements. 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 used should be carefully noted to make certain that resulting velocities are suitable for the type of occupancy.

Ceiling outlets will in general handle more air per outlet, without objectionable air motion, than either comparable sidewall or floor outlets.

Temperature Differential

The use of outlets that give rapid mixing permits use of higher supply temperatures for heating and lower supply temperatures for cooling. To facilitate control it is frequently desirable to handle more air at a lesser temperature difference with the room, rather than less air at a greater temperature difference, in order that two areas temporarily loaded differently will not differ too greatly in temperature.

TYPES OF SUPPLY AND RETURN OPENINGS

Wall type openings are: perforated plates, vaned outlets, registers, slotted outlets, and ejector nozzles.

Perforated Outlets

Due to the non-adjustability and small vane ratio perforated sheet metal outlets, although inexpensive, have not met with favor as wall type supply openings. They are useful primarily where directional air control is unnecessary, and for return air openings.

Vaned Outlets

Outlets equipped with both vertical and horizontal adjustable vanes are particularly suited to sidewall distribution. For proper control over the air flow, the vane ratio should be from 1 to 2. Outlets with non-adjustable vanes may be employed but they should only be used where the performance is not critical or can be adequately predicted. Vanes should be properly designed to prevent an increase of noise above permissible level.

Registers

Fixed vanes or perforated grilles equipped with a vane damper are termed registers. They are used primarily for residential heating systems, where the outlet distribution is not critical and low cost is of importance.

Slotted Outlets

Slotted outlets essentially consist of either flat steel plates containing a number of long narrow slots or a single long narrow slot. In order to give a good conversion from static pressure to velocity pressure, the sides of the slots are rounded to give a venturi effect. Due to their high aspect ratio, the slotted outlets have a greater induction effect than the comparable vaned outlets of equal area and consequently the throw is reduced. They are primarily useful where an unobtrusive means of distribution is desired, and where it is desirable to submerge the outlets into the room decoration and to minimize the effect of obstructions in the line of discharge. They are adaptable to narrow rooms of low ceiling. In this case the slots are run full length of the room.

Since these outlets offer greater induction at a given noise level, they are useful in obtaining proper air motion under certain design limitations. One disadvantage, however, is that air quantity and distribution must be carefully planned as correction after installation is difficult.

Ejector Nozzles

Ejector Nozzles refer to outlets which operate at high static pressures, which are constructed to give a high conversion from static in the duct to velocity pressure in the outlet, and which have a high induction effect due to their high outlet velocity. They are chiefly used for long throws and industrial process installations, such as drying, freezing, cooking, etc. Another type of ejector is sometimes referred to as a *louver nozzle* having 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, desirable for use in confined spaces where spot cooling is employed.

Ceiling Outlets

In ceiling outlets when the jet is circumferential and covers 360 deg, greater quantities of room air can be handled than is permissible with comparable side wall outlets, with a given air motion within the occupied space. Temperature differentials may be higher than for ordinary side wall outlets but may not be so high as for outlets with small widely placed slots.

Plaques

When plaques are used, the air from the supply opening impinges against a plate, which, with proper inlet conditions, permits the air to discharge radially. The stream is discharged horizontally in all directions. In certain applications a properly designed plaque yields satisfactory results.

Ceiling Diffusers

This type of outlet is a round, or, in some cases, rectangular outlet, which incorporates guide vanes into its construction.

The air pattern is determined from the contour and design of the guide vanes. In some units the air streams are projected horizontally, the performance being similar to that of well designed plaques. In others the air stream is projected downward. Others produce air streams at varying angles to the ceiling.

Perforated Ceilings

This method obtains air diffusion by discharging air through perforations in the ceiling, the free area being usually less than 15 per cent. The advantages are appearance and the ready application of sound absorbing material to the design. Also, if designed properly, this system provides a low rate of room air motion and consequently lends itself to applications having high load or high ventilating requirements. The perforations should be kept free of accumulations of dirt, as clogging will cause uneven distribution. Dirt will also smudge the ceiling. Best results are obtained in systems having efficient cleaning devices.

In using perforated ceilings for air distribution it is advisable to counteract the wall and glass heat losses by local heating means.

Perforated Panels

Perforated panels are used in some installations, their gross area usually being less than 25 per cent of the ceiling area. A downward jet pattern is obtained without any directional control. The jet velocities are higher than used in perforated ceilings, in order to produce proper diffusion.

Air volume is controlled by controlling the static pressure in the individual plenum chambers that serve the panels.

OUTLET LOCATIONS

In selecting the location of outlets, consideration must be given to the factors of physical construction, physical appearance, location of heating or cooling loads, and outlet performance.

1. The *physical construction* of a building, particularly of old buildings, immediately places limitations on the type of distribution system which

can be employed. Therefore, the first factor in the selection of outlet locations is a consideration of the possible location of the supply duct, that is, whether it is above the ceiling, within the walls, through furred spaces above corridors, or in the conditioned space, etc. A particular method of distribution may be highly desirable but its execution, due to the location of beams and masonry walls, may be an impossibility.

2. The *physical appearance* of the outlets should conform to the esthetic appearance of the room. In factories, warehouses, etc., the esthetic demand may not be high; however, in department stores, clubs, theaters, etc., the location of the grilles may be dictated largely by such demands.

3. The location of *heating or cooling loads* in a room dictate to a great extent the general location of the outlets. The outlets should be located to neutralize any undesirable cold drafts or radiation effects set up by a concentration of the heating or cooling load. The problem can be divided into *natural loads* due to outside weather and *internal heat loads*.

Natural Heating Load

Winter. In winter the primary heating load is caused by exposed walls, windows and skylights. Heat is lost primarily through convection to these exposed surfaces. The convection currents or cold drafts drop down the exposed surfaces and seriously impair the comfort conditions in the room, particularly at the floor level near the exposed surfaces. The outlets should be located to counteract these down drafts. Two methods may be employed:

1. Direct counteraction of convection currents from cold surfaces can be obtained by locating the outlets to blow upward from beneath windows or exposed walls or to blow across the exposed wall. This method is desirable in small offices or bedrooms, or any location where people are seated or working near exposed surfaces. In northern climates, where the outside temperature may be constantly below 40 F, and the construction consists of uninsulated walls and single glass, this method of distribution is particularly useful for the maintenance of comfort requirements.

2. High induction by ceiling or wall outlets may be employed to nullify the convection currents from exposed surfaces. If outside temperatures are consistently below 40 F, and the exposed surfaces are not well insulated, the induction effort required for neutralization of the down drafts is so great that the air motion in the room may exceed comfort limits unless care is taken in selection and location of the outlet. Where comfort conditions are not critical as in factories for heavy manufacturing, warehouses, etc., satisfactory results can be obtained even in cold climates. For uninsulated walls and glass areas some supplementary heating is recommended. Wall diffusers, direct radiation or warm panels will satisfy these requirements for supplementary heating.

3. The location of exhaust or recirculated air openings at the base of large areas of glass is sometimes effective in reducing cold downdraft into the occupied space. In this case, exhaust air quantity and velocity are important considerations in preventing the spill of cold air off the sill.

Internal Heat Load

If a concentrated source of heat is located at the occupancy level of the room, the heating effect may be countered by blowing the supply air toward the heat source or by locating an exhaust or return grille adjacent to the heat source. The latter method will prove more economical,

as heat will be withdrawn at its source rather than be dissipated into the conditioned space. Where a lighting load is particularly heavy (five watts per square foot) and located high in a conditioned space, it may be economically desirable to locate the outlets below the lighting load. Warm air from the lights will stratify near the ceiling and can be removed by an exhaust or return fan, the former being advisable if the wet-bulb temperature of the air is above the outside temperature, and the latter being preferable if the wet-bulb temperature is below that of the outside air. Either method reduces the requirements for supply air. If the lamps are exposed, less saving can be realized than if enclosed, as a considerable portion of the total energy is radiant.

Duct Approaches to Outlets

In order to obtain proper direction of flow and distribution of air from outlets it is necessary that the air stream approaching the outlet be of

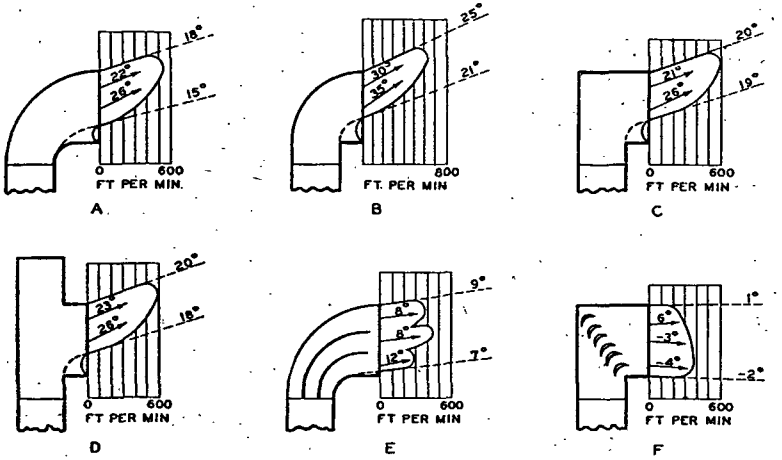


FIG. 4. OUTLET VELOCITY AND AIR DIRECTION DIAGRAMS FOR STACK HEADS WITH EXPANDING OUTLETS

Stack 14 in. x 6 in. Outlets 14 in. x 9 in. Stack Velocity 500 fpm

A. Rounded Throat and Round Back.
B. Square Throat and Round Back.
C. Square Throat and Back.
D. Square Throat and Cushion Chamber.
E. Rounded Throat and Back and 2 Splitters.
F. Square Throat and Back and 6 Guide Vanes.

uniform velocity over the entire duct connection and perpendicular to the face.

Grilles and directional outlets cannot compensate for improper approach. Any attempt to secure a low face velocity and a high duct velocity by constructing an expanding chamber directly behind the grille is likely to be unsuccessful because the enlargement angle in even a straight duct cannot be greater than 7 deg at each side if the stream is to fill the outlet without turbulence.

In elbow outlets or stack heads at the top of vertical stacks it is necessary to provide splitters or guide vanes in the elbows 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.

The direction of flow, distribution and velocity (measured 12 in. from outlet) of the air, based on tests [3], are shown in Fig. 4 for various types of stack heads expanding from a 14 in. x 6 in. stack to 14 in. x 9 in. outlets, without grilles. The air velocity for each was 500 fpm in the stack below the elbow, but the direction of flow and the distribution patterns are generally indicative of performance obtainable with non-expanding elbows of similar shapes for a range of velocities 200 to 1400 fpm. Some of the conclusions drawn from the tests were:

1. Experiments with various elbow outlets on the 14 in. x 6 in. vertical stack [3] with stack air velocities of 200 to 1400 fpm indicated that enlargement of the outlet area, whether used in connection with square or rounded elbows, would not reduce either the angle of discharge (which was 20 to 30 deg above the horizontal) or the outlet velocity. The effect of the enlargement of the outlet was mainly to increase the reverse flow area in the lower part of the outlet, but in each case enlargement of the outlet reduced the static pressure in the duct below the elbow.

2. Splitters in the elbows had the effect of dividing the air stream into a number of streams flowing through rounded elbows and therefore lowered the angle of discharge, reduced or eliminated the reverse flow area, and made the outlet velocity quite uniform.

3. Turning vanes having 2 in. inner and 1 in. outer radii located in the center of the elbow were found most effective in improving performance in regard to angle of discharge, outlet velocity, and elimination of reverse flow area.

4. Pressure loss through stack heads may be reduced by use of splitters or turning vanes or by increasing the inner radius of an elbow. Considering the sum of the velocity and static pressure as a measure of the energy required to change the direction of the air stream and to deliver the air into the atmosphere, and considering the energy required for a plain fitting as 100 per cent, it was found that *turning vanes* dropped the energy requirement of square type stack heads to 45 per cent. Splitters reduced the energy requirement to 90 per cent in long radius elbows and to 74 per cent in short radius turns. In expanding heads splitters reduced the energy requirement to 58 per cent.

Side Outlets in Air Ducts

When air is supplied to a room from side outlets in horizontal ducts it is necessary to use directive devices within the duct at each outlet in order to obtain a uniform velocity of delivered air and to obtain a direction of flow perpendicular to the face of the outlet. In tests [4] conducted with 3 in. x 10 in., 4 in. x 9 in., and 6 in. x 6 in. outlets in a 6 in. x 20 in. horizontal duct at duct velocities of 200 to 1400 fpm (in the 6 in. 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. Figs. 5 and 6 show the flow patterns and outlet velocities for two 3 in. x 10 in. and two 6 in. x 6 in. outlets respectively, at 500 and 1100 fpm duct velocities with outlets equipped with 1 in. and 13 in. extensions. The superiority of the multiple curved deflectors and the beneficial effect of the 13 in. extensions is evident.

Throttling Dampers

In the case where multiple louvers or single blade dampers are used for throttling, considerable deflection of the stream may result. This is particularly true when the fins of the grille core are perpendicular to the damper blades. If the core has sufficient depth and the fins are parallel to the blades, there is a marked tendency to straighten the air stream, although some deflection may still result.

Dampers of special construction, as illustrated in Fig. 7, may be used to maintain a constant direction of blow, approximate distance of blow, and constant outlet velocity regardless of the damper's position. The

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	500–700
Undercutting of doors (through undercut area)	600

capacity of dampers diagrammed as *A* and *B* will be roughly in proportion to the position of the operating lever. They are particularly effective for cooling work with oversized grilles. The single leaf damper shown as *C* in Fig. 7 is objectionable in that it frequently results in a condition whereby two high velocity jets are created along the sides of the duct, or the air spills immediately downward on the occupants below the outlet.

Another important consideration is the increased noise produced by high localized velocities. Obviously, uneven velocities over the outlet will cause the noise level to exceed the values obtained when face velocities are equalized. Also, excessive throttling of dampers to balance systems that are poorly designed will increase the noise level.

RETURN AND EXHAUST INTAKES

The factors that control the selection of return and exhaust intakes are: (1) velocity in occupied zone adjacent to intake, (2) permissible pressure drop through intake, (3) noise, and (4) location.

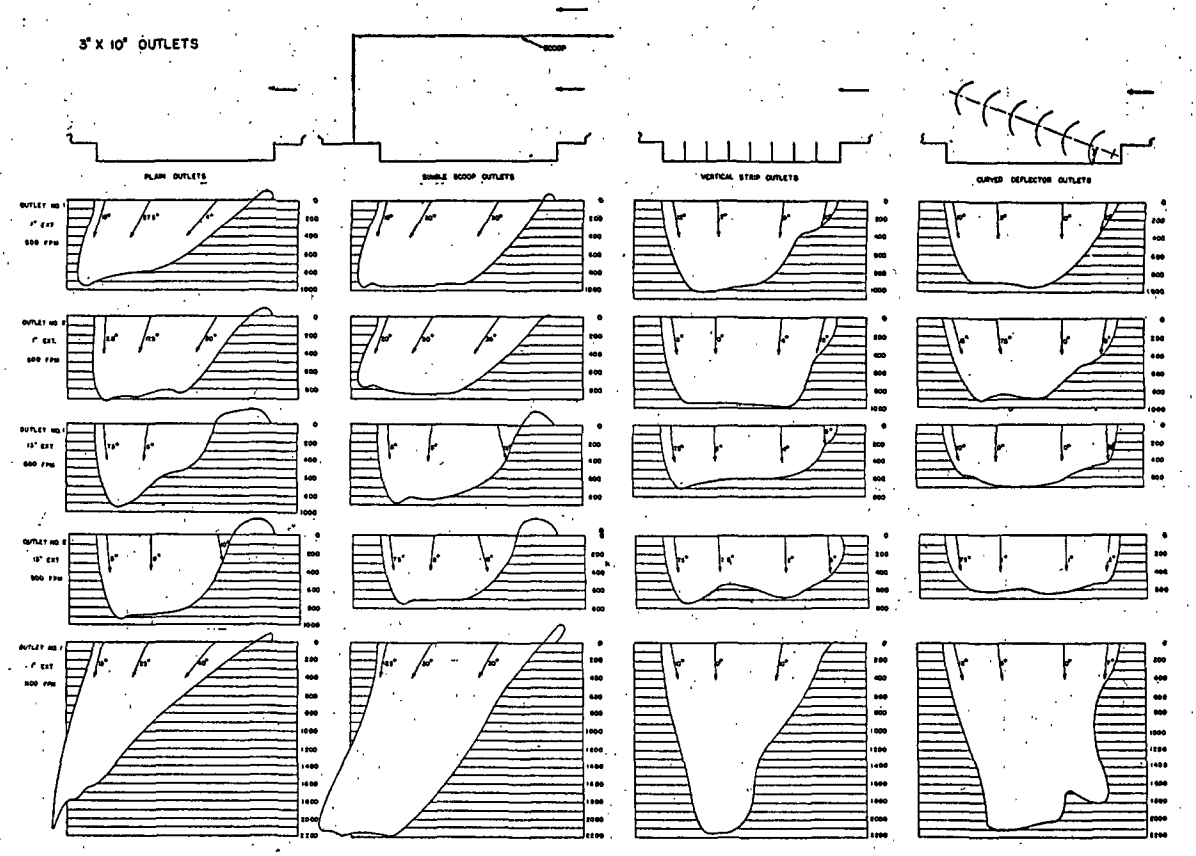
1. Air handled by an exhaust or return intake is drawn from all directions, the velocity dropping off rapidly in every direction. The only locality where drafts may prove objectionable is adjacent to the intake. To prevent excessive air motion in the occupied space due to the return system, it is advisable to compute the total air motion toward the exhaust opening as outlined in Equation 10 where *A* is the exhaust wall area in square feet. Recommended return intake face velocities are given in Table 2.

The withdrawal of air from a space through a return intake is a minor factor in control of the room air motion. The control of the room air motion for the maintenance of comfort conditions depends on the proper selection of the supply outlets. Thus the location of the return intake is not critical, nor the use of an elaborate return system necessary, provided the air motion in the occupied zone adjacent to the intake does not exceed comfort limits. A single return intake or a few large intakes will prove satisfactory provided no local high velocity zones are created.

TABLE 3. APPROXIMATE PRESSURE DROPS FOR LATTICE RETURN INTAKES

Inches Water Gage—Standard Air

PER CENT 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



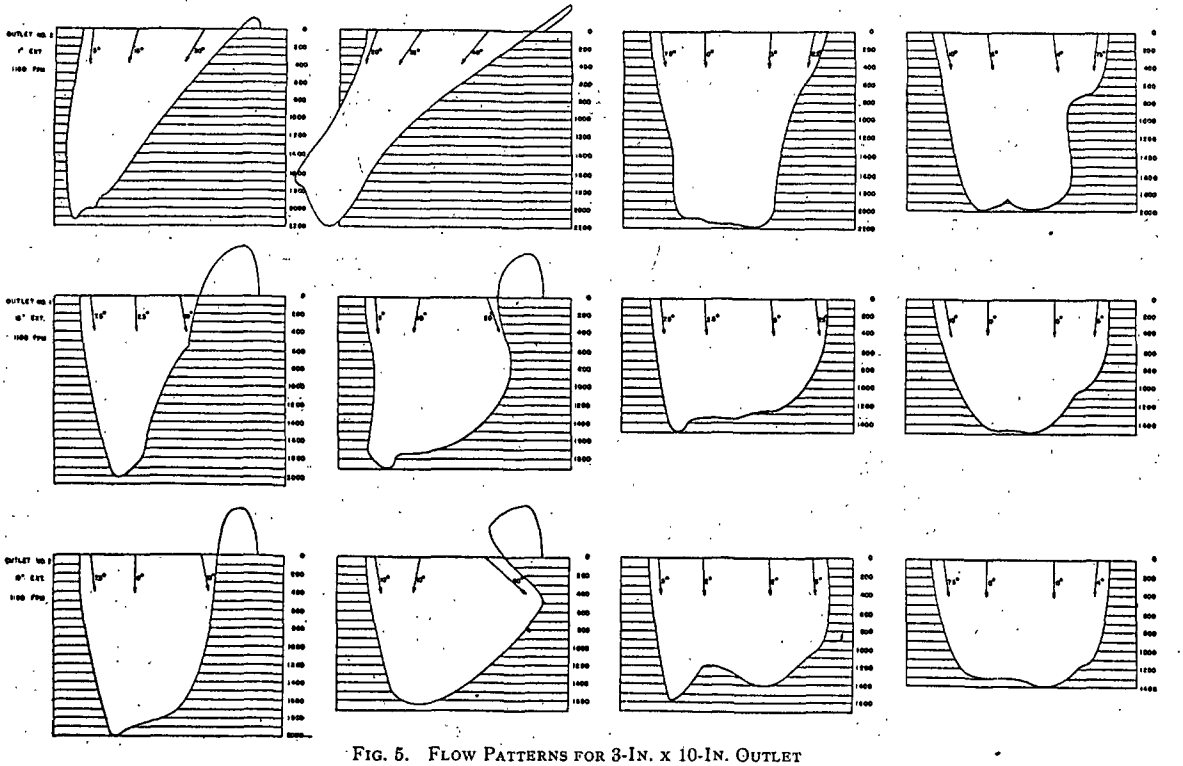
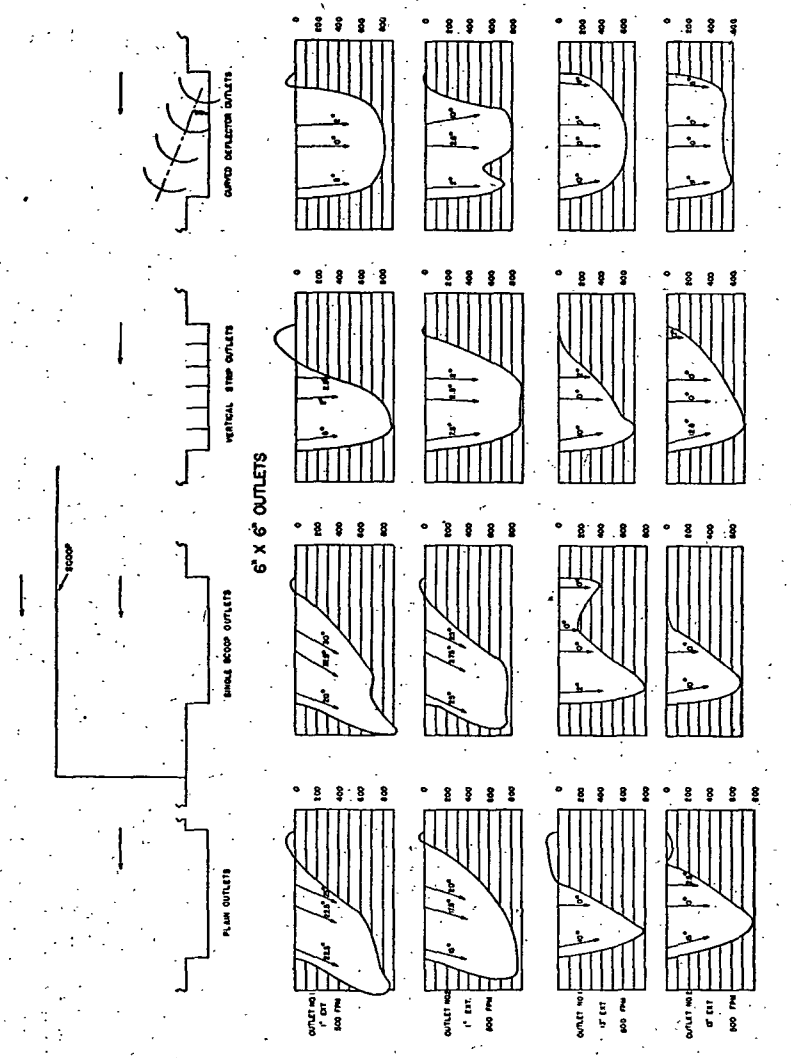


FIG. 5. FLOW PATTERNS FOR 3-IN. X 10-IN. OUTLET



2. 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.

3. The problem of *noise* generated by return intakes is the same as that generated by 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 in their frequent instal-

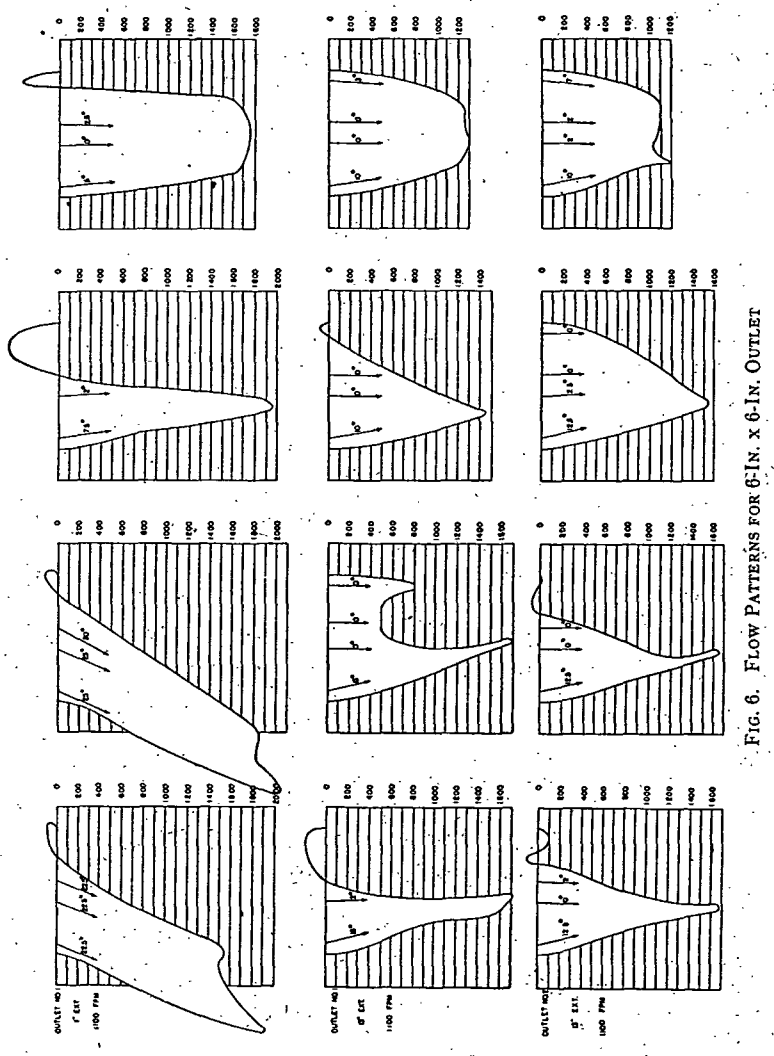


FIG. 6. FLOW PATTERNS FOR 6-IN. x 6-IN. OUTLET

lation at the ear level. When located at ear level, it is recommended that the return intake velocity be not in excess of 75 per cent of the maximum permissible outlet velocity.

4. *Ceiling locations* for outlets are recommended for bars, kitchens, lavatories, dining rooms, club rooms, etc., where warm air will gravitate to the ceiling level. Ceiling returns are less desirable in spaces with severe winter exposure, and where stratification of cold air may take place at the floor level. During the heating season the air will tend to short

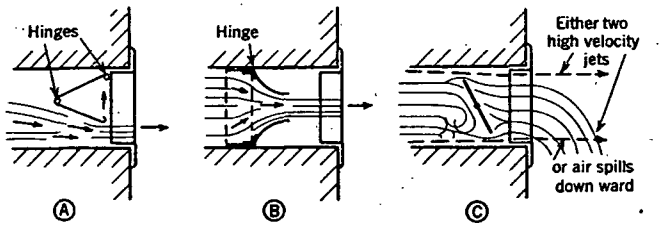


FIG. 7. EFFECT OF VARIOUS DAMPER ARRANGEMENTS DESIGNED FOR STRAIGHT BLOW

circuit between the supply and the ceiling exhaust or return intakes if supply velocities are low.

Some circular ceiling outlets combine the supply and return openings in a single unit. The return duct is in the center with the supply pattern on the outside. This method gives best results for cooling applications. 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 extreme cases, stratification of warm air will cause short circuiting. Where the wall losses are a small part of the total, little difficulty is encountered with stratification.

Floor locations 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, 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. In many cases, particularly in buildings with double glass or hollow glass block walls, forced return and relief systems are essential.

SPECIFIC APPLICATIONS

The two methods shown in Fig. 8 are suitable for application to theaters, churches, and auditoriums. In small or medium size theaters, it is sometimes practicable to use sidewall 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

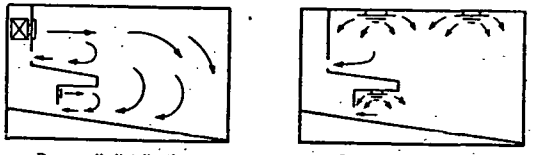


FIG. 8. AIR DISTRIBUTION METHODS FOR THEATERS, CHURCHES, AND AUDITORIUMS

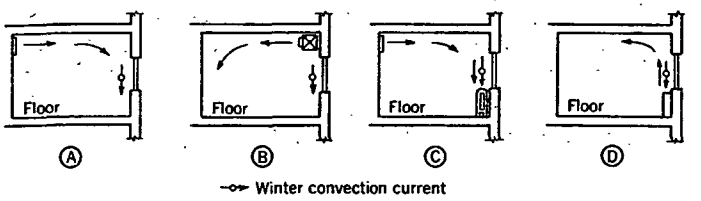


FIG. 9. DISTRIBUTION METHODS FOR SMALL ROOMS

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.

the front of the theater to prevent cold spots which may result from exposed wall convection or infiltration from exits. Return intakes may be located higher where the exits and stage have separate means of heating.

Diagrams shown in Fig. 9 illustrate distribution methods for small rooms with exposed walls, such as for offices, hospital rooms, hotel rooms, apartments, etc. The cooling performance of various distribution methods as applied to a small store is shown in Fig. 10.

For specific requirements in connection with air distribution in marine applications see Chapter 49.

BALANCING SYSTEM

In designing an air conditioning system, it should be the aim of the engineer to so proportion the duct system that proper distribution of air to every supply opening will be obtained. Since this is almost impossible to accomplish in practice, it becomes necessary to have means of balancing

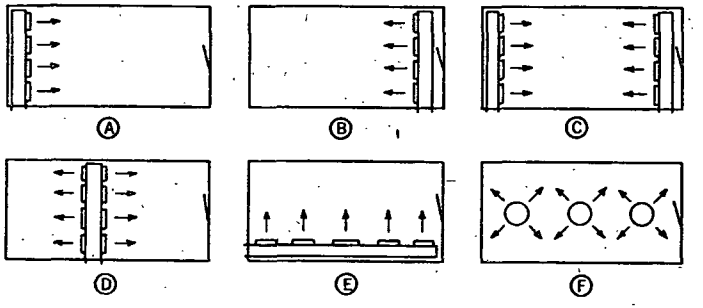


FIG. 10. SMALL STORE COOLING DISTRIBUTION

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 down drafts 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.

the system to secure the desired amount of air in each space. There are a number of ways in which this may be accomplished, some of which are:

1. Dampers on the supply and return faces.
2. Dampers in the supply and return ducts.
3. Reducing the effective area of some supply openings by blank-offs.
4. Combinations of dampers in both supply and return air.

Dampers on the supply faces themselves are objectionable unless of special design, because of their effect on the air stream and noise. Dampers on the return faces are frequently objectionable because of noise. A damper in the supply duct at some distance back of the supply opening forms a very satisfactory means of regulating the flow without disturbing distribution across the supply opening face. A damper in the return air duct has the advantage over one immediately behind the face in that it does not tend to create high localized velocities through the face as the latter might do if nearly closed. Blank-offs consisting of pieces of sheet metal covering a portion of the supply opening face are frequently used. Results are often unsatisfactory due to high velocity through the remaining open portion of the grille. Determination of just what is required is a matter of experiment, and the balancing of the system is not nearly so conveniently accomplished as with dampers. Dampers in both supply and return air form the most flexible means of controlling the supply to the room and the static pressure within the room. When feasible, these dampers, particularly those in the supply ducts, should be a substantial distance from the supply opening, and ahead of the acoustical duct lining if used. Due consideration should also be given to the use of the several volume control and uniform distribution devices now available. See *Catalog Data Section*.

REFERENCES

[1]—Control of Air Streams in Large Spaces, by G. L. Tuve and G. B. Priester (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 153).

[2]—Air Distribution, by W. L. McGrath, unpublished private paper.

[3]—A.S.H.V.E. RESEARCH REPORT No. 1155—The Performance of Stack Heads, by D. W. Nelson, D. H. Krans and A. F. Tuthill (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 205).

[4]—A.S.H.V.E. RESEARCH REPORT No. 1226—Performance of Side Outlets on Horizontal Ducts, by D. W. Nelson and G. E. Smedberg (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 58).

BIBLIOGRAPHY

A.S.H.V.E. RESEARCH REPORT No. 1076—Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 77).

A.S.H.V.E. RESEARCH REPORT No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 645).

A.S.H.V.E. RESEARCH REPORT No. 1162—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 313).

A.S.H.V.E. RESEARCH REPORT No. 1165—Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 351).

A.S.H.V.E. RESEARCH REPORT No. 1204—Entrainment and Jet-Pump Action of Air Streams, by G. L. Tuve, G. B. Priester and D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 241).

Air Flow at Discharge of Fan Pipe Lines in Mines, by G. E. McElroy, (*Bureau of Mines*, Report of Investigations—No. 3730, November, 1943).

Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne, and W. A. Grant (Pitman Publishing Corp., New York, N. Y., 1940).

The Rationale of Air Distribution and Grille Performance, by C. O. Mackey (*Refrigerating Engineering*, Vol. 35, No. 6, June 1938, p. 417).

CHAPTER 41

Air Duct Design

Pressure Losses, Friction Losses, Friction Loss Chart, Equivalent Circular and Rectangular Ducts, Elbow Friction Losses, Proportioning the Losses, Duct Sizes, Procedure for Duct Design, Velocities, Main Trunk Ducts, Velocity Method, Equal Friction Method, Duct Construction Details, Duct Heat Loss and Insulation

THE theoretical resistance of an air handling system can be computed from the methods and data given in this chapter. The actual resistance for any given installation, however, may vary considerably from the calculated resistance because of variation in the smoothness of materials, the type of joints used and the ability of the mechanics to fabricate in accordance with the design. It is best to select fans and motors of sufficient size to allow a factor of safety. Volume dampers should be installed in each branch outlet to balance the system, and the necessary allowance for this balancing should be made in calculating the pressure loss in the system.

The flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

The rate of flow of air in a duct may be obtained from Equation 1, which is developed as Equation 70 in Chapter 4:

$$V_m = 1096.5 \sqrt{\frac{h_w}{\rho_a}} \qquad (1)$$

where

V_m = velocity of fluid, feet per minute.
h_w = velocity head or pressure, inches of water.
ρ_a = density of air, pounds per cubic foot.

For dry air (69.41 F and 29.921 in. Hg barometer) $\rho_a = 0.075$ lb per cubic foot [1]. Substituting this value in Equation 1:

$$V_m = 1096.5 \sqrt{\frac{h_w}{0.075}} = 4005 \sqrt{h_w} \qquad (2)$$

The relation of air velocity and velocity head expressed in Equation 2 is shown diagrammatically in Fig. 1 for dry air at 69.41 F and 29.921 in. Hg barometer.

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses for turbulent flow (which occur in all practical air flow problems) are due to the friction of air against the sides of the duct and to internal friction between air molecules. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to

0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design.

FRICTION LOSSES

Friction loss in *straight* ducts is most readily calculated by means of the Air Friction Chart, Fig. 2, which was developed by recent research at the A.S.H.V.E. Research Laboratory [2]. This chart is constructed from the basic flow equation:

$$h_f = f \frac{l}{D} \frac{V^2}{2g} \tag{3}$$

where

h_f = head loss due to friction, in feet of fluid flowing.
l = length of conduit, feet.
D = inside diameter of conduit, feet.
V = fluid velocity, feet per second.
g = acceleration due to gravity, 32.17 fps per second.
f = a non-dimensional friction coefficient which, for ventilation work depends upon Reynolds Number and the relative roughness of the conduit. Appropriate values of f were taken from the work of Moody [3].

The air friction chart is based on standard air at 29.921 in. of mercury barometer and 70 F, flowing through average clean round galvanized metal ducts having approximately 40 joints per 100 ft. For the average application, it should prove sufficiently accurate, without corrections, for any air temperature from 50 F to 90 F, for any relative humidity, and for any normal variation in barometric pressure. For widely varying air pressures or temperatures or for unusual duct conditions, the friction values obtained from the chart should be corrected. If precise results are desired, corrections should be made by use of Equation 3, with the appropriate value of the coefficient of friction, f. However, the following approximate equations [2] are presented and will be found sufficiently accurate for most purposes.

For perfectly smooth ducts: $$h_o = h_s \left(\frac{\rho_o}{\rho_s}\right)^{0.81} \left(\frac{\mu_o}{\mu_s}\right)^{0.19} \tag{4}$$

For average ducts, $$h_o = h_s \left(\frac{\rho_o}{\rho_s}\right)^{0.90} \left(\frac{\mu_o}{\mu_s}\right)^{0.1} \tag{5}$$

For very rough ducts, $$h_o = h_s \left(\frac{\rho_o}{\rho_s}\right) \tag{6}$$

where

h = friction loss.
ρ = density.
μ = absolute viscosity.

h, ρ and μ may be in *any consistent units*. *Subscripts* o and s refer to *actual operating* and *standard* conditions respectively.

The absolute viscosity of dry air at various temperatures is given in Fig. 3. The effect of relative humidity on viscosity may be neglected.

Example 1. Assume that it is desired to circulate 10,000 cfm of air through 75 ft of 24 in. diameter pipe. Find 10,000 cfm on the left scale of Fig. 2 and move horizontally 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 × 0.50 = 0.38 in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

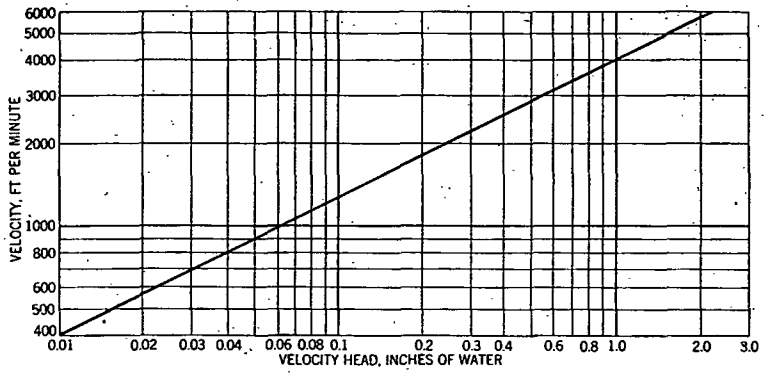


FIG. 1. RELATION BETWEEN VELOCITY AND VELOCITY HEAD FOR DRY AIR

Circular Equivalents of Rectangular Ducts

An air handling system is usually sized first for round ducts and, if rectangular ducts are desired, their sizes are selected to provide air carrying capacities equivalent to those of the round ducts originally selected. Table 1 gives directly the rectangular duct sizes equivalent to each size of circular duct in friction and capacity. The values in this table were obtained from the equation:

$$d = 1.265 \sqrt[5]{\frac{(a\,b)^3}{a+b}} \tag{7}$$

where

a = one side of rectangular duct feet or inches.
b = other side of rectangular duct, feet or inches.
d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity, feet or inches.

Rectangular equivalents of round ducts are also given in the curves of Fig. 4 which are plotted from data based on Equation 7. To use the chart, locate the diagonal curve giving the diameter of the round duct. The width and height of an equivalent rectangular or square duct may then be read as the abscissa and the ordinate of any point on the curve.

Multiplying or dividing the length of each side of a pipe 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 × 23.3 = 46.6 in.

A comprehensive study of equivalent duct sizes is to be made by the A.S.H.V.E. Research Laboratory in the near future for the purpose of determining whether Table 1 and Fig. 4 should be revised.

Elbow Friction Losses

It is customary to express the dynamic and friction losses in elbows as equal to a number of diameters of round pipe, or a number of widths of rectangular pipe, or equivalent length of duct [4]. The curves in Fig. 5 give the number of diameters or widths of pipe which have a frictional

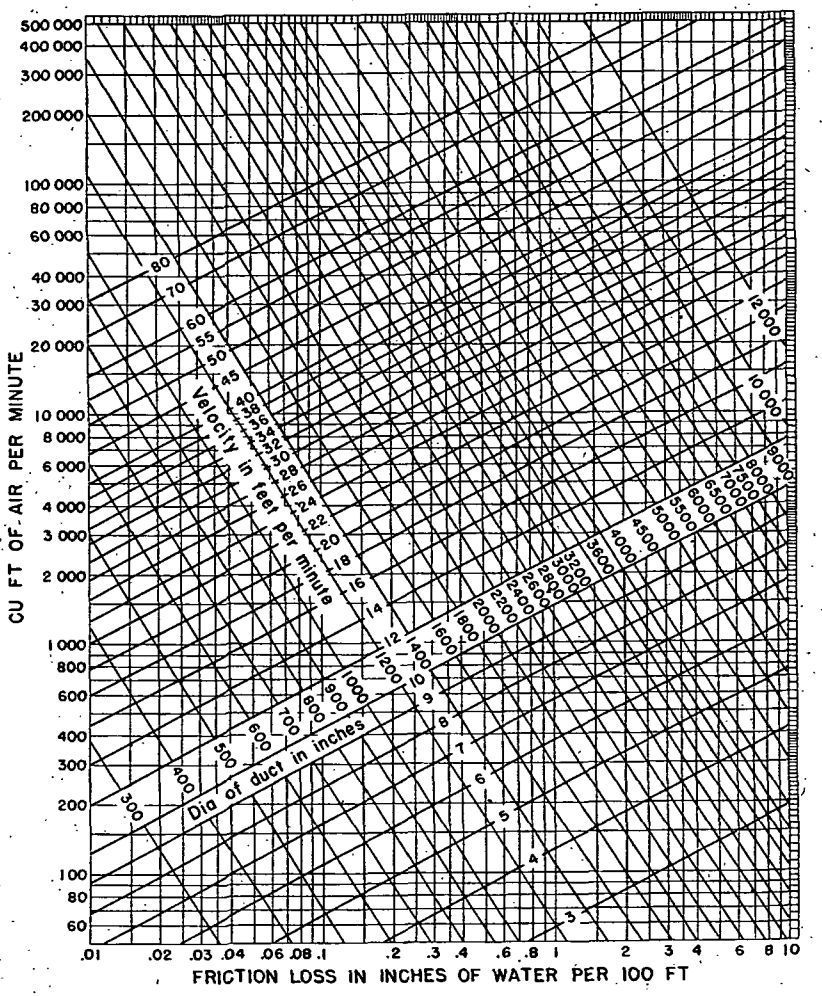


FIG. 2. FRICTION OF AIR IN STRAIGHT DUCTS

(Based on Standard Air at 29.921 In. of Mercury barometric pressure and 70 F flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.)

resistance equivalent to the pressure drop in the elbows. Curves *B* and *C* are based on tests of round and square elbows [5] of ordinary good sheet metal construction.

Values obtained from Curve *A* should be used when there is any doubt as to quality of duct construction. It is suggested that this curve be used for rectangular elbows and five-piece elbows as it will thus allow an additional factor of safety without seriously affecting the design.

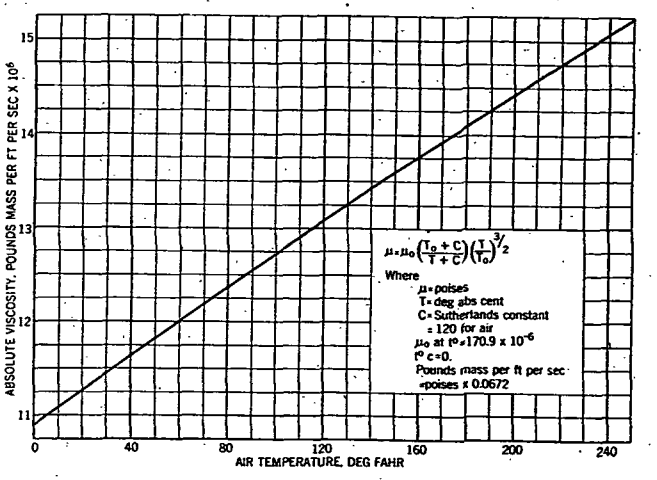


FIG. 3. TEMPERATURE—VISCOSITY CURVE FOR DRY AIR

As indicated in Fig. 5, long radius elbows will offer much less resistance to the flow of air than short radius elbows. Experience has shown that good results may be expected when the radius to the center of the elbow is 1.5 times the pipe diameter or duct width parallel to the radius. Examination of the curve will indicate that little advantage is to be gained by selecting elbows having a centerline radius of more than two diameters [6]. Elbows having a radius of more than three diameters show a slightly increased resistance due to the increased length of pipe but, when used, they reduce the over-all resistance of the system and therefore should not be avoided.

Where space conditions necessitate the use of short radius or miter elbows in square or rectangular duct work, turning vanes should be used to reduce the pressure losses. Rough or raw edges on the vanes should be avoided to prevent objectionable noise. Typical types of vanes are shown in Table 2 with the total equivalent length of duct that may be used in estimating the resistance of each type.

The pressure loss through elbows of less than 90 deg may be assumed to be directly proportional to the ratio of the angle through which the

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION[a]

Side Rectangular Duct	8	8.5	9	9.5	10	10.5	11	11.5	12	12.5	13	13.5	14	14.5	15	15.5	16
3	5.2	5.4	5.5	5.7	5.8	5.9	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.1
3.5	5.7	5.9	6.0	6.2	6.3	6.5	6.6	6.7	6.9	7.0	7.1	7.3	7.4	7.5	7.6	7.7	7.8
4	6.1	6.3	6.5	6.7	6.8	7.0	7.1	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.3	8.4
4.5	6.5	6.7	6.9	7.1	7.2	7.4	7.6	7.7	7.9	8.0	8.2	8.4	8.5	8.6	8.7	8.9	9.0
5	6.9	7.1	7.3	7.5	7.7	7.8	8.0	8.2	8.3	8.5	8.7	8.8	8.9	9.1	9.2	9.4	9.5
5.5	7.3	7.5	7.7	7.8	8.1	8.3	8.5	8.6	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.9	10.1

[a] Additional sizes: 4 × 5 = 4.9; 4 × 6 = 5.4; 4 × 7 = 5.8; 5 × 5 = 5.5; 5 × 6 = 6.3; 5 × 7 = 6.5.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Continued)

SIDE RECTANGULAR DUCT	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	8.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Concluded)

SIDE RECTANGULAR DUCT	26	28	30	32	34	36	38	40	42	44	46	48
26	28.6											
28	29.7	30.8										
30	30.7	31.9	33.0									
32	31.7	32.9	34.1	35.2								
34	32.7	33.9	35.1	36.3	37.4							
36	33.7	34.9	36.1	37.3	38.5	39.6						
38	34.6	35.9	37.1	38.4	39.5	40.7	41.8					
40	35.3	36.7	38.0	39.3	40.5	41.7	42.9	44.0				
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2			
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4		
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6	
48	38.5	40.0	41.5	43.0	44.4	45.6	46.9	48.1	49.3	50.5	51.6	52.8
50	39.2	40.8	42.3	43.8	45.2	46.5	47.9	49.1	50.4	51.6	52.9	54.0
52	40.0	41.6	43.1	44.7	46.1	47.5	48.9	50.1	51.3	52.5	53.8	55.0
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52.3	53.5	54.8	56.0
56	41.3	43.0	44.6	46.2	47.7	49.1	50.6	52.0	53.3	54.6	55.9	57.0
58	42.1	43.8	45.4	47.0	48.5	50.0	51.5	52.9	54.2	55.5	56.8	58.0
60	42.7	44.5	46.1	47.8	49.3	50.9	52.3	53.8	55.0	56.4	57.7	58.9
62	43.4	45.1	46.8	48.4	50.0	51.7	53.0	54.5	55.9	57.2	58.5	59.7
64	44.0	45.8	47.5	49.2	50.9	52.4	53.9	55.4	56.8	58.1	59.4	60.6
66	44.7	46.5	48.2	50.0	51.6	53.1	54.7	56.2	57.6	59.1	60.4	61.6
68	45.3	47.2	48.9	50.7	52.2	53.8	55.5	56.9	58.4	59.9	61.3	62.6
70	46.0	47.8	49.5	51.3	52.9	54.5	56.2	57.7	59.1	60.6	62.1	63.5
72	46.5	48.4	50.1	51.9	53.7	55.4	57.0	58.7	60.0	61.3	63.0	64.5

SIDE RECTANGULAR DUCT	50	54	60	66	72	78	84	88
50	55.0							
52	56.1							
54	57.2	59.4						
56	58.3	60.5						
58	59.3	61.6						
60	60.3	62.7	66.0					
62	61.3	63.7	67.1					
64	62.2	64.7	68.2					
66	63.2	65.7	69.3	72.6				
68	64.1	66.6	70.3	73.7				
70	65.0	67.6	71.3	74.8				
72	65.9	68.5	72.3	75.9	79.2			
74	66.8	69.4	73.3	76.9	80.3			
76	67.6	70.3	74.2	77.9	81.4			
78	68.4	71.2	75.2	78.9	82.5	85.8		
80	69.2	72.1	76.1	79.9	83.6	86.9		
82	70.1	73.0	77.1	80.9	84.6	88.0		
84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

turn is made. The resistance will vary widely for the large degree turns depending upon the aspect ratio and the length of straight pipe between the elbows, but for practical purposes, it may be assumed that the ratio remains proportional to the angle through which the turn is made. Reverse 90 deg elbow turns should be avoided wherever possible but,

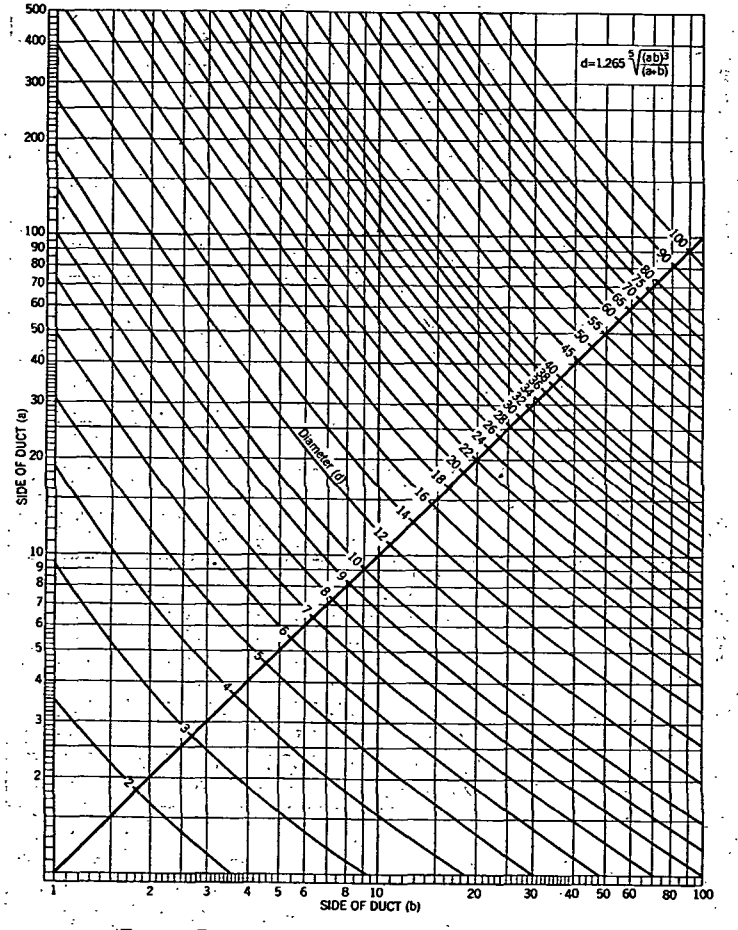


FIG. 4. RECTANGULAR EQUIVALENTS OF ROUND DUCTS

where used, the friction indicated in Fig. 5 should be doubled for the second elbow. Additional tests are needed on the loss of pressure in elbows and other types of duct fittings in order to reconcile difference between values shown in Fig. 5 and results of more recent tests shown in Table 2. A study of this subject will be undertaken at the A.S.H.V.E. Research Laboratory for the purpose of obtaining such data.

PROPORTIONING THE LOSSES

The entrance loss through the outside air intake louvers will vary with the design of the louvers and method of connection to the system. The louvers and connecting duct will have a friction resistance of from 0.25 to 1.00 times the velocity pressure. Therefore, the total entrance loss will vary from 1.25 to 2.00 h_w. Common practice is to use 1.5 h_w for a 75 per cent free area louver with connecting duct having 15 deg tapered sides. Wherever air passes through a plenum space having a negligible velocity, allowance must be made for the loss in velocity head. This may be taken as the velocity head corresponding to the difference in velocities in the plenum and the duct. Where the ducts are very smooth with long transformation fittings, a regain in static pressure is sometimes allowed,

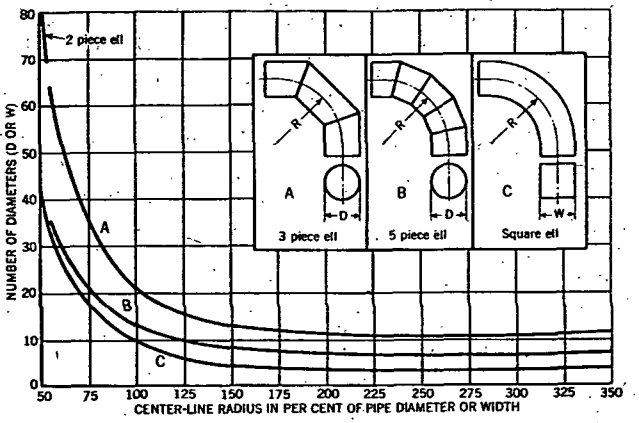


FIG. 5. LOSS OF PRESSURE IN ELBOWS

but ordinary construction does not warrant a consideration of this factor, and it is customary to neglect it. When it is allowed, the regain is estimated at one-half the difference between the velocity pressure at the fan outlet and at the last run of pipe.

Other losses of pressure occur through the heating units, at the air washer and at air filters. In the design of an ideal duct system, all factors should be considered and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

DUCT SIZES

Ducts and flues for gravity circulation must be sized so that the friction loss will not exceed 50 per cent of the available chimney effect due to the temperature and height of the column of heated air. Duct systems for mechanical circulation may be sized so as to have much higher pressure losses than gravity systems. The total pressure of these systems is limited to the pressure which the fan will produce.

The general rules to be followed in the design of a duct system are:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.

2. Sharp elbows and bends should be avoided unless turning vanes are used.

3. Transformation pieces should be made as long as possible. The angle between the sides and axis of the duct should never exceed 30 deg and, where possible, 15 deg should be made the maximum.

4. Especial care should be taken to maintain a true cross-section and not to restrict the air flow either in transformation pieces or in elbows.

5. Rectangular ducts or flues should be made as nearly square as possible. Good practice limits the ratio between the long side and the short side to 3 to 1. In no case should this ratio exceed 10 to 1.

6. Wherever possible, ducts should be constructed of smooth material such as sheet metal. Where masonry ducts are used, proper allowance for the surface friction coefficient should be made.

7. The use of furred spaces, spaces between joists, etc., should be avoided unless lined with sheet metal.

TABLE 2. EFFECT OF VANES ON PRESSURE LOSS OF 7-INCH SQUARE VENTILATING DUCT[a]

Expressed in feet of total equivalent length of duct (ELD)

SQUARE MITER ELBOW								STANDARD ELBOWS WITH VARIOUS RADII							
	Radius Ratio R_1/W	0	.2	.4	.6	.8	1.0		Radius Ratio R_1/W	0	.2	.4	.6	.8	1.0
	ELD, ft	41.1	30.5	27.5	30.1	37.7	38.5		ELD, ft	39.7	23.3	22.0	25.7	28.9	39.7
	Radius Ratio R_1/W	0	.2	.3					Radius Ratio R_1/W	0	.2	.3	.4	.5	.6
	Radius Ratio R_2/W	0	.4	.5					Radius Ratio R_2/W	0	.4	.5	.6	.7	.8
	ELD, ft	41.1	23.5	23.3					ELD, ft	39.7	20.0	22.0	23.0	23.8	25.7
	Radius R_1/W	0	.2	.3					Radius R_1/W	0	.4	.6	.8	1.0	1.2
	Ratio R_2/W	0	.4	.5											
	Ratio R_3/W	0	.6	.7											
	ELD, ft	41.1	20.7	22.2					$1\frac{1}{2}$" L.R. ELD, ft	25.3	17.7	16.5	18.7	23.5	25.6
									Radius R_1/W	0	.7	.8	.9	1.0	1.2
VANE	A	B		C		D									
ELD, ft	21.8	17.0		17.9		17.8			4" L.R. ELD, ft	14.2	13.3	13.0	12.7	12.5	12.7

[a]For more complete data see A.S.H.V.E. RESEARCH REPORT No. 1216—Effect of Vanes in Reducing Pressure Loss in Elbows in 7-Inch Square Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 409).

Note A: Vane *A* made up of a large number of small splitters; *B* made up of a small number of large splitters bent on a large radius; *C* hollow vanes having different outside and inside curvature; and *D* four splitters with $R/W = 0.4$. Elbow same as *D* except 2 in. trailing edge on the end of each splitter. ELD in feet = 17.0.

Note B: The air velocity has no effect on the loss of elbows when the loss is expressed as equivalent length of duct.

Procedure for Duct Design

The general procedure for designing a duct system is outlined in the several items listed herewith:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work and equipment, and at the same time maintaining a simple design.

2. Arrange the positions of duct outlets to insure the proper distribution of air.

3. Divide the building into zones and proportion the volume of air necessary for each zone.

4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity and throw.

5. Calculate the sizes of all main and branch ducts by either of the following two methods:

a. Velocity Method. Select the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections of the continuous run.

b. Friction Pressure Loss Method. Proportion the duct for equal friction pressure loss per foot of length.

6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained at the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The air velocities given in Table 3 have been found to give satisfactory results in engineering practice. Where the higher velocities are used, the ducts should be cross-braced to prevent breathing, buckling or vibration, and should be constructed of heavier gage metal. At the higher velocities it is particularly important to design the ducts for minimum resistance. As high velocities at one point offset the effect of proper design in all other parts of the system, emphasis should be placed on the importance of air velocities, elbow design, location of dampers, fan connections, grille and register approach connections, and similar details. For industrial buildings, noise is seldom given much consideration, and main duct velocities as high as 2800 or 3000 fpm are sometimes used but, when these velocities are used, due consideration should be given to duct design, resistance pressure, fan efficiencies and motor horsepower. For department stores and similar buildings, 2000 to 2200 fpm are sometimes used in main ducts where noise is not objectionable and space conditions warrant it.

Where high velocity diffusing outlets are used, the duct velocity should not be less than the throat velocity of the diffusers, as dynamic losses

TABLE 3. RECOMMENDED AND MAXIMUM DUCT VELOCITIES

DESIGNATION	RECOMMENDED VELOCITIES, FPM			MAXIMUM VELOCITIES, FPM		
	Residences	Schools, Theaters, Public Buildings	Industrial Buildings	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outside Air Intakes[a]	700	800	1000	800	900	1200
Filters[a]	250	300	350	300	350	350
Heating Coils[a]	450	500	600	500	600	700
Air Washers	500	500	500	500	500	500
Suction Connections	700	800	1000	900	1000	1400
Fan Outlets	1000–1600	1300–2000	1600–2400	1700	1500–2200	1700–2800
Main Ducts	700–900	1000–1300	1200–1800	800–1000	1100–1400	1300–2000
Branch Ducts	600	600–900	800–1000	700	800–1000	1000–1200
Branch Risers	500	600–700	800	650	800–900	1000

[a]These velocities are for total face area, not the net free area.

occur wherever velocities are stepped up or down. One recent trend in grille design is toward the use of much higher grille and branch duct velocities. Some installations have been made with velocities as high as 1600 fpm in branches and through the net area of grilles, but many of these have proven unsatisfactory because of noise and drafts.

Grille manufacturers publish selection tables which size the grilles for volume of air, temperature differential, and distance of throw. In following these tables, maximums should be avoided and the manner in which the duct connects to the grille should be given careful consideration. Most of the selection tables are based on straight approach to the grille. Elbow connections to supply grilles should be provided with turning vanes to equalize the face velocity. See Chapter 40 for a discussion of grilles.

Fan outlet velocities are discussed in Chapter 32 and will not be dealt with here except to indicate that fan noises should be given proper consideration.

Main Trunk Ducts

Main trunk ducts with branches are commonly used to convey the air from the fan to the grille or register outlets in preference to individual ducts from the fan to these outlets. The velocities in these ducts and branches vary according to the nature of the installation and the degree of quietness desired. The recommended velocities in Table 3, with good construction, should give satisfactory results. The maximum velocities indicated should not be used except where noise is not a deciding factor.

Velocity Method

The velocity method of designing a duct system involves the arbitrary selection of velocities for various sections of the duct system with the highest velocities at the fan and progressively lower velocities toward the duct openings to the room. To find the total static pressure against which the fan must operate, the static pressure loss of each section must be calculated separately and the total loss found by adding the individual losses of the various sections of the run having the highest resistance. Usually this is the longest run but in some cases a shorter run may have more elbows, transformations, booster heaters, etc., which will cause it to have a higher resistance pressure. This method requires judgment and experience in choosing the proper velocities to approach equal friction for all lengths of run but many engineers believe that the velocity method is handier to use than other methods and will give satisfactory results for most practical applications. The air velocities given earlier in this chapter are helpful in choosing proper velocities. Adjustable dampers or splitters are used to regulate air quantities delivered.

Equal Friction Method

The equal friction method of design is sometimes preferred because it does not require nearly so much judgment and experience in selecting the proper velocities in the various sections of a system. The usual procedure in this method of design is to select the main duct velocity to be consistent with good practice from a standpoint of noise for a particular type of building. This velocity should be less than the fan outlet velocity. All main ducts and branch ducts are sized for equal friction by the use of Fig. 2, and Table 1 or Fig. 4.

In cases where the fan or factory assembled air conditioning unit has a limited external resistance, it is necessary to divide the available resistance

by the total equivalent length of the longest or most complicated run of duct to determine the resistance per 100 ft and then to size all ducts at this resistance value. This will automatically determine the duct velocities and give the desired total duct resistance. A further refinement, which is sometimes used in large systems, is to size each branch duct so that it has a resistance equal to the resistance of the main system at the point of juncture. Even when this refinement is added, regulating dampers are recommended in each branch.

After the duct system is designed the frictional resistance is calculated and tabulated together with the resistance of all component parts. The

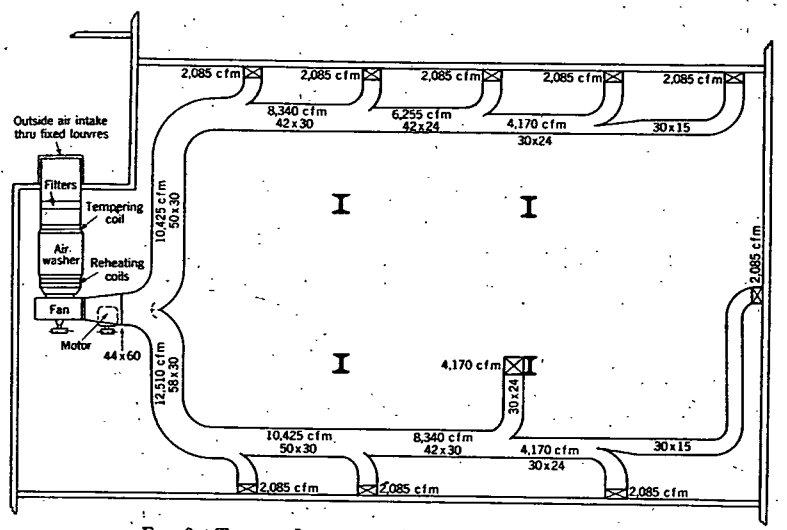


FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

fan is then selected for the required volume of air, static pressure and outlet velocity.

Example 2. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms and offices. The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-min air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The free area of the outdoor air inlet is based on a velocity of 1000 fpm or 22,935 ÷ 1000 ≐ 22.94 sq ft. The main duct velocity selected from Table 3 is 1250 fpm which gives a main duct area of 22,935 ÷ 1250 = 18.354 sq ft (60 × 44 in.). From Table 1 a 60 × 44 in. duct is approximately equivalent to 56 in. diameter.

Referring to Fig. 2, a volume of 22,935 cfm through a 56 in. diameter duct gives a resistance of 0.033 in. per 100 ft. The amount of air to be handled by each section of pipe is shown in Fig. 6, and by locating each of these values on the 0.033 in. friction line, the round pipe sizes are obtained and then, referring to Table 1, the equivalent rectangular sizes are selected as shown in Table 4.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base

TABLE 4. PIPE SIZES FOR EXAMPLE 2[a]

VOLUME OF AIR (CFM)	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
22,935	56	60 x 44
12,510	45	58 x 30
10,425	42	50 x 30
8,340	39	42 x 30
6,255	35	42 x 24
4,170	29.5	30 x 24
2,085	23	30 x 15

[a]Velocity through grilles (not shown) to be approximately 300 fpm.

of each riser, or in each branch duct, and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to those shown in Fig. 6 where the main duct divides into the 58 x 30 in. and 50 x 30 in. branches.

Resistance Losses for the System

(1) Outdoor air intake, 1000 fpm velocity (1.5 heads × 0.0625) ..... 0.094 in.

(2) Filters (from manufacturer's tables) ..... 0.250 in.

(3) Tempering coil loss (from manufacturer's tables) ..... 0.074 in.

(4) Air washer loss (from manufacturer's tables) ..... 0.250 in.

(5) Reheating coil loss (from manufacturer's tables) ..... 0.083 in.

(6) Duct resistance:

The longest run is ..... = 150 ft

Two, 58 x 30 in. elbows (150% ratio) $\frac{2 \times 13 \times 58}{12}$ ..... = 126 ft

Two, 30 x 15 in. elbows (150% ratio) $\frac{2 \times 13 \times 30}{12}$ ..... = 65 ft

Three, 15 x 30 in. elbows (75% ratio) $\frac{3 \times 35 \times 15}{12}$ ..... = 131 ft

Total equivalent run ..... 472 ft

472 ft at 0.033 in. per 100 ft ..... 0.156 in.

(7) Allowance for damper adjustment, 25% of 0.156 ..... 0.039 in.

(8) Supply grille resistance (from manufacturer's tables) ..... 0.036 in.

Total static pressure loss of system ..... 0.982 in.

The fan is selected from the manufacturer's ratings to deliver 22,935 cfm at a static pressure of 0.982 in. as outlined in Chapter 32.

Example 3. If the rooms and offices of the hotel building of Example 2 are to be served from a manufactured unit with a capacity of 22,935 cfm against an external resistance of 0.35 in., the known resistances are calculated as:

(1) Outdoor air inlet ..... 0.094 in.

(2) Supply grille resistance (from manufacturer's tables) ..... 0.036 in.

Total known resistance ..... 0.130 in.

Subtracting this from the total available resistance: 0.35 in. − 0.130 in. = 0.220 in. available for duct resistance and damper adjustment.

If 25 per cent is allowed for damper adjustment, the total allowable duct friction = 0.220 ÷ 1.25 = 0.176 in.

Known length of run ..... 150 ft

The duct width is then estimated for the following elbow calculations:

Four 150% ratio elbows, 4 x 13 x 3.5 ft ..... 182 ft

Three 75% ratio elbows, 3 x 35 x 1.5 ft ..... 158 ft

Total estimated length ..... 490 ft

The duct friction per 100 ft is then 0.176 ÷ 4.90 = 0.036 in. and the mains and branches are sized from the 0.036 in. friction line in Fig. 2.

If it is desired to size each branch for equal resistance, the total resistance back to the point of juncture is calculated and the branch is then sized in a manner similar to that outlined in Example 3.

DUCT CONSTRUCTION DETAILS

Straight sections of round duct are usually formed by rolling the sheets to the proper radius and grooving the longitudinal seam. Rectangular ducts are generally constructed by breaking the corners and grooving the longitudinal seam, although some fabricators still use the standing seam due to lack of equipment. 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 complicated fittings such as double compounded elbows are usually constructed with double seam corners. The construction of these various seams as well as the types of girth connections are shown in Fig. 7. The application of the various slips and connections is outlined in Table 5. The end slip may be used wherever *S* slips are recommended. Where drive slips are used the end slip may be applied on the narrow side of the duct and the drive slips on only the maximum side. Ducts 25 to 30 in. in size should be reinforced between the joints, but not necessarily at the joint. Ducts 31 in. and up should be reinforced at the joint and between the joints; if drive slips are used the angles are usually riveted to the duct about 2 in. from the slips. It is good practice to cross-break or kink all flat surfaces to prevent vibration or buckling due to the air flow and accompanying variations in internal pressure. Round ducts are sometimes swedged 1.5 in. from the ends so that the larger end will butt against the swedge and are held in place with sheet metal screws. Where swedges are not used it is general practice to paste the joint with asbestos paper to insure a tight joint.

The construction of elbows and changes of shape cannot be definitely outlined because of the varied conditions encountered 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 (asbestos cloth if there is a fire hazard) are recommended on both the inlet and outlet to all fans. The fan discharge connections shown in Fig. 7 are marked *good*, *fair*, and *poor* in the order of the amount of turbulence produced. An inspection of the heater connections shown in Fig. 7 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 per cent 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.

The recommended gages for sheet metal duct construction are given in Table 5. Weights of sheet metal per square foot of surface for different gages are given in Table 6. The weights of various gages and the areas for any length of run of rectangular sheet metal ducts may also be determined from Fig. 8. The bottom scale represents the sum of the two

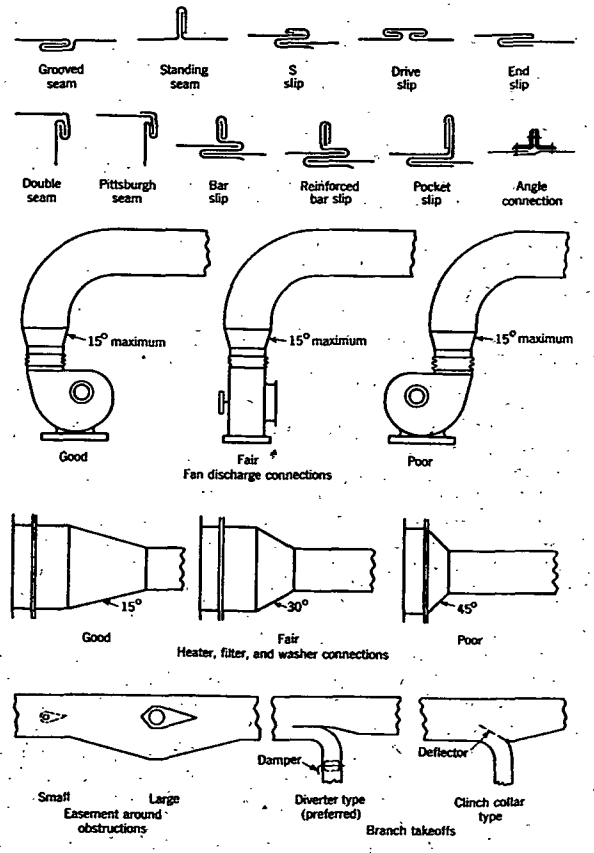


FIG. 7. SHEET METAL DUCT AND ARRANGEMENT DETAILS

sides of the duct and the oblique lines give the length of run in feet. Proceeding horizontally to the right from the intersection of vertical and oblique lines on the chart, the area of the duct may be determined in the first vertical scale. The scales to the right give the weights of the duct run for different gages of metal. In calculating the weights of duct, it is considered good practice to allow 20 per cent additional for weights of joints and bracings. Various weights and thicknesses of standard copper sheets will be found in Table 7.

TABLE 5. RECOMMENDED SHEET METAL GAGES FOR RECTANGULAR DUCT CONSTRUCTION[a]

U. S. STD. GAGE	MAXIMUM SIDE, INCHES	TYPE OF TRANSVERSE JOINT CONNECTIONS[b]	BRACING
26	Up to 12	S, Drive, Pocket or Bar Slips, on 7 ft 10 in. centers	None
24	13 to 24	S, Drive, Pocket or Bar Slips, on 7 ft 10 in. centers	None
	25 to 30	S, Drive, 1 in. Pocket or 1 in. Bar Slips, on 7 ft 10 in. centers[c]	1 x 1 x ⅛ in. angles 4 ft from joint
22	31 to 40	Drive, 1 in. Pocket or 1 in. Bar Slips, on 7 ft 10 in. centers[c]	1 x 1 x ⅛ in. angles 4 ft from joint
	41 to 60	1½ in. Angle Connections, or 1½ in. Pocket or 1½ in. Bar Slips with 1⅜ in. x ⅛ in. bar reinforcing on 7 ft 10 in. centers[c]	1½ x 1½ x ⅛ in. angles 4 ft from joint
20	61 to 90	1½ in. Angle Connections, or 1½ in. Pocket or 1½ in. Bar Slips 3 ft 9 in. maximum centers with 1⅜ x ⅛ in. bar reinforcing	1½ x 1½ x ⅛ in. diagonal angles, or 1½ x 1½ x ⅛ in. angles 2 ft from joint
18	91 and up	2 in. Angle Connections or 1½ in. Pocket or 1½ in. Bar Slips 3 ft 9 in. maximum centers with 1⅜ x ⅛ in. bar reinforcing[d]	1½ x 1½ x ⅛ in. diagonal angles, or 1½ x 1½ x ⅛ in. angles 2 ft from joint

[a]For normal pressures and velocities (see Table 3) utilized in typical ventilating and air conditioning systems. Where special rigidity or stiffness is required, ducts should be constructed of metal two gages heavier. All uninsulated ducts 18 in. and larger should be cross-broken. Cross-breaking may be omitted on uninsulated ducts if two gages of heavier metal are used.

[b]Other joint connections of equivalent mechanical strength and air tightness may be used.

[c]Duct sections of 3 ft 9 in. may be used with bracing angles omitted, instead of 7 ft 10 in. lengths with joints indicated.

[d]Ducts 91 in. and larger require special field study for hanging and supporting methods.

HEAT LOSSES FROM DUCTS

The thermal transmittance U for ducts can be found as follows:

For uninsulated metal duct, $$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o}} \quad (8)$$

For uninsulated non-metallic ducts, $$U = \frac{1}{\frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o}} \quad (9)$$

where U, f_i, f_o, x and k are as defined on page 759.

Where x is small and k is large, however, this factor $\frac{x}{k}$ is of little importance and may be neglected.

Film conductance f_i for air flowing in ducts apparently depends only on the velocity of the air and the diameter of the duct. A fairly reliable inside coefficient can be calculated from Schultz's modified equation:

$$f_i = \frac{0.32\,V_a^{0.8}}{D^{0.25}} \quad (10)$$

where

V_a = velocity of air in duct, feet per second.
D = inside diameter of duct, feet.

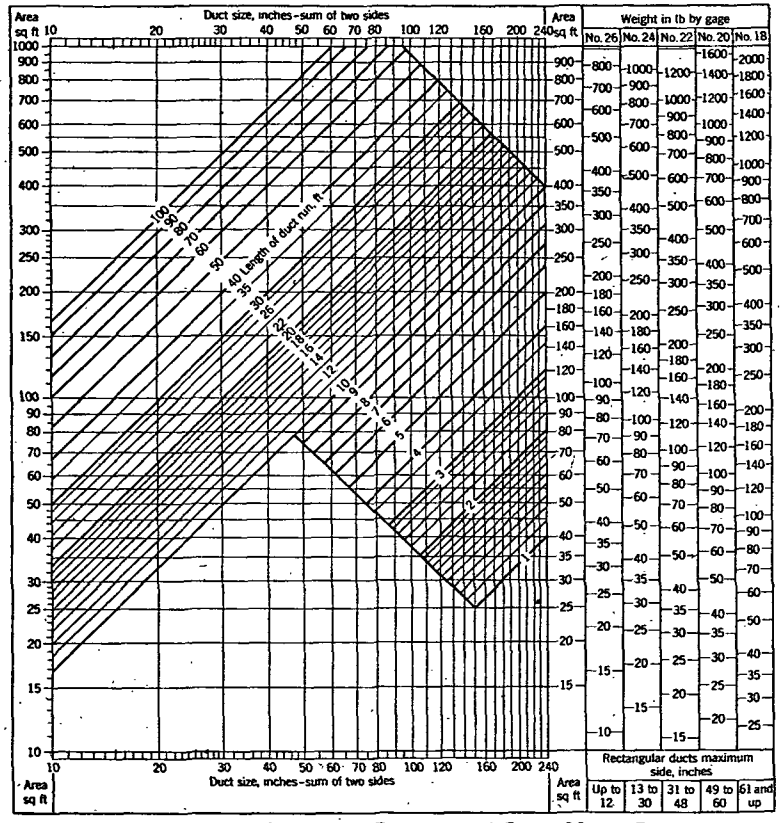


FIG. 8. AREA AND WEIGHT OF RECTANGULAR SHEET METAL DUCTS

Film conductance f_o depends on a number of variables including temperature, diameter, and emissivity of the outer surface and can be calculated from data in Chapter 5. From this explanation, it is seen that it is unwise to recommend a given value of U for all uninsulated metal ducts.

The heat loss from a given length of duct can be expressed by:

$$Q = UPl\left[\left(\frac{t_1 + t_2}{2}\right) - t_s\right] \tag{11}$$

The heat given up by the air in the duct is:

$$Q = 0.24\ W\ (t_1 - t_2) = 14.4\ A\ V_m\ \rho_v\ (t_1 - t_2) \tag{12}$$

Equating (11) and (12):

$$\frac{t_1 + t_2 - 2t_s}{t_1 - t_2} = \frac{28.8\ A\ V_m\ \rho_v}{UPl}$$

Let $y = \dfrac{28.8\ A\ V_m\ \rho_v}{UPl}$ for rectangular ducts, and $\dfrac{7.2DV_m\ \rho_v}{Ul}$ for round

TABLE 6. WEIGHTS OF SHEET METAL USED FOR DUCT CONSTRUCTION

U. S. Std. Gage	Black Sheets: Approximate Thickness, In.		Black Sheets: Weight Per Square Foot		Galvanized Sheets[a]: Approximate Thickness, In.		Galvanized Sheets[a]: Weight Per Square Foot	
	Steel	Iron	Ounces	Pounds	Steel	Iron	Ounces	Pounds
30	0.0123	0.0125	8	0.500	0.0163	0.0165	10.5	0.656
28	0.0153	0.0156	10	0.625	0.0193	0.0196	12.5	0.781
26	0.0184	0.0188	12	0.750	0.0224	0.0228	14.5	0.906
24	0.0245	0.0250	16	1.000	0.0285	0.0290	18.5	1.156
22	0.0306	0.0313	20	1.250	0.0346	0.0353	22.5	1.406
20	0.0368	0.0375	24	1.500	0.0408	0.0415	26.5	1.656
18	0.0490	0.0500	32	2.000	0.0530	0.0540	34.5	2.156
16	0.0613	0.0625	40	2.500	0.0653	0.0665	42.5	2.656
14	0.0766	0.0781	50	3.125	0.0806	0.0821	52.5	3.281
12	0.1072	0.1094	70	4.375	0.1112	0.1134	72.5	4.531
11	0.1225	0.1250	80	5.000	0.1265	0.1290	82.5	5.156
10	0.1379	0.1406	90	5.625	0.1419	0.1446	92.5	5.781

[a]Galvanized sheets are gaged before galvanizing and are therefore approximately 0.004 in. thicker.

ducts, and solve for t_1 and t_2:

$$t_1 = \frac{t_2\ (y + 1) - 2t_s}{(y - 1)} \tag{13}$$

$$t_2 = \frac{t_1\ (y - 1) + 2t_s}{(y + 1)} \tag{14}$$

For low velocities and long ducts of small cross-section, a somewhat more accurate formula may be used as follows:

$$t_2 = \frac{t_1 - t_s}{\left(\dfrac{UPl}{14.4\ A\ \rho_v\ V_m}\right)} + t_s \tag{15}$$

Symbols used in these equations are listed at end of chapter.

In using Equations 13, 14 and 15, 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. 9. The conductivities of various materials, which are based on mean temperatures, ranging from about 70 to 90 F, will be found in Table 2 of Chapter 6. For cases where the mean temperature

TABLE 7. WEIGHTS AND THICKNESSES OF STANDARD COPPER SHEETS[a]
Rolled to Weight

Weight per Square Foot: Ounces	Weight per Square Foot: Pounds	Thickness, Inches: Decimal Equivalent	Thickness, Inches: Nearest Fraction	Nearest Gage No.: B. & S.	Nearest Gage No.: Stubs	Nearest Gage No.: 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.

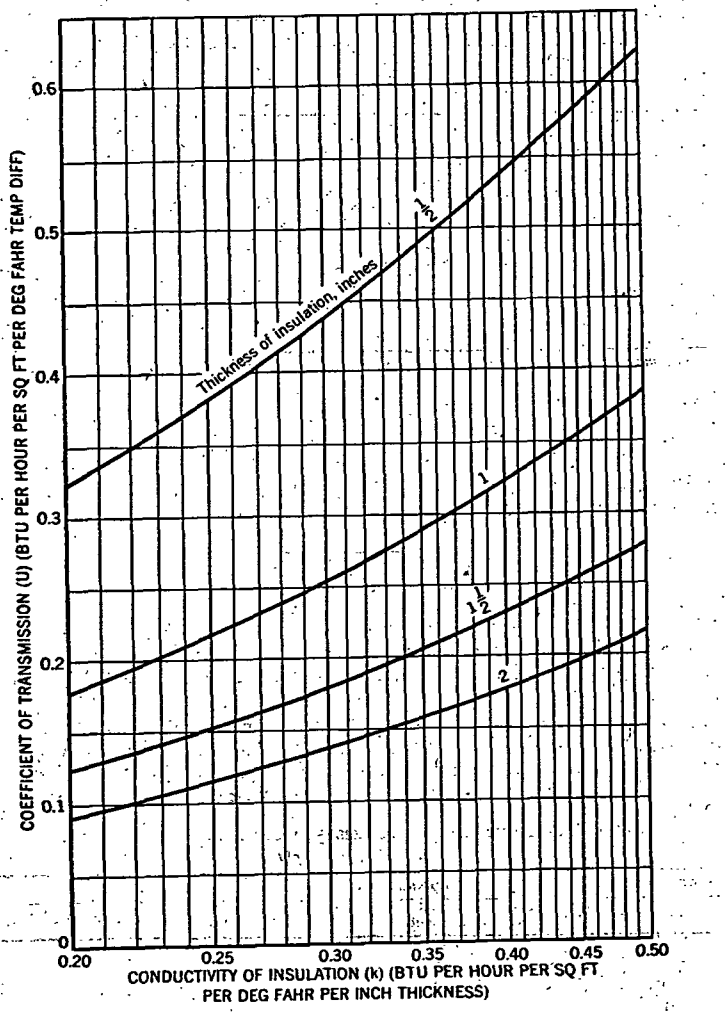


FIG. 9. HEAT LOSS COEFFICIENTS FOR INSULATED DUCTS[a]

[a]For round ducts less than 30 in. diameter, increase heat transmission values by the percentages shown.

THICKNESS OF INSULATION (Inches)	½	1	1½	2
21 to 30 in. Duct Diameter	1%	2%	3%	4%
12 to 21 in. Duct Diameter	3%	5%	7%	9%

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.

Example 4. Determine the entering air temperature and heat loss for a duct 24 × 36 in. cross-section and 70 ft in length, insulated with ½ 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. 9, the over-all heat transmission coefficient is found to be 0.49 Btu. From Table 1, Chapter 3 the density of air at 70 F and 29.921 in. Hg is found to be 0.0749 lb per cubic foot. Substituting these and the other given values in Equation 13:

$$y = \frac{28.8 \times 6 \times 1200 \times 0.0749}{0.49 \times 10 \times 70} = 45.3$$

$$t_1 = \frac{120\ (45.3 + 1) - 80}{45.3 - 1} = 123.7$$

Substituting in Equation 11:

$$Q = 0.49 \times 10 \times 70 \left[\left(\frac{123.7 + 120}{2} \right) - 40 \right] = 28{,}100 \text{ Btu per hour.}$$

For special considerations which apply to insulation of ducts in marine installations see Chapter 49.

LETTER SYMBOLS USED IN CHAPTER 41

μ_o = absolute viscosity of air under actual (operating) conditions, any consistent units.
μ_s = absolute viscosity of air under standard conditions, any consistent units.
ρ_a = density of air, pounds per cubic foot.
ρ_o = density of air under actual (operating) conditions, any consistent units.
ρ_s = density of air under standard conditions, any consistent units.
ρ_v = density of air at specified temperature at which velocity, V_m, is measured, pounds per cubic foot.
A = cross-section area of duct, square feet.
a = length of one side of rectangular duct, feet or inches. (Other side is b.)
b = length of one side of rectangular duct, feet or inches. (Other side is a.)
D = inside diameter of conduit, feet.
d = equivalent diameter, feet or inches.
e = Naperian base of logarithms = 2.718.
f = non-dimensional friction coefficient.
f_i = surface conductance (inside) Btu per (hour) (square foot) (Fahrenheit degree).
f_o = surface conductance (outside) Btu per (hour) (square foot) (Fahrenheit degree).
g = acceleration due to gravity, 32.17, feet per (second) (second).
h_o = friction loss under actual (operating) conditions, any consistent units.
h_f = friction loss, feet of fluid flowing.
h_s = friction loss under standard conditions, any consistent units.
h_w = velocity head or pressure, inches of water.
l = length of conduit, feet.
P = perimeter of duct, feet.
Q = heat loss through duct walls, Btu per hour.
t_1 = temperature of air entering duct, Fahrenheit degrees.
t_2 = temperature of air leaving duct, Fahrenheit degrees.
t_s = temperature of air surrounding duct, Fahrenheit degrees.
U = thermal transmittance coefficient, Btu per (hour) (square foot) (Fahrenheit degree).
V = fluid velocity, feet per second.
V_a = velocity of air in duct, feet per second.
V_m = velocity of fluid, feet per minute.
W = weight of air through duct, pounds per hour.
x = thickness, inches.

REFERENCES

[1]—See Chapter 1 for definition of standard air.

[2]—A New Friction Chart for Round Ducts, by D. K. Wright, Jr. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October-November, 1945, p. 577).

[3]—Friction Factors for Pipe Flow by L. F. Moody (*A.S.M.E. Transactions*, Vol. 66, 1944, p. 671).

[4]—A.S.H.V.E. Research Report No. 1211—Pressure Loss Caused by Elbows in 8-inch Round Ventilating Duct, by M. C. Stuart, C. F. Warner and W. C. Roberts (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 335).

[5]—Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes or Ducts, by F. L. Busey (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913, p. 366).

[6]—Pressure Losses in Rectangular Elbows, by R. D. Madison and J. R. Parker (*Heating, Piping and Air Conditioning*, July, p. 365, August, p. 427, September, p. 483, 1936).

BIBLIOGRAPHY

Method of Determining Rectangular Equivalents and Weights of Ducts, by Peter Franck (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, December, 1940).

The Flow of Liquids, by W. H. McAdams (*Refrigerating Engineering*, February, 1925, p. 279).

Air Conditioning and Engineering (American Blower Corp.).

Fan Engineering (Buffalo Forge Co.).

Heat Power Engineering, by W. N. Barnard, F. O. Ellenwood, and C. F. Hirshfeld, Part III (John Wiley and Sons).

Mechanical Engineers' Handbook, by Lionel S. Marks (McGraw-Hill Book Co.).

Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten, J. B. Schmieler, J. A. Zalovcik, and N. Ivanovic (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 35).

Analysis of Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten, and H. T. Olson (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 193).

The Flow of Fluids in Closed Conduits, by R. J. S. Pigott (*Mechanical Engineering*, Vol. 55, 1933, p. 497).

Mechanical Similitude and Turbulence, by T. von Karman (translated and reprinted as *Technical Memorandum N.A.C.A.* No. 611, 1931).

Turbulent Flow in Pipes, with Particular Reference to the Transition Region between the Smooth and Rough Pipe Laws, by C. F. Colebrook (*Journal, Institute of Civil Engineers*, Vol. II, 1938-39, p. 133).

Evaluation of Boundary Roughness, by H. Rouse (Proceedings Second Hydraulics Conference, *University of Iowa Bulletin* 27, 1943).

The Flow of Air in Ducts, by E. Kemler (*Heating and Ventilating*, May 1936, p. 38).

Performance Tests of Asbestos Insulating Air Duct, by R. H. Heilman and R. A. McArthur (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 197).

A Study of the Data on the Flow of Fluids in Pipes, by E. Kemler (*A.S.M.E. Transactions*, Vol. 55, 1933, Hydraulics, p. 7).

A Rational Method of Duct Design, by L. G. Miller (A.S.H.V E. TRANSACTIONS, Vol. 43, 1937, p. 71).

CHAPTER 42

Sound Control

Unit of Noise Measurement, Apparatus for Measuring Noise, General Problem, Kinds of Noise, Noise Transmitted Through Ducts, Design Room Noise Level, Noise Generated by Fan, Natural Attenuation of Duct System, Duct Sound Absorbers, Air Supply Noises, Grille Selection, Cross Transmission Between Rooms, Controlling Vibration from Machine Mountings

IN ventilating and air conditioning a building or a room, the effect of the mechanical system employed must be considered on the acoustics of the space conditioned. It is important to consider also that the use of air conditioning often permits keeping the windows closed, thus giving relief from certain external noises, but at the same time increasing the necessity of providing adequate sound control.

It is assumed that in a given space the architect and acoustical engineer have produced a room or rooms which are satisfactory for speech, music, or other uses. The ventilating engineer's sole function is to ventilate and air condition these rooms properly so that they will be physically comfortable without adding any acoustical hazards.

UNIT OF NOISE MEASUREMENT

According to an international standard, two terms are used for noise measurement. The *decibel* (db) is the physical unit for expressing intensity or pressure levels. The *phon* is the unit of loudness level. The loudness level, in phons, of any sound is by definition equal to the intensity level in decibels of a thousand cycle tone which sounds equally loud. The decibel is defined by the relation $N = 10 \log_{10} \frac{I_i}{I_o}$, where N is the number of decibels by which the intensity flux I_i exceeds the intensity flux I_o. The intensity flux is the measure of the intensity of a sound wave and is defined in terms of watts per square centimeter passing through a unit area of wave front in a freely traveling plane wave. It is usually more convenient to select an arbitrary reference intensity for I_o and express all other intensities in terms of decibels above that level. For this purpose a reference intensity of 10^{-16} watts per square centimeter has been selected. This intensity is slightly less than the threshold of audibility for the average ear at a frequency of 1,000 cycles per second. This reference level also corresponds to a pressure of 0.0002 dynes per square centimeter for sound in air at usual room temperatures.

A stated sound level in decibels, unless otherwise defined, will thus be related to a threshold of 10^{-16} watts. For example, a level of 60 db above this reference threshold is 10^{-10} watts. In a similar manner, when sound measurements are given in actual intensity or energy units, they can be converted to decibels by this relation.

Since the decibel is based on a ratio, it can only be employed when related to a reference threshold level as given. Noise levels, which vary with frequency as well as intensity, must not only be related to this reference threshold level, but also to a reference frequency, which is taken as 1000 cycles. These terms and procedures may be found in Standards [1] published by the *American Standards Association*.

APPARATUS FOR MEASURING NOISE

Since the relative loudness to the ear, rather than the actual physical intensity, is the quantity in which engineers are usually interested, it has been found necessary to allow for the varying response of the ear at different frequencies in designing noise measuring equipment. It is most satisfactory to measure noise by means of a sound level meter, which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels. This meter is calibrated to give readings above the standard reference level and usually contains a weighting network to make it less sensitive at those frequencies where the ear is less sensitive. Three types of weighting networks may be provided. The "A" network is intended to provide relative frequency sensitivity corresponding to the characteristics of the ear at a loudness level of 40 decibels. The "B" network is weighted to correspond to the ear characteristic at a loudness level of 70 decibels. The meter may also be provided with a "flat" network which provides no weighting by frequency. For complete details on standards for sound level meters, refer to the information[2] published by the *American Standards Association.*

GENERAL PROBLEM OF SOUND CONTROL

As previously stated, the problem confronting the air conditioning engineer is that of designing a system which will operate without increasing the noise level in the conditioned space. To be sure that this is accomplished, it is necessary:

1. To determine the noise level existing without the equipment.
2. To ascertain the noise level which would exist if the equipment were installed without sound control.
3. To provide as a part of the installation sufficient sound control appliances to reduce the noise level substantially to that found in Item 1.

To accomplish this the engineer should have information of three kinds:

1. A knowledge of the noise levels currently considered acceptable in various rooms in order that he may have a basis on which to proceed.
2. A knowledge of the nature and intensity of the noise created by the various parts of the equipment.
3. A knowledge of how, when necessary, to vary and control the noise level between the equipment and the conditioned space.

In addition, the engineer should have sufficient information to predict the levels produced by noises which may be transmitted by the duct system from one conditioned space to another or from an outside space to the conditioned space. In either case, the designer must know the probable noise level at the point where the noise originates. From this he can compute the attenuation or transmission loss required in order to bring this level down to that required in the conditioned space. If there is likelihood of direct transmission through a duct, the attenuation required may be computed as shown on page 763. If the transmission is through dividing walls, it will be necessary to refer to published information on the transmission losses of standard building constructions[3].

Information concerning the noise levels created by ventilating and air conditioning equipment such as fans, motors, air washers, and similar items is not yet on a basis which permits tabular presentation, although certain manufacturers are prepared to offer such data and do state the noise producing properties of their products. A sound test code for fans

has been developed by the *National Association of Fan Manufacturers* which uses the flat response network of the sound level meter. However, when determining the noise generated by an air distribution system, it is customary to use the noise level of the fan as determined by the weighting network most nearly approaching the noise level of the space. In most cases this will be the noise level of the fan as determined on the 40 decibel weighting network. Refer to Table 1 for typical noise levels.

KINDS OF NOISE

To solve a sound problem of this type it is desirable to consider separately the several means by which noise reaches the room. This avoids to some extent the necessity of knowing the noise level at the source, and instead, places the emphasis on ascertaining the level at the point where the sound enters the room.

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 with various sub-divisions:

1. Noise transmitted through the ducts.
   - *a.* From equipment such as sprays, fans, etc.
   - *b.* From outside, and transmitted through duct walls into air stream.
   - *c.* From air currents, including eddying noises.
   - *d.* Cross talk and cross noises between rooms connected by the same duct system.
   - *e.* Noise produced by the grilles.
2. Noise transmitted through the building construction.
   - *a.* From machine mountings as vibration.
   - *b.* From equipment through room wall surfaces.

The next step in the solution of this problem is to present data and discuss methods whereby solutions to 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 TRANSMITTED THROUGH DUCTS

Operation of an air distribution system results in the generation of noise which may be transmitted through the ducts to the ventilated or conditioned room. The transmission of this noise may be controlled by the proper application of sound absorptive material within the ducts. The application of the absorptive material is a problem in balancing the room noise level requirements against the intensity of the noise generated. The four steps in the problem are:

1. Determination of acceptable room noise level resulting from the operation of the equipment.
2. Determination of noise level generated by the equipment.

The difference between items 1 and 2 in decibels is the over-all noise reduction required between the equipment and the room. In the discussion which follows reduction of noise will be referred to as *attenuation* of noise.

3. Determination of the natural attenuation of the duct system.
4. Selection of the proper sound treatment for the duct system.

The difference in decibels between the over-all attenuation required and the natural attenuation (3) is the additional sound attenuation to be provided by absorptive materials installed in the duct system or by special constructions designed to absorb sound. Experience has shown, for example, that where ventilating requirements permit, introduction of an expansion chamber or a change in area in the duct will frequently provide more reduction in low frequency noise.

DESIGN ROOM NOISE LEVEL

Measurements of noise levels have been observed by several investigators in various rooms and locations and are listed in Table 1. The values given were determined with the air conditioning or ventilation equipment not in operation, and with all windows and doors closed simulating the conditions of an actual installation.

This is an important consideration, for in offices or stores adjacent to busy thoroughfares the difference between the typical noise level in the space with the windows and doors open and closed may be as high as 10 db. Minimum, representative, and maximum levels are given for each type of space. The values are intended to give the variation with respect to location and not to time, and may be roughly classified by the following:

TABLE 1. TYPICAL NOISE LEVELS[a]

ROOMS	NOISE LEVEL IN DECIBELS TO BE ANTICIPATED		
	Min.	Representative	Max.
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc.	33	40	48
Theaters, Legitimate	25	30	35
Theaters, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc.	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	30	38	45
Private Offices, Acoustically Untreated	35	43	50
General Offices	50	60	70
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Buildings, Post Offices, etc.	45	55	60
Court Rooms	30	35	45
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	65	77	90
Office Machine Rooms	60	70	80
VEHICLES			
Railroad Coach	60[b]	70	80
Pullman Car	55[b]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	75	80	90

[a]These values are tentative. More detailed measurements by D. F. Seacord, Bell Telephone Laboratories (*Journal Acoustical Society of America*, Vol. 12, pp. 183-187, 1940) give average values and standard deviations of room noise in residences, offices, stores, factories, etc., in large American cities.

[b]For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

Minimum loudness refers to: Spaces of expensive construction, typified by double windows, carpeted floors, heavy upholstered furniture, or acoustically treated walls and ceilings.

Representative loudness refers to: Spaces of average construction and furnishings which are exposed to external noises typical of the locality in which the space is usually found.

Maximum loudness refers to: (1) Any space of inexpensive construction, and bare furnishings where noise is not an important factor. (2) Spaces in close proximity to very intense street traffic or to intense factory noise.

In general, if the noise level in the space resulting only from the operation of the air conditioning equipment is equivalent to or less than the typical level given in Table 1, the installation will prove satisfactory. If the typical level and the equipment level are equal and heard together the resultant level will be 3 db higher than either of them alone.

In some cases it is desirable to keep the equipment noise level in the ventilated or conditioned room at such a value that it actually will not increase the noise level in the room to any measureable degree. This can be accomplished if the equipment noise at the room can be kept 10 db below the noise levels shown in Table 1.

NOISE GENERATED BY FANS

Noise generated by fan wheels may be divided into two classifications, rotational noise and vortex noise. In ventilation and air conditioning work, where the maximum ratio between the fan tip speed and the velocity of sound is not greater than 0.12, vortex noise is by far the most important. The rotational noise may be described as that due to the thrust and torque applied to the air. Vortex noise is that due to the shedding of vortices from the blade and is dependent on the angle of attack, velocity, air turbulence, and blade shape. Vortex noise is due to pressure variations on the blade as a result of variations of air circulation. Given the noise level at the outlet or inlet of one type of fan construction under specific conditions of size, tip speed, and total pressure, noise levels at other values of tip speed, total pressure, and size may be approximated by the relationships:

1. For constant size and point of rating, the noise level of a fan will increase with increasing speed.

$$db \text{ (change)} = 55 \log_{10}\left(\frac{RPM_2}{RPM_1}\right) \qquad (1)$$

2. For constant pressure and tip speed, the noise level of a given type of fan will increase with increasing fan size.

$$db \text{ (change)} = 20 \log_{10}\left(\frac{\text{Size}_2}{\text{Size}_1}\right) \qquad (2)$$

Fan size refers to wheel diameter, housing height or some dimension that is directly proportional to lineal units. Fan sizes based on arbitrary systems or systems of preferred numbers have no significance.

The noise of a given fan is not constant at constant speed if the air delivery changes due to change of resistance. In general, a backward curved blade fan is lowest in noise at or near the point of maximum efficiency; a forward curved blade fan at or between the point of maximum efficiency and shut-off; an axial flow fan at or between the point of maximum efficiency and free delivery. The noise level of a double width fan

TABLE 2. ATTENUATION IN STRAIGHT SHEET METAL DUCT RUNS

DUCT	SIZE, IN.	ATTENUATION PER FT, db
Small	6 x 6	0.10
Medium	24 x 24	0.05
Large	72 x 72	0.01

TABLE 3. ATTENUATION OF ELBOWS[a]

ELBOW	SIZE, IN.[b]	ATTENUATION PER ELBOW, db
Very small	2 wide	3
Small	3 to 15	2
Medium	15 to 36	1.5
Large	36 plus	1

[a]The attenuation in vaned elbows should be considered the same as in elbows having the same dimensions as the radius of curvature of the vanes. If the vanes are lined for the purpose of damping any vibrations in them, one third may be added to the attenuation values listed.

[b]These attenuation values are based on elbows having a center line radius 1.5 to 2 times the diameter or width of the duct. The attenuation will be greater if the ratio is less than 1.5 and less when the ratio is greater than 2.

may be taken as 3 db higher than a similar single width fan operating under the same conditions of speed and pressure.

NATURAL ATTENUATION OF DUCT SYSTEM

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[4]. Attenuation values are given in Table 2. In general, this attenuation is so negligible except for long runs that it may be disregarded for all practical purposes.

Elbows and Transformations. Due to reflective interference, attenuation will take place at elbows and transformations. The magnitude of the attenuation will depend on the size and abruptness of the elbow or transformation as shown in Table 3.

When the area of a duct increases abruptly, an attenuation of noise level takes place in the duct. In duct design practice the total area of the branch ducts is greater than the supply duct. Similarly with outlets, the area of the outlet plus the area of the duct after the outlet is greater than the duct area before the outlet. Therefore in an outlet run, attenuation occurs in the duct as it passes each outlet. Table 4 gives the *db* reduction for various ratios of total branch duct and outlet area to supply duct area.

Grilles to Room. The large abrupt change in area between the grilles and the surfaces within a room results in an appreciable noise attenuation. This attenuation is a function of the total grille area (supply and return) and the total sound absorption of the room in *sabines*. (The sound absorption of a room in *sabines* is the summation of the products of each surface of the room measured in square feet multiplied by its corresponding absorption coefficient). The attenuation is given in Equation 3 as:

TABLE 4. ATTENUATION AT DUCT BRANCHES OR OUTLETS

RATIO: BRANCH DUCT + OUTLET AREA / SUPPLY DUCT AREA OR SUM OF BRANCH AREAS / SUPPLY DUCT AREA	ATTENUATION PER TRANSFORMATION, db
1.00	0.0
1.20	0.8
1.35	1.3
1.50	1.8
1.75	2.5
2.00	3.0

$$db\left(\begin{matrix}\text{Attenuation between}\\ \text{grilles and room}\end{matrix}\right) = 10\log_{10}\frac{\text{Total Room Absorption in Sabines}}{\text{Total Grille Area}} \qquad (3)$$

Values in Table 5 approximate the attenuation for various rates of air change, and general types of room surfaces.

DUCT SOUND ABSORBERS

The difference between the required sound attenuation and the natural attenuation is that which must be supplied by the proper sound treatment of the ducts.

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, several points should be considered.

TABLE 5. APPROXIMATE ATTENUATION BETWEEN GRILLES AND ROOM

OUTLET VELOCITY FPM	AIR CHANGE MIN.	LIVE ROOM[b] $\alpha^a = 0.05$ db	MEDIUM ROOM[c] $\alpha = 0.15$ db	DEAD ROOM[d] $\alpha = 0.25$ db
500	5	11	16	18
	10	14	19	21
	15	16	21	23
	20	17	22	24
750	5	13	18	20
	10	16	21	23
	15	18	23	25
	20	19	24	26
1000	5	14	19	21
	10	17	22	24
	15	19	24	26
	20	20	25	28
1250	5	15	20	22
	10	18	23	25
	15	20	25	27
	20	21	26	28

[a]Average absorption coefficient for the room.

[b]Live room-average absorption coefficient 0.05. Bare wood or concrete floor—hard plaster walls and ceiling—minimum of furniture.

[c]Medium room-average absorption coefficient 0.15. Carpeted floor, upholstered furniture, hard plaster walls and ceiling or bare room with acoustically treated ceiling.

[d]Dead room-average absorption coefficient 0.25. Heavy carpeted floor. Walls and ceiling acoustically treated. Upholstered furniture.

1. For the absorption of the low frequencies below 500 cycles per second the material should be at least 1 to 2 in. thick. Thin materials, particularly when mounted on hard solid surfaces, will absorb the high frequencies and reflect the low.

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[5], (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.

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 [6] that the reduction, in decibels per linear foot, of sound transmitted through 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 char-

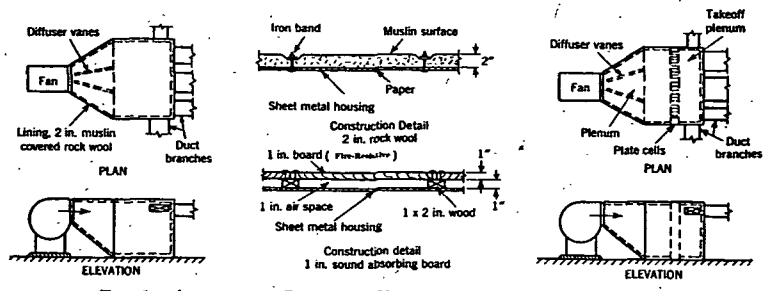


FIG. 1. ABSORPTION PLENUMS WITH AND WITHOUT SOUND CELLS

acteristics 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.

The noise reduction varies to a considerable extent with the frequency of the sound. In calculating noise reduction, consideration should be given both to the comparative efficiency of the duct lining material at different frequencies, and to the frequency distribution of the noise to be quieted. In the case of fan noise, it is recommended that calculations be based on the frequency 256 cycles, since most of the noise energy is in the region of this frequency. In quieting noise due to air turbulence and eddy currents, in which the high frequencies predominate, the frequency 1024 cycles should be used.

Since ventilating system noise contains many frequencies, an exception should be noted to the statement above that attenuation in decibels is directly proportional to length of duct. Most sound absorbent materials are more efficient at high frequencies than at low frequencies. In consequence, the attenuation in the first five or ten feet of lined duct will be greater because the high frequencies are being absorbed. Thereafter,

since low frequencies will be predominant, the over-all noise attenuation per foot will gradually be less.

Acoustic Impedance of Absorptive Materials

In the past five years considerable literature has collected describing methods of determining the acoustic impedance of sound absorbent materials and methods of utilizing this quantity for predicting the acoustics of rooms [7] and the attenuation of sound in ventilating ducts[8]. Acoustic impedance as a concept is derived from electrical circuit theory. The effect of the sound absorbent sound material upon incident sound waves is described in terms of a resistive and a reactive component which may be determined by specifically devised apparatus[9].

Generally speaking, however, the use of acoustic impedance theory involves rather elaborate mathematical calculations, and the improved accuracy obtained is largely off-set by variations in the materials themselves and in their methods of mounting. It has been difficult to measure the acoustic impedance of large areas of material mounted in a manner typical of standard construction. P. E. Sabine [7] concludes that the assumptions required by acoustic impedance theory make this method of calculation of no immediate practical advantage in the measurement of sound absorption coefficients.

Beranek [8] compares results of sound attenuation observed for rectangular ducts lined with absorbent material computed by acoustic impedance theory with data reported by H. J. Sabine[10], using the methods of this chapter. Beranek concludes that the conventional P/A relation is valid for rectangular ducts not too far from square. His analysis indicates that other cases require more exact theory. However, it seems questionable whether the improvement in accuracy offered by the impedance method overbalances the additional computation time required and out-weighs other sources of error such as variations between samples of material.

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. 1 will often prove the most economical arrangement. The absorption in the plenum may be approximated by Equation 4.

$$db\ (\text{Attenuation}) = 10 \log_{10} \frac{\text{Plenum Absorption in Sabines}}{\text{Area Fan Discharge}} \qquad (4)$$

The area of the plenum should be at least ten times as great as the fan discharge area. The plenum should be lined with 2 in. of muslin covered rock wool blanket or 1 in. sound absorbing board preferably nailed to wood strips on the inside of the plenum. With such a lining the plenum

TABLE 6. END REFLECTION OF PLATE ABSORBERS

PERCENTAGE FREE AREA OF ABSORBER	ATTENUATION, db
50	1
40	2
30	4
25	5
20	6

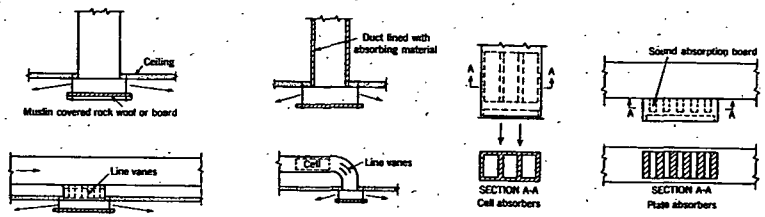


FIG. 2. OUTLET CELLS FOR PAN OUTLETS OR GRILLES

is particularly effective in reducing low frequency fan noise. The absorption of the plenum in sabines is the sum of the products of each interior area of the plenum measured in square feet multiplied by its corresponding absorption coefficient.

Plate Cells

One of the most economical methods of applying sound absorbent material from the standpoint of both labor and material is the plate cell. The plate cell consists of ½ or 1 in. sound absorbent board, spaced on 2, 3 or 4 in. centers. The attenuation due to the plate cell may be divided into two parts. There is reflection at each end due to the change in area and the absorption at the ends. Values for this attenuation are given in Table 6 which depend on the spacing. There is also attenuation due to absorption of sound within the passages of the cell, which depends on the length and the spacing. The attenuation within the cell for 1 in. board neglecting the end effect is given approximately by Equation 5.

$$R = \frac{10La^{1.4}}{S} \qquad (5)$$

where

R = attenuation, decibels.
L = linear length of duct, feet.
S = spacing in inches between plates up to 3 in.
a = absorption coefficient for the full thickness of the cell material. For typical value of a see Table 7.

An important objection to the plate cell is the increase in duct cross-sectional area required. Often on the fan discharge, particularly with unitary equipment, where a number of branch ducts take off, the plate cell may be installed with little or no difficulty.

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

TABLE 7. ATTENUATION FORMULAE FOR 1 IN. THICK TYPICAL DUCT LINING BOARD

FREQUENCY	ABSORPTION COEFFICIENT	ATTENUATION REDUCTION, db
256	0.37	3.0 L P/A
512	0.69	7.5 L P/A
1024	0.78	9.5 L P/A
2048	0.78	9.5 L P/A

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. 2.)

Duct Lining or Rectangular Cells

One series of experiments[10] made on a commonly used type of duct lining material (1 in. rock wool sheet) has shown that, subject to certain restrictions, the attenuation of single-frequency sounds may be expressed by the approximate Equation 6. This equation is accurate within plus or minus 10 per cent for duct sizes ranging from 9 x 9 in. to 18 x 18 in., for

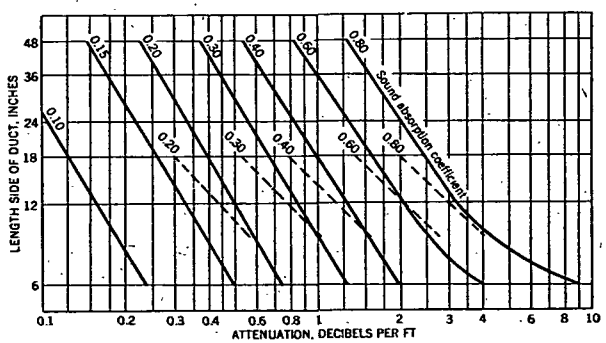


FIG. 3. SOUND ATTENUATION FOR VARIOUS ABSORBING DUCT LINERS

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.

$$R = 12.6\, L \frac{P}{A} a^{1.4} \qquad (6)$$

where

R = attenuation, decibels.
L = length of lined duct, feet.
P = perimeter of duct, inches.
A = cross-sectional area of duct, square inches.
a = absorption coefficient of lining.

In Table 7, the absorption coefficients at different frequencies of a material of the previously mentioned type are listed, together with the corresponding values for Equation 6.

Results of other experiments indicate, however, that Equation 6 may be in error when applied to other types of duct lining material and to duct sizes and shapes outside of the range specified. An empirically derived chart[11] representing the *average* experimental data on a number of different types of materials including the rock wool sheet mentioned as applicable to Equation 6 is shown in Fig. 3. Since individual materials vary, the curves in Fig. 3 are given only as representing the best available averages for duct sizes of square cross-sections from 6 x 6 in. to 48 x 48 in. As an illustration, the dotted lines in the chart show values calculated from Equation 6 which indicate that the slope for this particular material is somewhat different than from the average curves. The curves in Fig. 3,

as well as Equation 6, show that the attenuation in decibels is directly proportional to the length of duct lined, and that the larger the duct the greater will be the length which must be lined in order to obtain a given noise reduction.

If the length of duct from the main duct to a grille is shorter than the length of lining indicated by the calculations, this duct may be subdivided into smaller ducts, as shown in Fig. 4. The increase in noise reduction thus obtained may be calculated from Equation 7, provided the splitters are installed parallel to the long side of the duct:

$$R_s = R_o \frac{a + bn}{a + b} \tag{7}$$

where

R_s = reduction with splitters, decibels.
R_o = reduction in same length of duct, without splitters, decibels.
a = dimension of short side of duct, inches or feet.
b = dimension of long side of duct, inches or feet.
n = number of channels formed by splitters.

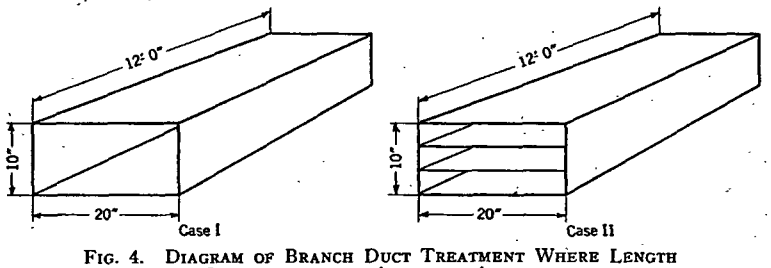


FIG. 4. DIAGRAM OF BRANCH DUCT TREATMENT WHERE LENGTH IS INSUFFICIENT FOR ADEQUATE ABSORPTION

Example 1. An air conditioning installation is to be installed in a small theater. Determine the necessary sound treatment for the air distribution system to provide a satisfactory noise level in the theater utilizing these conditions:

Fan tip speed 4000 fpm, total pressure 1.25 in	77 db
Acceptable room noise level (Table 1)	40 db
Required attenuation	37 db

Solution: Natural attenuation of supply duct.

Sheet metal duct 50 ft long 48 in. x 36 in. (Table 2) 50 x 0.01	0.5 db
Elbows, two size 48 in. x 36 in. (Table 3) 2 x 1	2.0 db
Attenuation grilles to theater air change 10 min (Table 5) outlet velocity 1000 fpm	22.0 db
Total natural attenuation	24.5 db

Difference between required and natural attenuation, 37 minus 24.5, is 12.5 db. This attenuation must be supplied by sound treatment in the duct, either in the form of duct lining or plate cells.

A similar analysis of the return duct system shows that 15 db attenuation are to be furnished by absorptive material. An inspection of the installation shows that the lining of the plenum on the suction side of the fan would prove the most economical, where it would secure the dual function of heat insulation and sound absorption.

Example 2. A 10 x 20 in. duct is connected to a private office space in a quiet location. Determine the length of lining necessary to attenuate average fan noise satisfactorily, using a lining material of a type to which Equation 6 applies, and having an absorption coefficient of 0.40 at 256 cycles. Assume that the duct is only 12 ft long as shown in Fig. 4, and that a 30 db reduction is required in this length.

Solution:

Case 1. (No splitters), From Equation 6,

$$R_o = 12.6 \times 12 \times \frac{60}{200} \times 0.40^{1.4} = 12.6 \text{ db}$$

Case 2. (Two splitters, three channels), From Equation 7,

$$R_s = 12.6 \times \frac{10 + (20 \times 3)}{10 + 20} = 29.6 \text{ db}$$

AIR SUPPLY OPENING NOISES

When air is introduced into a room through a grille or register at a constant velocity, sound energy is being introduced into the enclosure at a constant rate [12]. Due to partial reflection at the boundaries of the enclosure, the intensity of sound at any point in the space builds up to some maximum value. In a large room at a point remote from the source of sound (the supply opening) the intensity can be shown to be substantially proportional to the rate at which sound energy is generated and inversely proportional to the number of sound absorption units (sabines) in the

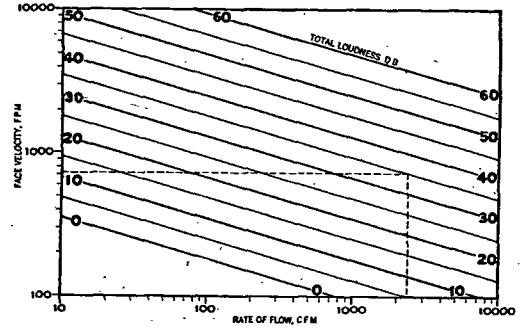


FIG. 5. AIR FLOW AND LOUDNESS CHART

room. It would thus appear that doubling the sound absorption of the room would halve the intensity and result in a noise level decrease of 3 db.

Grille noise is similar in character to fan vortex noise. Knowing the noise level at the face of a grille for a given grille blade setting, the noise will vary as given in Equation 8 where V is the velocity of the air through the grille.

$$db \text{ (change)} = 55 \log_{10}\left(\frac{V_2}{V_1}\right) \tag{8}$$

For a change in blade setting Equation 9 applies and in this case the total pressure is measured directly behind the face of the grille. For a typical air conditioning grille the noise level at the grille face may be approximately 48 db with a total pressure behind the grille of 0.1 in.

$$db \text{ (change)} = 27.5 \log_{10}\left[\frac{(\text{Total Pressure})_2}{(\text{Total Pressure})_1}\right] \tag{9}$$

The resultant room noise level can be approximated by Equation 10.

$$\text{Room Level} = \left[\begin{matrix}\text{Noise Level at Face} \\ \text{of Grille}\end{matrix}\right] - 10 \log_{10} \frac{\text{Total Room Absorption in Sabines}}{\text{Total Grille Area}} \tag{10}$$

Grille Selection

In practice the allowable total sound and the required air flow are usually known, and it is desired to determine the maximum allowable velocity. In comparing sound ratings of various grilles several factors must be known if the information is to be properly applied:

1. The threshold intensity on which the decibel ratings are based.
2. The distance from the grille at which data were taken.
3. If stated as loudness level versus velocity for a given grille, the *core area* (not nominal area) must be known.
4. The sound absorbing characteristics of the test room.
5. Whether or not corrected for test room loudness level: if not, the room level (without grille noise) must be known.
6. Methods used for recording data. (Characteristics of sound meter).

Since total loudness and air flow are both functions of velocity and area, the solution of the problem implies a trial and error method. It has been found possible to present these data with sufficient practical accuracy as a family of uniform curves, as illustrated in Fig. 5, which are based on these assumptions:

1. Threshold intensity = 10^{-16} watts per square centimeter[1].
2. Microphone location 5 ft from lower edge of supply opening on a line downward at 45 deg and in a plane bisecting the supply opening perpendicularly.
3. Where data are given as loudness level versus velocity, the rating is per square foot of core area.
4. The room is assumed to have 100 sabines absorption.
5. Plotted data are loudness levels of *supply openings only*, correction having been made for test room level.
6. Data taken with a direct reading sound-level meter with frequency weighing network intended to approximate the response of the human ear.

If the published ratings are in terms of decibels per square foot, correction must be made for area to secure the total sound level of supply openings of more or less than one square foot area from Equation 11.

$$\text{Decibel Addition} = 10 \log_{10} A \qquad (11)$$

where

A = core area, square feet.

With Fig. 5 it is possible to find directly the velocity in feet per minute which will give a predetermined total loudness at a predetermined rate of flow expressed in cubic feet per minute. The values used are arbitrarily chosen for the purpose of discussion and do not necessarily represent data referring to any particular design of air supply opening. A correction chart is shown in Fig. 6 for a room having a sound absorption other than 100 sabines.

Example 3. Determine the core area (see Chapter 40) of an air supply grille which will maintain a noise level of not more than 40 db in a room having 100 sabines of sound absorption, if an air volume of 2400 cfm is required to maintain the proper air conditioning.

Solution. Assuming a grille noise rating of at least 5 db below the noise level of the room, Fig. 5 shows that the limiting grille velocity for a total loudness of 35 db is about 725 fpm and the core area becomes fixed at 2400 ÷ 725 or 3.31 sq ft.

If the room absorption had been greater, the previously selected velocity of 725 fpm would be safe, since the loudness reduces. If the room absorption had been 200 sabines a correction of plus 1.3 should be made by reference to Fig. 6, and the permissible velocity becomes that corresponding to a total loudness of 36.3 or approximately 800 fpm.

If the room had been highly reflective with an absorption of less than 100, the correction would be much more important. For instance, for a room of 35 sabines a correction of minus 3 db should be made and the maximum velocity corresponding to the 32 db total loudness would be approximately 600 fpm.

Where more than one supply opening must be considered, the problem is more complicated. If a similar supply opening is added in a far corner of a highly absorbent room, the change in noise level at the 5 ft station at the first supply opening is small; however, if the room is small, or highly reverberant or both, the intensity at the 5 ft station may be almost doubled and the noise level increased nearly 3 db thereby. The simplest method of handling this problem is to treat the room as though all the air were being supplied by one supply opening. Thus, if two outlets, each supplying 1000 cfm are used, the value 2000 cfm should be used with

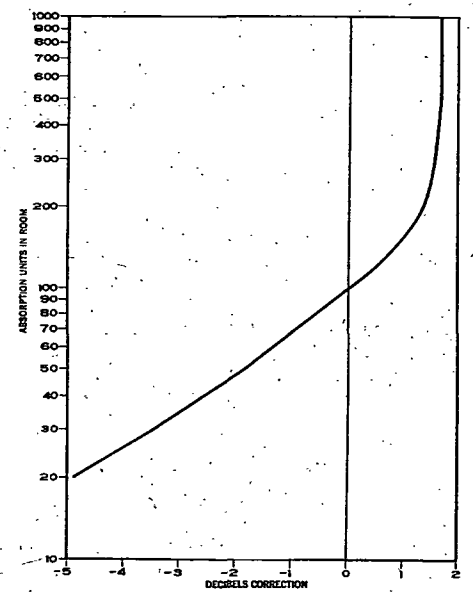


FIG. 6. ROOM ABSORPTION CORRECTION CHART

Fig. 5. Although this method may place an unwarranted limit on velocity when used in a large room, it is seldom that such a room has a noise level low enough to justify a more complicated though more exact procedure.

In general, return grilles are selected for velocities about half the supply velocity, and when this is done, they may be neglected in sound computations. However, if supply and return grilles are the same size, resulting in the same face velocity, they must be treated as two supply openings. That is, if 1000 cfm are supplied and exhausted through grilles of the same area, 2000 cfm must be used in the solution with Fig. 5.

CROSS TRANSMISSION BETWEEN ROOMS

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 sub-divided 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 lined some of the mechanical noise from the equipment room air may be transmitted through the wall of the duct, thus reaching the air stream and be 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 where there is 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 floor upon which it is supported.

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:

$$T = \frac{1}{1 - \left(\frac{f}{f_n}\right)^2} \tag{12}$$

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 12 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 is first accomplished. 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, but for the lower speeds as experienced with compressor work the higher ratios become uneconomical.

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 13.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{d}} \tag{13}$$

where

g = gravitational constant.

d = static deflection of supporting material.

f = frequency of the vibratory force.

f_n = natural frequency of the machine unit on its support (damping = 0).

By the use of Equation 13 a set of curves may be plotted as shown in Fig. 7. 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 two more curves EF, 3 times curve AB, and GH, 5 times curve AB, shows area $EGHF$ 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 is belt connected to the compressor. Total weight of the compressor and motor is 4500 lb.

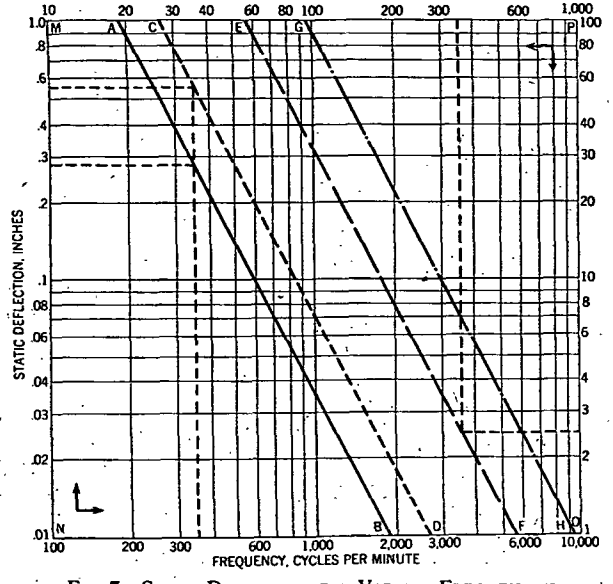


FIG. 7. STATIC DEFLECTION FOR VARIOUS FREQUENCIES

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. 7 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 the minimum 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	4,500 lb
Concrete foundation	9,000 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. 7, it is found that for 360 rpm the static deflection required for a ratio of f/f_n 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 1.75 in. For these conditions rubber in shear serves as a rather satisfactory material if protected from oil. For speeds higher than 1200 rpm cork specially made for vibration damping 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 only in two ways 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 Standards for Noise Measurement, Z24.2-1942, *American Standards Association.*

[2]—American Standards for Sound Level Meters for Measurement of Noise and Other Sounds, Z24.3-1944, *American Standards Association.*

[3]—Sound Insulation of Wall and Floor Constructions (*U. S. Department of Commerce, National Bureau of Standards,* Building Materials and Structures Report BMS17 and Supplement).

[4]—A.S.H.V.E. RESEARCH REPORT No. 1205—Determining Sound Attenuation in Air Conditioning Systems, by D. A. Wilbur and R. F. Simons (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 267).

[5]—For coefficients of commercial sound absorbent materials see Bulletin *Acoustical Materials Association,* 919 No. Michigan Ave., Chicago, Ill.

6—Sound Propagation in Ducts Lined with Absorbing Materials, by L. J. Sivian (*Journal Acoustical Society of America*, Vol. 9, 1937-38, pp. 135-140).

7—Specific Normal Impedance and Sound Absorption Coefficients of Material, by P. E. Sabine (*Journal Acoustical Society of America*, Vol. 12, pp. 317-323, January, 1941).

Application of Wave Theory of Room Acoustics to the Measurement of Acoustic Impedance, by C. M. Harris (*Journal Acoustical Society of America*, Vol. 17, pp. 35-46, July, 1945).

8—Sound Absorption in Rectangular Ducts, by L. L. Beranek (*Journal Acoustical Society of America*, Vol. 12, pp. 228-237, October, 1940).

The Transmission of Sound Inside Pipes, by Philip M. Morse (*Journal Acoustical Society of America*, Vol 11, pp. 205-210, October, 1939).

9—Precision Measurements of Acoustic Impedance by L. L. Beranek (*Journal Acoustical Society of America*, Vol. 12, pp. 3-14, July, 1940).

Notes on Acoustic Impedance Measurement, by H. J. Sabine (*Journal Acoustical Society of America*, Vol. 13, pp. 143-150, October, 1942).

10—The Absorption of Noise in Ventilating Ducts, by Hale J. Sabine (*Journal Acoustical Society of America*, Vol. 12, p. 53, 1940).

11—The Prediction of Noise Levels from Mechanical Equipment, by J. S. Parkinson (*Heating and Ventilating*, March, 1939, pp. 23-26).

Methods of Rating the Noise from Air Conditioning Equipment, by J. S. Parkinson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1940, p. 447).

12—The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 81).

CHAPTER 43

Central Systems for Comfort Air Conditioning

Types of Systems for Ventilating, Heating, Air Conditioning, Factors Involved in Use and Design of Systems, Design Procedure

THE purpose of this chapter is to present a discussion of types of central systems usually encountered, together with a discussion of the factors involved in use and design and an outline of design procedure.

Insofar as this chapter is concerned, a central system is defined as a field assembled apparatus, comprising such elements of equipment as are necessary to fulfill the purpose for which it is designed, and serving one or more conditioned spaces. A factory produced unit, including all the essential items of equipment may be employed as a central system. Unitary equipment is discussed in Chapter 36. Further, this chapter is confined to comfort air conditioning systems as such, and ventilating systems, warm air heating systems, together with central systems of a special nature, are excluded from the discussion.

This chapter assumes a knowledge of all the component parts of a system and the reader is referred specifically to other chapters covering design conditions and physiological principles, cooling and heating loads, spray equipment, heat transfer surface coils, cooling, dehumidification and dehydration, fans, air cleaning devices, refrigeration, air distribution and air duct design, automatic controls and instruments. In addition, the engineer should refer to the Code of Minimum Requirements for Comfort Air Conditioning [1] prepared by the joint committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers*, and to national, state or local codes that may apply.

CLASSIFICATION OF SYSTEMS

The generally accepted method of classifying systems is with regard to their function. A given type of system may be changed by the omission of certain of its functions or by the inclusion of others. As an example, a winter air conditioning system, by the omission of humidifying sprays, air cleaning devices, etc., will become a simple warm air heating system, and by further omissions will become a simple ventilating system. On the other hand, with the inclusion of a dehumidifier with its source of cooling, the system can become a year 'round air conditioning system. The three major types of systems are:

1. *Winter Air Conditioning Systems.* The function is to ventilate, heat and humidify in winter the spaces under consideration, and provide the desired degree of air motion and air cleanliness. The equipment required normally consists of a preheater, filters, humidifying sprays or air washer, reheater, fan, distributing ducts, and the necessary manual or automatic means of control. See Fig. 1.

2. *Summer Air Conditioning Systems.* The function is to ventilate, cool and dehumidify the spaces under consideration and to provide the desired degree of air motion and cleanliness. The normal complement of equipment includes filters, dehumidifier with its source of cooling, reheaters or by-pass if required, fan, distributing ducts, and the necessary manual or automatic means of control. See Fig. 2.

3. *Year 'Round Air Conditioning Systems.* The function is to ventilate, heat and humidify in winter and cool and dehumidify in summer the spaces under consideration, and to provide the desired degree of air motion and cleanliness. The equipment usually comprises preheater, filters, spray or surface dehumidifier, reheaters and by-pass if

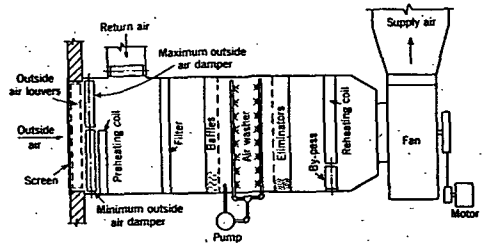


FIG. 1. CENTRAL SYSTEM FOR WINTER AIR CONDITIONING

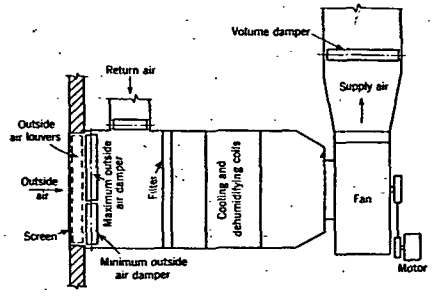


FIG. 2. CENTRAL SYSTEM FOR SUMMER AIR CONDITIONING

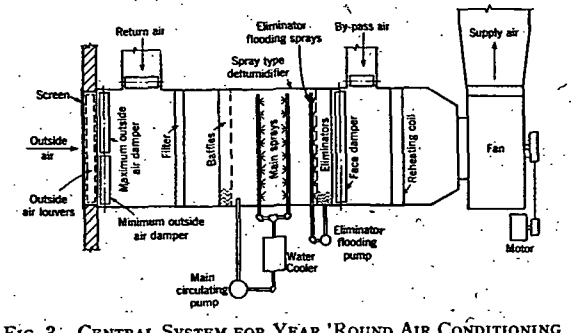


FIG. 3. CENTRAL SYSTEM FOR YEAR 'ROUND AIR CONDITIONING

required, fan, system of distributing ducts, and necessary means of manual or automatic control. See Fig. 3.

Items of equipment in the foregoing may, of course, be replaced with others fulfilling the same purpose. As an example, an absorption type dehydrator with an aftercooler may replace a dehumidifier.

Modifications

All of the general types of central systems may be modified in various ways. These modifications do not affect the functions of the system. In general, these consist of changes in or additions to the normal complement of equipment, or variations in arrangement and design of equipment and distributing ductwork in order to provide better control of conditions, greater flexibility, or improve the over-all economy and performance of the system. Some of these are applicable to certain systems only, while others are applicable to all types. The most commonly encountered modifications are:

1. Zoning.
   - *a.* Separate equipment.
   - *b.* Reheating or recooling. (See Figs. 4 and 5).
   - *c.* Multiple fans with individual by-pass or reheat. (See Figs. 6 and 7).
   - *d.* Volume control. (See Fig. 8).
   - *e.* Dual duct system. (See Fig. 9).
2. Induction units. (Low pressure type). (See Fig. 10).
3. Induction units. (High pressure type). (See Fig. 11).
4. Evaporative cooling.
5. Precooling.
6. Sensible cooling with dry coils.
7. The run-around system.

There are many other possible modifications and the special requirements of some installations may warrant consideration of these, but space limitations prevent a discussion of all. The modifications mentioned are discussed further in this chapter.

USE OF CENTRAL SYSTEMS

Several factors must be considered in deciding on whether or not to use a central system and in deciding on the type of central system and modifications required. These factors are:

1. Comparative effectiveness.
2. Characteristics and requirements of the load.
3. Space requirements.
4. Initial cost.
5. Operating costs and maintenance.

The comparative effectiveness of the type of system and modification plays a major part in the choice and is to a large extent affected by the other factors. One great advantage provided by the central system lies in its ability to diffuse odors and smoke which may occur in parts of the system, so that the outside air is determined by the average instead of the sum of the peak requirements. However, caution must be used where odors are apt to be objectionable even if greatly diluted. In such cases a positive exhaust to the outdoors or a separate treatment for the particular locality is recommended. The *averaging ability* both as to odors and

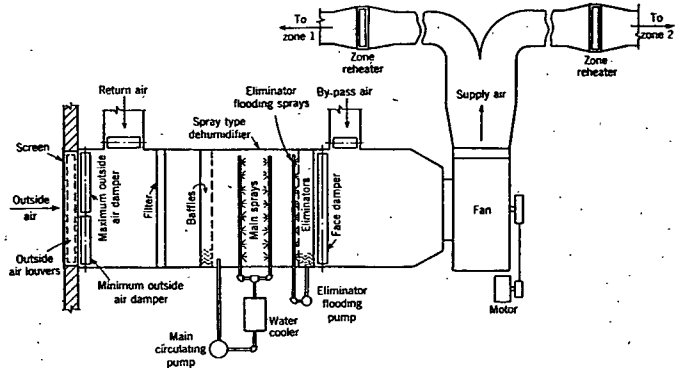


FIG. 4. CENTRAL SYSTEM WITH ZONING BY REHEATING

thermal effects of the system is one item to be considered in studying the comparative effectiveness, particularly where adequate zoning is to be provided. Full advantage of diversity and non-simultaneous peak load requirements can be taken in determining the dehumidified air quantity where adequate zoning is provided.

The characteristics and requirements of the load frequently are the deciding factors. Wide, non-simultaneous variations in load between spaces or parts of the same space indicate the necessity of zoning. Isolated spaces having a short time occupancy or brief load duration may be handled by units advantageously at times. The occurrence of simultaneous heating and cooling requirements in spaces having the same exposure or on the same zone presents a problem to be studied. Ability to maintain conditions during the intermediate seasons without the use of refrigeration must be considered at all times, particularly in those applications where a high internal load exists.

The matter of space requirements may rule out one type of system or another. The avoidance of the use of rentable and usable space for equipment and ductwork, in office buildings, stores, etc., is most im-

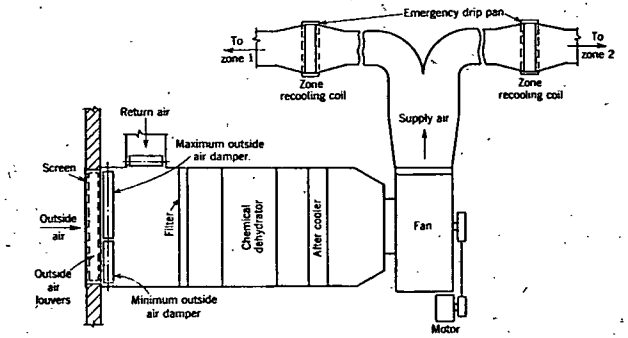


FIG. 5. SUMMER CENTRAL SYSTEM WITH ZONING BY RECOOLING

portant since the loss of revenue is directly chargeable to the operating cost of the system. Consideration of the spaces available with respect to the type of system and method of distribution used and their effect on over-all cost is essential.

Initial cost is given first consideration all too frequently. While elaborately designed installations are seldom justifiable, proper consideration must be given to operating costs and maintenance, performance required, and the life of the system. The increase in cost incurred by suitable zoning can be offset partially by reduced quantities of dehumidified air, and partially by reduced refrigeration requirements. In the end it may prove less expensive than an unsatisfactory single zone system. A small increase in initial cost to provide better access to equipment, better airflow and distribution, and better zoning, usually will pay for itself.

Operating costs often receive too little attention. Proper relationship between equipment selected and the load to be carried must be obtained.

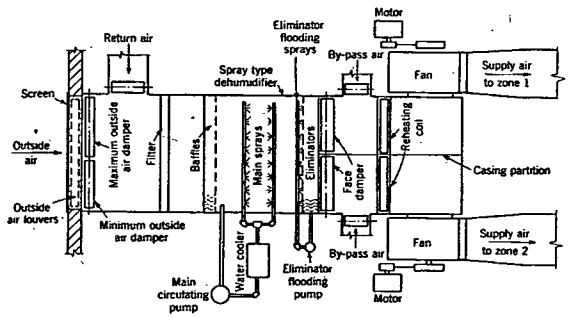


FIG. 6. CENTRAL SYSTEM USING MULTIPLE FANS WITH BY-PASSES AND REHEATING FOR ZONING

Cooling systems in general operate at maximum capacity less than 20 per cent of the time and heating systems operate under design conditions but a few days in the year. Good partial load performance is essential for low operation costs. Proper zoning, good arrangement and selection of equipment and type of system all tend to reduce operation costs. Central systems having the bulk of equipment in a central location properly arranged, and having only the equipment required for zoning distributed through the conditioned space or spaces, will have relatively low maintenance costs. Care must be used to provide ease of access to important equipment and those items requiring servicing.

DESIGN OF SYSTEMS

Various factors are to be considered in the design of a central system, some of which have been touched upon in the foregoing as a modification or economic factor, while others are a part of the normal design procedure. These, in the order of usual occurrence are discussed herewith. For practical purposes these factors cover *Year 'Round Air Conditioning Systems*. Those directly applicable to summer only or winter only systems are to be viewed accordingly.

Design Conditions

Physiological principles and design conditions are covered in Chapter 12. A detailed study of these is beyond the scope of this chapter other than certain recommendations with regard to their application.

Where extreme or unusual outdoor conditions prevail for long periods of time, such as in the tropics, at high altitudes, in regions of extremely low humidities, etc., due allowance must be made for the fact that the people have become acclimatized, to a certain degree at least, to these conditions and the inside conditions should be selected accordingly.

A change in the inside conditions from a set standard sometimes is warranted from an economic standpoint. It is possible at times to make substantial savings in initial and operating costs by maintaining a lower

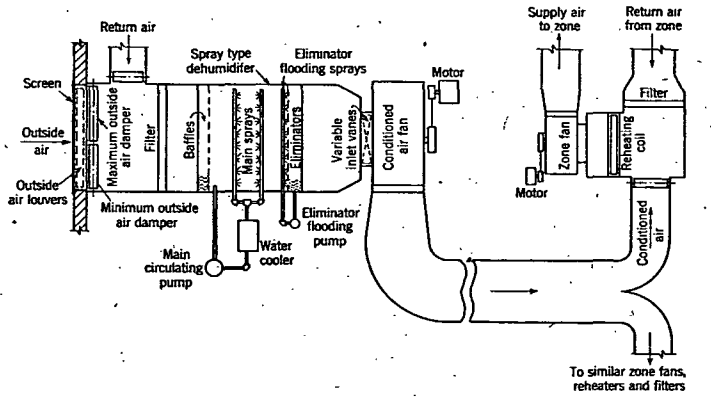


FIG. 7. CENTRAL SYSTEM WITH CENTRAL FAN AND CONDITIONER AND WITH INDIVIDUAL ZONE FANS

temperature and higher humidity in the conditioned space while maintaining the same effective temperature.

In winter, the matter of condensation on windows, walls, etc., is of extreme importance. Humidities low enough to avoid this should be maintained, and when it is necessary to carry the higher humidities, double glass should be used or suitable means of handling the condensation should be provided.

During intermediate seasons the use of refrigeration as a source of cooling may be undesirable from an operating cost standpoint depending on the local energy rate structure and demand charges. As a result the use of outdoor air, either directly or indirectly, as a source of cooling may be required and the system may have to operate on an unusual basis with regard to the temperatures and humidities that can be maintained. This situation should be carefully investigated. As a rule, provision should be made for introducing all outdoor air into the system during intermediate seasons.

With regard to outside conditions it must be noted that where systems are to operate at certain hours of the day only, the outside conditions that

occur during these times can be used provided the cumulative effect or heat gain lag is taken into consideration in the estimate.

Outdoor Air

Standards affecting the quantity of outdoor air have been established in Chapter 12. These standards relate the minimum amount of outside air to be introduced into the conditioned space for both the number of occupants and the type of occupancy, *i.e.*, people smoking, etc. This, of course, is the common-sense approach to the problem, but there are some cases, such as those spaces having a very low occupancy with regard to the cubical content where mustiness may develop unless a sufficient air change is provided. Where such a condition exists in a few spaces out of several which are being handled by a central system, the *averaging ability* of the central system may cope with the situation satisfactorily. This is

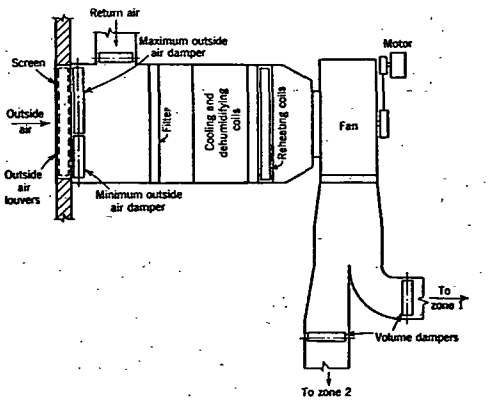


FIG. 8. CENTRAL SYSTEM WITH ZONING BY VOLUME CONTROL

due to the fact that the return air from the particular space is mixed thoroughly with the return air from all the other spaces and all the outside air supplied.

It should be noted that the minimum quantity of outdoor air 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.

Most summer and year 'round air conditioning systems should be so designed and arranged that a quantity of outside air at least equal to the quantity of cooled or dehumidified air can be taken in when desired. In the intermediate seasons and in the cooling season it is more economical to take all outside air into the dehumidifier when the outdoor air wet-bulb temperature is lower than the inside wet-bulb temperature to be maintained. Also, where using a spray type dehumidifier or air washer, whenever the outdoor wet-bulb temperature is below the apparatus dew-point temperature required, the system can be operated on an evaporative cooling basis, dispensing with the need for refrigeration though cooling may be required. Automatic controls are available for accomplishing this and on reasonably large systems are justifiable from an economic standpoint.

Cooling Load

The method of determining the cooling load for a conditioned space or spaces is outlined in Chapter 15. 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.

A large difference in the 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 gain forms a fair share of the cooling load, the times of their individual peaks are apt to be hours apart, and the peak load of one plus the off peak load of the other will be

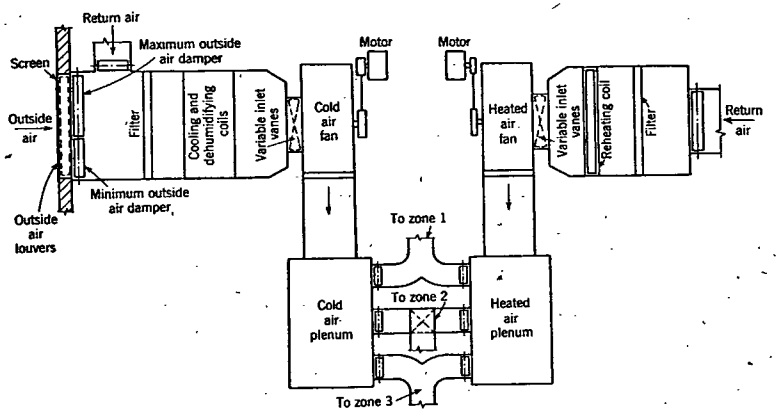


FIG. 9. CENTRAL SYSTEM USING DUAL DUCT METHOD OF ZONING

substantially less than their combined peak loads. Proper zoning will permit taking full advantage of this condition or similar conditions of non-simultaneous peaks and will result in a lower total load, reflecting itself in savings in equipment.

A factor, similar in effect and closely related to the non-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 must 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 on the first floor and basement. 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 building construction the effect of solar radiation is still apparent 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, having been warmed by the sun, may radiate heat long after the passing of the sun, thus requiring lower inside temperatures to offset the radiant energy.

Storage effect is usually present in some degree. Often it can be utilized to great advantage and it has more than once provided an unknown safety factor. If a space is kept below the design inside temperature for a period of time, the interior walls, floors, furniture and fixtures begin to assume the temperature of the space. Where the period of time is sufficient, the entire mass may reach the room temperature, rather than just the skin or surface of the item. Thus, when a space has been precooled below the

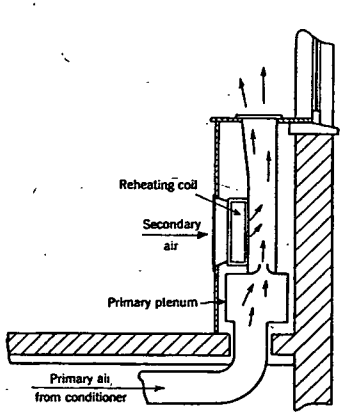


FIG. 10. INDUCTION UNIT (LOW PRESSURE TYPE)

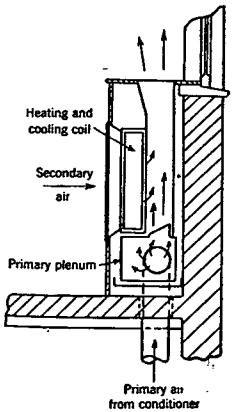


FIG. 11. INDUCTION UNIT (HIGH PRESSURE TYPE)

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 it is often a matter of experience rather than calculation.

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 41). 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.

Heating Load

Methods of calculating the heating load are shown in Chapter 14. Many of the factors outlined previously under Cooling Load, such as zoning, non-simultaneous peaks, and diversity, apply in the reverse manner due to the heating requirement instead of the cooling requirement. However, these factors enter into the heating load picture from the standpoint of control of inside conditions, over-all 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, 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, or after a week-end or holiday shut-down.

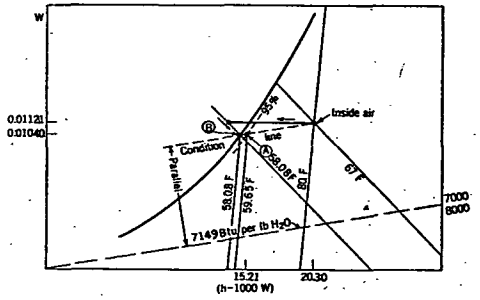


FIG. 12. DIAGRAM OF MOLLIER CHART ILLUSTRATING EXAMPLE 1

However, in normal operation during week-ends and holidays, buildings are usually kept at a *holding* temperature to prevent the freezing of services and conserve fuel. In many cases it requires less fuel to keep a building or space at a temperature of 50 to 65 F for some time than to shut the system down and then bring the temperature up again.

Apparatus Dew-Point

The method of locating the condition line for a given air conditioning problem has been explained in Chapter 3, Examples 12 and 13. Briefly, the method consists in estimating the net energy gain (or loss) per hour and the net moisture gain (or loss) per hour from data on location, exposure, construction, appliances, occupants, ventilation requirements, inside and outside design conditions. In computing the quantities of energy and moisture introduced and displaced by the ventilating air, only that portion of the ventilating air admitted directly to the conditioned space is considered. With this understanding, the ratio of the net energy gain (or loss) to the net moisture gain (or loss) determines the slope of the condition line through the state point of the inside air on the Mollier Chart.

The condition line may or may not cross the saturation curve. If it does, the intersection is called the *apparatus dew-point*, with application

to summer cooling in mind. Thus, if the air conditioning apparatus can be set to take inside air, process it, and return it to the conditioned space completely saturated at the apparatus dew-point, the cooling load requirements can be exactly met both as to removal of energy and simultaneous removal of moisture.

In actual practice with commercial apparatus, it is rarely possible to obtain complete saturation and there may be several degrees difference between the dry-bulb and wet-bulb temperatures of the air returned to the conditioned space. This causes no difficulty. In fact, the only special significance of the apparatus dew-point is that, when it exists, it provides

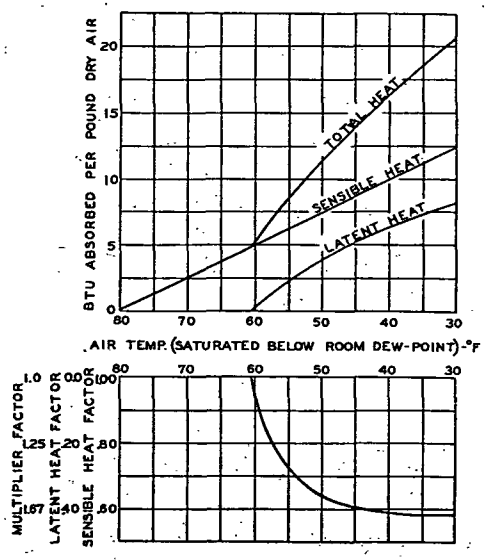


FIG. 13. HEAT AND MOISTURE ABSORBING POWER OF AIR INTRODUCED INTO A ROOM HELD AT 80 F AND 50 PER CENT RELATIVE HUMIDITY (60.1 F ROOM DEW-POINT) AND VARIOUS SENSIBLE HEAT FACTORS, LATENT HEAT FACTORS, AND MULTIPLIER FACTORS CORRESPONDING TO THE SUPPLY AIR CONDITIONS

a convenient control point at which to regulate the operation of the apparatus, provided complete saturation is attainable.

In order to illustrate the effect of incomplete saturation in the conditioning apparatus, consider the cooling load problem of Chapter 3, *Example 12*. In this problem, 114,510 Btu of energy and 16.018 lb of moisture per hour are to be removed simultaneously. The slope of the condition line is determined by the ratio $q = 114{,}510 \div 16.018 = 7149$ Btu per pound of water and the apparatus dew-point is 58.08 F, as shown in Fig. 12. (Point *B*).

But suppose that the air delivered by the apparatus is only 95 per cent saturated. Then the temperature at which the condition line crosses the 95 per cent saturation curve is the proper temperature at which to regulate the apparatus. This temperature is easily found to be 59.65 F dry-bulb temperature (Point *A* in Fig. 12). Of course, a somewhat larger quantity of air will have to be recirculated, namely $114{,}510 \div (31.514 - 25.631) = 19{,}460$ lb of dry air per hour, where the enthalpy of the air at 95 per cent saturation on the condition line is 25.631 Btu per pound of dry air and $h = 31.514$ for the

inside air. Only 18,050 lb of dry air had to be recirculated per hour with complete saturation. (Example 13, Chapter 3.)

The supply air must have the capacity to absorb the room sensible heat gain and the room moisture (latent heat) gain simultaneously and in the correct ratio. The top portion of Fig. 13 shows the relationship of sensible heat and latent heat absorbing power of air supplied to an 80 F room when room relative humidity is 50 per cent[2]. On some psychrometric charts the line connecting the room condition and the apparatus dew-point on the saturation line is equivalent to the condition line on the previously mentioned Mollier Diagram.

Most practicing engineers prefer other methods of calculating apparatus dew-point which use one of several forms to express the ratio of room sensible heat gain to room moisture (latent heat) gain. The room latent gain is the heat equivalent, in Btu, of the amount of moisture liberated within the space or introduced to the space. This heat equivalent is approximately equal to the latent heat of vaporization at the apparatus dew-point temperature. Since this is variable, an average value of 1050 Btu per pound of moisture gain may be used without serious error for normal air conditioning calculations, when the room and apparatus dew-point temperatures are above 32 F. Since the load calculation, particularly the room latent heat gain, is only approximate, it is believed that these approximate methods for calculation of apparatus dew-point are sufficiently accurate for most commercial applications. Several methods are commonly used to express the relation of room sensible heat gain to room latent heat gain and these are described in the following:

$$\text{Sensible Heat Factor, (SHF).} \quad \text{SHF} = \frac{H_s}{H_s + H_l} \tag{1}$$

$$\text{Latent Heat Factor, (LHF).} \quad \text{LHF} = \frac{H_l}{H_s + H_l} \tag{2}$$

$$\text{Multiplier Factor, (MF).} \quad \text{MF} = \frac{H_s + H_l}{H_s} \tag{3}$$

where

H_s = room sensible heat gain, Btu per hour.
H_l = room latent heat gain, Btu per hour.

These equations are equivalent to one another as shown by the following:

$$\text{LHF} = 1 - \text{SHF} \tag{4}$$

and

$$\text{MF} = \frac{1}{\text{SHF}} \tag{5}$$

The sensible heat gain in the room or space must be absorbed by the supply air in order to maintain a given room temperature. Then the following heat balance for saturated supply air may be written as:

$$H_s = (t_r - t_s)\, Q_m \times \rho \times c_p \times 60 \tag{6}$$

where

H_s = room sensible heat gain, Btu per hour.
t_r = room temperature, dry-bulb, Fahrenheit degrees.
t_s = apparatus dew-point temperature, Fahrenheit degrees.

Q_m = quantity of supply air, cubic feet per minute.
ρ = density of supply air, pounds per cubic foot.
c_p = specific heat of air, at constant pressure, Btu per (Fahrenheit degree) (pound).

Similarly the latent heat (moisture) gain in the room must be absorbed by the lower moisture content of the supply air in order to maintain a given relative humidity, and writing a heat balance for saturated supply air:

$$H_l = (W_r - W_s)\, Q_m \times \rho \times h_v \times 60 \tag{7}$$

where

H_l = room latent heat gain, Btu per hour.
W_r = moisture content of room air, pounds per pound of dry air.
W_s = moisture content of air, pounds per pound of dry air at apparatus dew-point.
h_v = heat of vaporization at apparatus dew-point temperature.

Equation 1 shows how to determine the sensible heat factor of the cooling load. The sensible heat factor of the supply air must equal the sensible heat factor of the room load in order to maintain design conditions.

If a dehumidifier that will produce a saturated condition of leaving air is used then Equations 6 and 7 may be solved for H_s and H_l and substituted in Equation 1. This for practical purposes can be shown to be equal to:

$$\text{SHF} = \frac{0.24\,(t_r - t_s)}{h_r - h_s} \tag{8}$$

where

0.24 = specific heat of air, Btu per (pound) (Fahrenheit degree).
t_r = room temperature, dry-bulb, Fahrenheit degrees.
t_s = apparatus dew-point, Fahrenheit degrees.
h_r = enthalpy of air at room conditions, room wet-bulb, Btu per pound of dry air.
h_s = enthalpy of air at apparatus dew-point, Btu per pound of dry air.

Practically, the apparatus dew-point temperature may be defined, for a cooling coil, as the average coil surface temperature; and for a spray dehumidifier of commercial design, (where the saturation efficiency approaches 95 per cent to 100 per cent) as the dew-point of the air leaving the dehumidifier.

Having estimated the room sensible heat factor the problem is reduced to the solving of Equation 8. Since the relation between h_s and t_s is only available in tabular or graphical form, a cut and try or graphical solution is required. Fig. 14 has been calculated from Equation 8 and, of course, can be used only for one room dry-bulb temperature, 80 F. The lower portion of Fig. 13 also gives the apparatus dew-point required for various sensible heat factors. Note that if air is supplied at any temperature above the room dew-point, only sensible heat is absorbed (SHF = 1.0); and that below this point moisture is absorbed in a very rapidly increasing proportion to sensible heat. A point is finally reached where the proportion of moisture absorbed to sensible heat absorbed is practically constant and a lower apparatus dew-point will not remove a greater proportion of moisture. A room condition can be maintained if the load sensible heat factor lies on the curve corresponding to the design room humidity in Fig. 14. For example, if the load SHF was 0.55 for a design room condition of 80 F dry-bulb and 50 per cent relative humidity, it would not be possible to maintain this condition unless reheat

is used to raise the sensible heat factor at least to 0.59. Reheat should be added in the room, in the supply air duct, or in the dehumidifier after the cooling coil or spray section. Reheat is usually supplied by steam coils or by a hot gas interchanger using the hot discharge gas from the refrigerating machine.

Example 1. From a cooling load analysis of the room as determined in Example 12, Chapter 3, the net sensible and latent heat gain of 114,510 Btu per hour would be called room total heat; the net moisture gain of 16.018 lb per hour, multiplied by the approximate value of 1050 Btu per pound of moisture, would be called the room latent heat; namely, 16,820 Btu per hour; and 114,510 − 16,820 = 97,690 Btu per hour which is

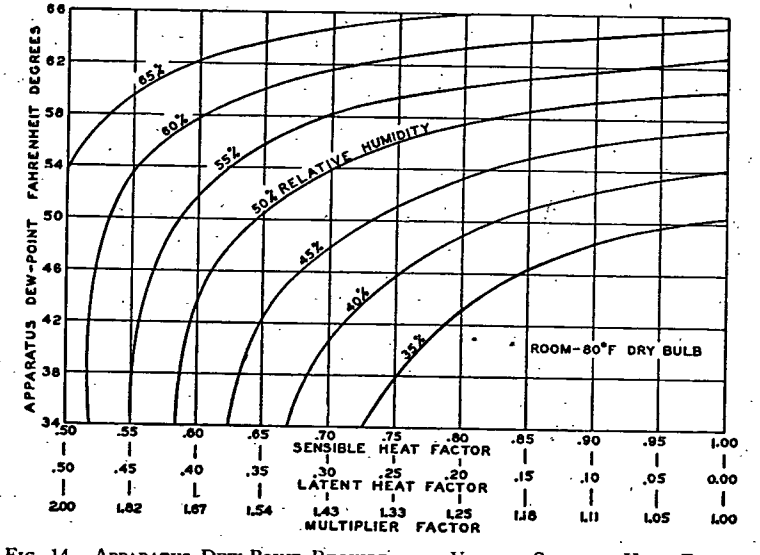


FIG. 14. APPARATUS DEW-POINT REQUIRED FOR VARIOUS SENSIBLE HEAT FACTORS (SHF), LATENT HEAT FACTORS (LHF), AND MULTIPLIER FACTORS (MF) CORRESPONDING TO VARIOUS ROOM RELATIVE HUMIDITIES, FOR ROOM DRY-BULB TEMPERATURE OF 80 F

the room sensible heat. Determine the apparatus dew-point if the room conditions to be maintained are 80 F dry-bulb and 67 F wet-bulb temperature (51.1 per cent relative humidity).

Solution. From Equation 1 the sensible heat factor is SHF = 97,690 ÷ 114,510 = 0.853. Solving Equation 8 by cut and try or by plotting this equation on a curve similar to that in Fig. 13, the apparatus dew-point is 58.6 F. The apparatus dew-point is 58.08 F as determined in Example 12 of Chapter 3. This should be sufficiently accurate for practical applications.

In actual practice, with commercial apparatus, complete saturation is seldom obtained. Four-row finned coils contact approximately 80 per cent of the air, whereas six-row finned coils contact approximately 95 per cent of the air. With 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 1 deg. A spray dehumidifier having sufficient length of spray chamber and density of spray, together with proper arrangement of nozzles, may approach absolute saturation very closely.

The slope of the condition line, or 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 rate of supply air must be used to satisfy the room load if the coil does not contact 100 per cent of the air. For a given room load, the same apparatus dew-point will be required whether the coil contacts all the air or only part of the air.

From the point of view of satisfying the given cooling load requirements, the result of air passing through the apparatus without being contacted produces two effects: (1) It increases the air quantity to be passed through the dehumidifier. Thus, if 20 per cent of the air passing through the coil is contacted, then 25 per cent (0.20 ÷ 0.80 × 100) more air must be used than would be necessary if all of it were contacted. This is readily seen by examining the heat balance in Equations 6 and 7. (2) It may change the room load which in turn may change the sensible heat factor. If only return air is passed through the dehumidifier or if only room air is by-passed, the room load would not change, but if some outside air is passed through the coil, the room sensible heat gain and room latent heat gain will be changed due to the addition of untreated outside air to the room, which changes the sensible heat factor. When a load calculation is made, it is necessary to know the per cent of air contacted in the cooling coil and calculation made accordingly.

If the ventilation air is drawn through the cooling coil before it goes into the room, only that portion of the air not contacting the coil surface 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 outside air that the temperature difference between room and outside air and moisture content between room and outside air should be used and not the difference between outside air and apparatus dew-point, since the rise from the apparatus dew-point to room conditions is charged against the dehumidifier as the cooling and dehumidifying load.

In winter, room humidities in excess of 30 per cent are seldom required and a low saturating efficiency may be desirable, or even necessary, where the full summer air circulation through the conditioner is maintained. With a spray type dehumidifier the main sprays may be shut off and an eliminator flooding pump provided which may give sufficient saturation. In other cases, such as where 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 the 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. This precaution should be observed also where open water heaters or ejector water heaters are used.

Air Quantity and Effective Temperature Difference

The difference between the room air temperature and the supply air temperature at the outlet to the room is known as the *effective temperature difference*. In the theoretical case of a dehumidifier having a 100 per cent saturating efficiency and where this air is delivered directly to the room without temperature increases due to heat gain, then the *effective temperature difference* 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 was modified in direct proportion, and that if room air passed through untreated no effect on the heat balance resulted. 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 *effective temperature difference*. Thus if the difference between the room temperature and the apparatus dew-point indicates that 10,000 cfm at 30F below room temperature will be required to hold conditions, that quantity can be passed through the dehumidifier and cooled to 30 F below the room, then mixed with 10,000 cfm of room air resulting in a supply air quantity of 20,000 cfm and an *effective temperature difference* of 15 F instead of 30 F. Supply air outlets and grilles that have a high induction ratio (that is, a large amount of room air is mixed with the air leaving the outlet within a short distance of the outlet through the induction effect of the air stream) are available as well as induction units. A proper selection of outlets or units 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. Lower *effective temperature difference* may be required for this reason. While the use of a high *effective temperature difference* results in a saving in initial cost of fans and ducts and in the operating cost of fans, this difference should be carefully considered. If the sensible heat load of a space is subjected to substantial variations, lower *effective temperature differences* should be considered, since systems employing a *low effective temperature difference* will be less exacting in control requirements.

Assume that a space has a sensible heat load so that 10,000 cfm of air supplied at a 30 F *effective temperature difference* would be required to maintain a room temperature of 80 F. If the load is suddenly reduced 50 per cent with the air supplied at the same temperature, the resultant room temperature would become 65 F. On the other hand, if 20,000 cfm were supplied at an *effective temperature difference* of 15 F and the load suddenly reduced 50 per cent the resultant room temperature would be 72.5 F. This, while a rather extreme example, indicates the less exacting demand on the controls brought about by the use of the lower *effective temperature difference*. Of even greater importance is the case where two or more spaces are controlled from an average condition such as by a thermostat located in the return air stream. From the previous example, it

can be seen that a large variation in the load in one of the spaces will not reflect itself in such a large change in room temperature. Thus in the long run the larger *effective temperature difference* may not be the most economical.

The analysis in the foregoing applies largely to summer air conditioning. The same analysis will apply to some extent in winter. The mathematical relationship is revised due to a heating requirement rather than cooling. Present practice indicates a high temperature difference in winter in comparison to that for summer. It is due to the fact that the heat losses in winter in Btu per hour far exceed the summer heat gains in Btu per hour, and particularly to the fact that in winter, reheating the air to produce desired room conditions is not reflected as a load on the system as it is in the summer, but merely accomplishes the necessary work.

In line with the latter, reduction of air quantity by slowing down the fans for the winter season and increasing the temperature difference often is feasible, creating a saving in fan horsepower at no expense to the final heat balance, providing the air distribution is not seriously affected.

Extremes should be avoided in all cases. For summer air conditioning low supply air temperatures will result in larger heat gains to the air passing through the ducts, poor control, etc. Too high a supply air temperature may result in excessive initial and operating costs. Suggested limits for the *effective temperature difference* are from 12 to 20 F, 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 mixture is insured, while too low supply air temperatures may cause drafts, high operating costs, etc. Suggested limits are from 15 to 35 F. There can be no set rule and each case should be judged according to its particular requirements.

By-Pass

The by-pass, in its accepted form, consists of an arrangement of ducts and apparatus connections with the necessary dampers which will permit air to pass around the dehumidifier or conditioner without being treated. It has two functions which may be employed separately or simultaneously.

The first of these is to provide a means of temperature control at a substantially constant total air quantity. If in summer, the load within the conditioned space is reduced and the temperature begins to fall, this drop in temperature can be offset by passing some of the air around the conditioner instead of through it, while the total quantity of air in circulation remains unchanged. When used for this purpose, it is termed an *adjustable* or *automatic* by-pass. The second function is to maintain a lower effective temperature difference between the air supplied to the room and the room temperature than could be obtained if air at the apparatus dew-point were supplied, and when so used is called a *fixed by-pass*. As discussed previously, if return air from the conditioned space is passed around the conditioner and mixed with the conditioned air, the effect on the heat balance is the same as if the air were removed from the space and immediately reintroduced. This is not strictly true, due to the fact that when ducts pass through unconditioned spaces, there is a heat gain by this air, and an additional gain is imposed by the heat of compression of the circulating fan in moving the air against the resistance of the system. However, the heat gain, where the by-pass is used to lower the effective temperature difference, usually favors its use due to the fact

that the increased volume and the resultant higher temperature of the mixture of conditioned air and room air may show a lower net duct heat gain with a smaller temperature increase per unit of volume of supply air. The advantages thus obtained may offset the increased fan power.

The adjustable or automatic by-pass can be made to serve the purpose of the fixed by-pass by providing a stop on the by-pass damper so that it cannot close completely. In some cases this stop is unnecessary since commercial dampers will permit an air quantity of 10 to 20 per cent of the conditioner capacity, depending on the resistance of the conditioner, to leak through even when fully closed.

Where a reduction in room sensible heat is not accompanied by a reduction in room latent heat, the by-pass is to be used with care. If the dew-point temperature of the air leaving the dehumidifier is controlled at a fixed value the reduced quantity passed through the conditioner may be sufficient to handle the sensible heat load but insufficient to handle the latent heat load, resulting in humidities that are too high. If the dew-point is not controlled, as when cold water is supplied at a constant temperature or where direct expansion cooling coils are used, the reduced loading on the conditioner brought about by the reduced air quantity will result in a lower dew-point temperature which often is entirely adequate. This condition should be investigated in each case.

The by-passing of outdoor air is to be avoided in general. While the sensible heat requirements of the space may be such that the by-passing of high temperature outdoor air will aid in controlling room temperature, high moisture content air introduced to the space may raise the humidity to an objectionable amount. If return ducts from which the by-pass air is taken run through unconditioned spaces and there is an inward leakage of outdoor or moist air, the effect, in a lesser degree, is that of by-passing outdoor air. Therefore the location of such return ducts and the points from which such air is taken are of importance. Exceptions to this are where the moisture content of the outdoor air is lower than that of the room, and where the sensible heat to latent ratio increases at partial loads.

The foregoing applies largely to summer air conditioning. In winter the by-pass usually is kept closed and room temperature control obtained by means of regulating the amount of heat supplied to the air. In some instances when the by-pass is located after the reheaters, the operation of the by-pass damper may be reversed and the by-pass still used as a means of control with the reheaters full on or operated in sequence with the by-pass. In other cases the reheater may be located in the by-pass as described further.

The principle of the by-pass may be applied to items of equipment other than conditioners, such as humidifiers, dehydrators, heaters, etc., in much the same manner. Where a heater has a capacity such that the temperature rise of the air is higher than desired, a smaller heater may be used and a portion of the air by-passed around the heater. Throttling of the sprays in a humidifier is, in effect, a by-pass since it increases the portion of air passing through without being contacted.

Reheating

Reheating the supply air is necessary in winter where this air is used to offset heat losses. Tempering of the supply air (merely reheating to a lesser degree) is required where other means of heating, such as direct radiation or panel heating, are used to carry the main heating load. Supply air at the required apparatus dew-point would add to the load to be carried

by the direct radiation and in addition may create a movement of cold air that while desirable in summer may be undesirable in winter. Modulation of the amount of steam supplied to the coils can be used to control temperature, or air can be by-passed around these heating or tempering coils.

Since reheating or tempering coils are required for winter and year 'round air conditioning, their use as a means of summer as well as winter temperature control is indicated. Where heat is available in summer, as the room sensible heat falls off, the low temperature of the supply air can be raised by means of these coils to maintain the desired room temperature while still providing adequate air at the proper dew-point.

Reheating presents an excellent method of accurate temperature control since the quantity of air passed through the conditioner is not changed as in the case of the by-pass, and since the distribution and circulation are not affected as when the volume of supply is reduced. However, reheating in summer has one disadvantage. It has the effect of maintaining a constant internal sensible heat load on the system. This means that when an effective temperature difference of 15 F is being maintained at the maximum room sensible heat load, this temperature difference must be reduced by means of heating to 7.5 F at half of the sensible heat load. The room sensible heat usually is about 35 to 45 per cent of the total cooling load and thus the penalty imposed on the refrigeration cycle is not extremely large. When the outdoor wet-bulb temperature is less than the desired room wet-bulb temperature, all outdoor air should be passed through the conditioner and under these circumstances reheating does not impose a load on the refrigeration cycle since the heated air is not returned to the conditioner. Further, where the volume of outdoor air introduced to the system, which for all practical purposes is wasted after its work has been done, is sufficiently large in relation to the amount of reheat used (that is, if the heat required does not exceed that necessary to raise the outdoor air only from dew-point to room temperature) the use of reheat does not impose a load on the refrigeration cycle. This, of course, assumes that there is no economic penalty involved in the use of heat itself.

One of the best applications of reheating for summer purposes is in combination with other means of temperature control for the purpose of leveling off accentuated demands for temperature control. As an example, the by-pass can be applied to a certain extent, or the volume of supply air throttled to a limited degree, or both of these used in sequence, and reheating can be used as the final step in temperature control.

In the foregoing it has been assumed that summer reheating is derived from an extraneous source of heat such as steam, electric heaters or hot water. The economics of reheating as outlined may be improved by the use of sources of heat that are available within the equipment used.

Where certain types of refrigeration cycles are used, auxiliary refrigerant condensing coils can be placed in the air stream. At partial loads these can be used as additional refrigerant condensing surface and improve the performance of the refrigeration plant. This is generally known as hot gas reheating. Less effective is the use of coils in the air stream through which liquid refrigerant from the condenser is passed before being delivered to the evaporator, sub-cooling the liquid refrigerant and improving the performance of the refrigeration cycle. The use of refrigeration condenser water as a source of reheating provides definite economies. If the condenser water is passed through coils in the air stream before being used for refrigerant condensing purposes, the lowering

of the condenser water temperature accomplished by reheating the air will result in savings in the refrigeration plant power consumption and increased refrigeration capacity. The latter two methods are at some disadvantage in that the amount of reheat available decreases as the need for reheating increases, particularly where evaporative condensers or cooling towers are used.

Zoning

Zoning consists of an arrangement of equipment or a division of equipment into sections that will permit individual control of the temperature and humidity of those spaces or groups of spaces that do not have simultaneous variations in sensible or latent heat load. The equipment is so arranged or divided that air can be supplied to spaces or groups of spaces in accordance with the individual load requirements of that space or group of spaces.

Solar heat gain is one of the major causes of the zoning requirement since its effect and amount vary with the season, time of day and exposure. Other sources of heat gain, subject to variations, such as large changes in the number of occupants in one space with a constant occupancy in another indicate the necessity of zoning. Some of the various methods of zoning are:

1. Separate equipment.
2. Reheating or recooling. (See Figs. 4 and 5.)
3. Multiple fans with individual by-pass. (See Figs. 6 and 7.)
4. Volume control. (See Fig. 8.)
5. Dual duct system. (See Fig. 9.)
6. Combinations of the above methods.

Zoning by separate equipment represents the extreme in zoning. Individual conditioners, fans, heaters, controls, distributing duct work, etc., are provided for each zone and are separate from those of other zones. Each assembly of equipment is arranged to operate at full or partial load according to the requirements of that zone. In extreme cases an individual refrigeration plant may be provided for each zone. In general, this method of zoning is uneconomical since each piece of equipment is large enough to handle the full load requirements of that zone and no advantage can be taken of the fact that while one zone is at its full load others may be operating at considerably less than full capacity. This applies both to the initial cost of a system and the operating cost. There are, however, many cases where this method of zoning when used to a limited degree or combined with other methods is most desirable.

Zoning by reheat is one of the more simple methods of approaching the problem of differing variations in load. Reheating has been discussed in the foregoing. Heating coils located in the distributing duct work or apparatus connections which supply only those spaces having substantially the same load variations will, by adding heat when the cooling load falls off (or the reverse in the case of a heating load), maintain the desired temperature conditions. It has been shown previously that where the amount of reheating required is not excessive and that heat for this purpose is available from an economic standpoint this method of zoning may be entirely practical and even highly desirable.

Zoning by recooling is literally the reverse of zoning by reheating. It is not used often but there are many cases where it is desirable. Its most practical application is limited to a sensible heat removal function where the latent heat requirements are handled by another source such as a

dehumidifier or dehydrator, and where the recooling equipment (usually cold water cooling coils) can be utilized for reheating (usually with hot water) in the heating season. In using this method of zoning, when the cooling load of a given space is reduced, the cooling effect produced by the recooler is also reduced.

The use of *multiple fans with individual by-passes* presents a simple and sometimes inexpensive method of zoning. Two or more fans may be arranged so that each draws its treated air from the same conditioner. The connection between the conditioner and fans is divided or partitioned in such a manner that a by-pass connection can be made to each fan. See Fig. 6. In this way the amount of by-pass air can be regulated according to the requirements of the zone served by that fan. In this application a face damper must be used for each segment of the conditioner, arranged to close as the individual by-passes open or an unbalanced system may result. A much better adaptation of this principle, though slightly higher in initial cost, is obtained by using a single conditioner or dehumidifier, having a central conditioned air fan delivering the treated air to the necessary zone fans where the conditioned air is mixed with return air according to the requirements of the zone. See Fig. 7. With this method of zoning, the zone fans are provided with casings to which both return air and conditioned air are delivered, their proportions being regulated by dampers working in opposite directions. If conditioned air is delivered to the casing at slight positive pressure the return air damper may be omitted. Reheating can be effectively combined with either of these for year 'round use or for winter use only. The reheater is usually located in the air stream to the zone fan. Where a central conditioned air fan is used to deliver conditioned air to a number of zones, a static pressure regulator controlling a volume damper or inlet vanes on this fan should be installed to prevent unbalancing the system, when one or more zone fan conditioned air dampers are throttling. This is one of the better methods of zoning and is particularly effective when used in combination with reheating for winter or year 'round conditioning as previously outlined.

Volume control is the least expensive and most frequently used method of zoning. It is usually obtained by placing a throttling damper in the supply duct feeding a particular zone and operating the damper to restrict flow of air as the heating or cooling load is reduced. The damper may be operated manually or automatically. Volume control, however, has two serious disadvantages. The first of these is that any large reduction in the quantity of air supplied may impair the ventilation. The second is that a large reduction of the air supply may entirely upset the distribution from the room outlets causing dead pockets, stratification, lack of air motion, or the reverse—undesirable drafts. Where the degree of volume control is large or the system extensive and where volume control is applied to one portion of a system and not to another, the use of a static pressure regulator controlling a fan discharge damper or fan inlet vanes is indicated. Its best application is in combination with some other method of zoning such as reheat or by-pass where it is used as one step in a control sequence, and the reduction in volume limited to a proper amount.

The *dual duct method* of zoning, sometimes referred to as the schoolhouse system, can be successfully applied to comfort air conditioning though certain precautions must be observed. Essentially this method employs a source of warm air and a source of cold air both of which are delivered to a common point where either or a mixture of both are

delivered to a particular zone according to the requirements of that zone. See Fig. 9. Several variations of this method are possible, some of which do not employ dual ducts as denoted by the name but which utilize the principle. An example of this is found in a blow-through system where the fan is located on the entering side of the conditioner and the conditioned air passes from the conditioner into a plenum from which distributing ducts for the various zones are taken. A by-pass connection around the conditioner from the fan to the zone duct is made and dampers provided so that conditioned or untreated air, or a mixture of both is passed into the zone supply duct. In this method of zoning and in most of the variations of this method, the matter of by-passing untreated or outdoor air presents itself. This has been discussed earlier in this chapter. If a return air fan is used and the by-pass connection made from the return fan to the zone supply duct, then the by-passing of outdoor air does not need to be considered. Complication of ducts and connections should be avoided since it may result in difficult sheet metal work with accompanying leakage of air and waste of cooling or heating effect.

Combinations of these several methods of zoning usually provide the most effective zoning. It is then possible to use each method to its greatest advantage without incurring operative or economic penalties which may be inflicted by the exclusive use of any one. Thus, where wide variations in load occur, volume control can be used to reduce the air quantity a limited amount, then as the load continues to fall off, reheat, or by-pass, or both can be used. By-pass and reheat can be used in series very effectively. Many combinations are possible and each case should be considered with regard to its particular requirements when deciding on the method of zoning.

Induction Units—Low pressure type

Induction units are essentially induction type convectors. These units utilize a jet of conditioned air (or primary air) to induce into the unit a flow of room or secondary air which mixes with the primary air. The mixture is discharged into the room through a grille at the top of the unit. Heating coils are located in the secondary air stream for use in heating. Control is obtained by either manually or automatically throttling the jet, and, in addition, heat may be supplied to the secondary coils in summer as well as winter to provide control by reheating. The use of these induction units presents several advantages. Since the secondary air stream is thoroughly mixed with the high velocity low temperature air stream before leaving the discharge of the unit, 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. A unit is usually provided under each window in place of the customary direct radiation, and combines the air distribution system with the heating system. With a conventional system it may be necessary to provide supplementary heating in the form of direct radiation. These induction units may be selected so that their heating coils will have sufficient capacity under gravity conditions (that is, with the fan system off and no primary air entering the unit) to maintain the building or spaces at a reasonable temperature. The use of low temperature dehumidified air which has not been reheated or mixed with room air before delivery to the room results in a reduction in fan capacity and smaller sized duct work. In some cases the use of the by-pass may be desirable in order to keep the primary air volume up and provide additional control. This system can provide a degree of zoning that is usually impossible with conventional

systems since each unit can be put under manual or automatic volume control and reheat control. The selection of units should be made with regard to noise level when related to the noise level of the spaces. The inductive capacity of the unit increases with the jet velocity but too high jet velocities result in a high noise level.

Induction Units—High pressure type

A recent development of the induction unit as previously outlined is the high pressure type of unit. This unit employs nozzles which produce a high velocity jet quietly. The term, high pressure type of unit, is to some extent inaccurate since the pressure at the nozzles, while several times that of the low pressure unit, is still less than the total resistance pressure of a conventional central system. The high velocity jet of primary air induces a flow of secondary room air through coils located in the secondary air stream. The coil in the secondary air stream is supplied with chilled water in summer and hot water in winter and thus handles a large portion of the room sensible heat gain in summer and of the room sensible heat loss in winter. The primary air is supplied at a sufficiently low dew-point to take care of room latent heat gain in summer. In winter, it is supplied at a sufficiently high dew-point to take care of room latent heat losses. Control of temperature is obtained by throttling the water quantity supplied to the secondary coils. The quantity of primary air is greatly reduced due to the fact that a portion of the room sensible heat load is carried by the secondary air stream coil. Since the primary quantity is small, very high velocities can be carried in the supply ducts without requiring fan power in excess of that required for a conventional system. This means that the supply ducts or pipes can be very small and can be run in chases, or furred in at columns with the water pipes. The primary air is treated in the usual manner to provide air at the required dew-point and either 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. The water piping for the units can be so arranged and valved that hot water can be supplied to one zone that may require heating while cold water may be supplied to a zone that requires cooling.

This system is usually limited in application to hotels, apartments, office buildings and other multi-room installations having a large perimeter with relation to the floor area. The units are usually installed beneath the windows, replacing direct radiation or convectors. Where the spaces to be conditioned extend a large distance from the outer wall into the interior of the building, a separate system or zone for the conditioning of the interior portions may be required.

Evaporative Cooling

In climates where on design maximum days, the outdoor wet-bulb depression is relatively high it may be possible to dispense with refrigeration or other cooling sources by use of 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 and 60 F wet-bulb temperature a leaving condition of 62 F dry-bulb, nearly saturated can be obtained. Under some conditions of latent and sensible heat load the results may be entirely satisfactory.

Under those conditions when the outdoor wet-bulb temperature is not quite low enough to permit the use of straight evaporative cooling it is

possible to use precooling coils with refrigeration, well water or a cooling tower as the basic source of cooling to lower the wet-bulb temperature (by sensible heat removal) of the air before it enters the air washer. Where internal heat loads are high, this may be more economical than using return air. Under other conditions where the required supply air dew-point is too low to permit straight evaporative cooling and the sensible heat load not too great, intentional partial saturation may be employed. That is, the low dew-point of the outdoor air is utilized by permitting some of it to pass through the humidifying sprays untreated or pass around the humidifier. All of these remarks with regard to evaporative cooling are based, as indicated, on the assumption that the supply air will consist entirely of outside air. Provision should be made for the return of air from the conditioned spaces for control purposes as well as for winter use in all cases.

Precooling

Where sufficiently cold water from wells or streams is available a saving in the refrigeration cycle may be obtained by the use of precooling. Cooling coils are placed ahead of the dehumidifier or conditioner and the cold water from a well or stream circulated through the coils. 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 is delivered to the refrigeration plant for condensing purposes. The economic advantages of this scheme are apparent and it is frequently used.

Sensible Cooling with Dry Cooling Coils

Under certain atmospheric conditions where a large wet-bulb depression exists and the dew-point of the outdoor air is sufficiently low at all times, proper inside conditions may be obtained by removing sensible heat only from the outdoor air and supplying it to the spaces. Under this condition of a high wet-bulb depression a cooling coil may be located in the air stream and this coil supplied with water from a cooling tower of one type or another. When humidity control is desired sprays to saturate or partially saturate the air may be used after the dry cooling coil. Saturation or partial saturation after the dry air cooler will further reduce the dry-bulb temperature of the supply air and reduce the supply air quantity required. This system has very definite application in hot dry climates and in general is most economical.

Run-Around System

An interesting method of control is found in the use of combined reheating and precooling usually termed the *run-around system.* Coils are placed in the air stream before and after the conditioner and water or brine is circulated around the conditioner from one coil to the other. The water passing through the reheating coil is cooled by the air leaving the dehumidifier and the air is heated by the water. The cooled water is then circulated through the precooling coil where the entering air is cooled by the water and the heated water sent back to the reheating coil. The run-around has the advantage of permitting a higher supply air dew-point temperature than would be possible otherwise. This is due to the fact that continual reheating is available which is not a large penalty on the refrigeration plant since it provides precooling at the same time. This reheating at peak load creates an artificial sensible heat gain which increases the ratio of room sensible heat to room total heat and for a given room temperature results in a higher apparatus dew-point. Thus, while

the volume of supply air is increased, the low side temperature level of the refrigeration plant is raised and this may effect savings in initial and operating costs. This system has the disadvantage of providing a decreasing amount of heat for reheating as the demand for reheating increases.

RELATION TO BUILDING TYPE

Few buildings or spaces are physically identical and those that are similar in this respect may have marked differences in internal loading, zoning requirements, and economic limitations. Consequently it is virtually impossible to establish fixed rules governing the type of system to be used. Each case must be considered on its own merits with due regard to all engineering and economic factors. However, some generalizations are possible.

In small single spaces the use of an elaborate system is undesirable from an initial cost standpoint. Zoning may be often eliminated. This also applies to large single spaces except that in very large spaces the necessity of providing adequate zoning is encountered more frequently. If the spaces are extremely large, physical and economic limitations such as the size of equipment, size and length of ducts, may require the division of the space into sections. Whether these sections are to be made according to zones or whether each section is to be zoned will depend on the particular case.

Where groups of spaces or small buildings are encountered, simple systems still prove the most economical. A single central station with zoning by means of volume control and reheat combined may be entirely satisfactory. If the perimeter of the building is large with regard to the area, induction units of the low or high pressure type may be employed, particularly if it is a multi-room application. Occasionally the dual duct system may be considered, but is infrequently used due to its complications.

Low buildings with large floor areas, such as large department stores and large general offices, may have to be divided into sections for treatment. In the case of large department stores it may be possible to provide a single conditioner with a fan delivering the conditioned air to recirculating fans which supply the various departments or spaces. This application is limited by practicability of running the large conditioned air ducts to the various recirculating fans. In some cases the use of separate systems for each section will be indicated with the added necessity of dividing into horizontal as well as vertical sections. If the latter is required, each vertical section may be handled by a separate system consisting of a single conditioner and fan delivering conditioned air to recirculating fans supplying the horizontal sections. Zoning is obtained by proper allocation of the recirculating fans or other conventional methods used in conjunction with the recirculating fans. For general offices in particular, or types of buildings having a large perimeter, the use of one of the induction type units for perimeter treatment, combined with a conventional system for the treatment of the interior portions, offers possibilities.

High buildings having large floor areas may be successfully handled in many ways. Horizontal or vertical sectionalizing or both may be required, as determined by economic factors and physical limitations. Where vertical sections only are required, the use of a single conditioner and fan for each section delivering conditioned air to zone fans at various floors may be used. These zone fans can be so arranged that one recirculating

fan can handle similar zones on several floors, thus reducing the number of fans required and providing a degree of vertical zoning. 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 extremely large buildings apparatus rooms for the systems may be located in the basement and in the attic and on intermediate floors.

In high buildings having small floor areas the treatment required may be the same as for that of a vertical section of one having a large floor area. A single conditioner and fan can be used to deliver conditioned air to recirculating fans located at various floors.

In all cases of high buildings the necessity for horizontal sectionalizing is indicated by the economical size of air supply and return risers and by the extent to which they encroach upon usable space. In all buildings the necessity for vertical sectionalizing is indicated by the economical size of horizontal supply and return ducts and the space requirements of these ducts.

Usually the most simple systems are best adapted to theaters, auditoriums, and similar applications. Zoning is seldom required other than in connection with auxiliary spaces served by the same system. A conventional central system with by-pass control possibly augmented by reheat usually will suffice. The auxiliary spaces may be supplied with air from the same system controlled by volume reduction and reheat. Balconies and large lobbies frequently justify the use of separate zoning fans.

The foregoing are merely generalizations and suggestions. It is the responsibility of the engineer to explore thoroughly the possibilities of all types of systems and employ that best suited to the purpose from a standpoint of maintained economy, maintenance, life, operation and physical applicability.

EQUIPMENT SELECTION

Other chapters cover in detail most of the items of equipment used in a central system. Each item must be selected not only on its own merits, but in relation to all the other items that go to make up the complete central system. Each item should be considered from the standpoint of both initial cost and operating costs. Consideration must be given to performance at partial loads since most systems operate at full load but a small percentage of time. Many items of equipment have been well standardized and are manufactured in certain definite sizes. The fullest advantage of this should be taken. One item that may be oversized of necessity may permit the use of a smaller piece elsewhere.

Fans operate at full capacity continually in many systems and therefore should be selected for good efficiencies. In winter where higher temperature differentials are used it is possible to use lower air quantities and a two-speed motor may be provided for the fan, resulting in a power saving. In such cases the air distribution under the reduced volume should be investigated before providing this feature.

In selection of the dehumidifier or conditioner the relation of this item to the refrigeration plant is to be given careful consideration. Frequently, it is possible to make a saving in the refrigeration plant by providing more surface in the dehumidifier or conditioner. On the other hand, an excess of capacity in the refrigeration plant can be used to lower the apparatus dew-point (if the lower room humidity is satisfactory) resulting in a reduced quantity of dehumidified air and smaller dehumidifier.

In winter, where preheating coils in outside air intakes are required and subjected to entering air temperatures below freezing, the coils should be selected for operation at full capacity whenever the entering air temperature is below 35 F or where throttling of the steam supply to the coils is desirable. A type of coil that is designed especially for this must be used. In many cases the use of preheaters is not justified since the temperature of the mixture of outside and return air may be entirely satisfactory.

Where reheating coils are used after the supply fan, either as zone control reheaters or boosters, and are relatively near outlets, care is to be used to install these coils so that any stratification of temperature produced by the throttling of the steam or water supply does not result in cold air being delivered to one outlet and warm air to another. Some types of coils that do not produce stratification under throttled conditions are commercially available.

The selection of the refrigerating plant is in itself an economic study. The availability, consumption, and costs of condenser water are to be compared to the initial costs involved and water savings produced by the use of cooling towers or evaporative condensers. Whether or not a direct expansion or flooded system, cold water or brine type of plant will be used is not only a matter of initial cost but one of over-all performance, operating economy, compactness, and in some cases; one of safety. Where water or brine is used as a cooling medium, the possibilities of using lower temperatures and decreased quantities of water or brine, with resultant savings in pumping power and line sizes, are to be compared with the increased power consumption and probable increased cost of the refrigeration machine.

Air cleaning devices are to be selected according to the particular requirements of the project as well as to existing atmospheric conditions. In some applications such as certain types of stores or departments in large stores, lint screens should be provided for the return air as well as filters. Whether or not return air is to be filtered will depend upon the individual case and will be related to the amount of dirt or dust generated in or brought into the conditioned space from various sources. The type of filter or cleaning device to be used depends largely on economic considerations. Obviously an expensive high efficiency cleaning device is not warranted where atmospheric dust or dirt is of such a nature that a less expensive, less efficient device will remove the more objectionable matter. Interior cleaning costs, dust and dirt damage, and hazard due to an accumulation of inflammable dust or dirt within the system are most important factors.

Automatic instruments are nearly always used in present practice for the control of temperature and humidity and to an increasing extent in the control of large refrigerating plants as well as small ones. Whether electric or pneumatic controls are to be used is a function of the particular requirements of the application and with certain exceptions a function of economy. Either electrically or pneumatically operated controls can be made to serve the same purpose though the individual case may favor one or the other. A detailed discussion of automatic controls may be found in Chapter 34. One point is to be emphasized. In general, the simpler the control system the better it will perform. Control systems seldom receive the maintenance they deserve and the fewer the instruments and devices the better the care. Further, it is poor policy to provide, at additional expense, instruments of extreme sensitivity to devices incapable of responding to the control demands.

Insulation is an important factor in air conditioning systems. Its economy with regard to steam and water or brine piping are well known and need no comment. The insulation of duct work is not merely a matter of economics but sometimes is a necessity from the standpoint of limiting the temperature rise of the air even when the ducts are in conditioned spaces. This temperature rise of the air should always be taken into account when apportioning the air and sizing the ducts and will indicate the necessity for insulation. In some cases, leakage of air from the duct where the duct is in a furred space or chase will eliminate the need for insulation by maintaining a reasonable temperature surrounding the duct.

There are practically no items of equipment associated with central air conditioning systems that are not subject to economic limitations as well as those of performance and duty and all of them should be considered in the selection.

ARRANGEMENT OF EQUIPMENT

A proper arrangement of equipment is essential for the proper functioning of any system. Where systems are to be installed in existing buildings the arrangement of equipment may be limited by structural or space considerations, but no compromises that may prevent satisfactory operation or maintenance should be considered.

The location of the apparatus room is often determined by building construction or available space. The closer the apparatus room to the conditioned space, the less expensive is the duct-work. On the other hand if the equipment generates noises it may be necessary to locate the room some distance from the spaces or provide adequate sound and vibration treatment. The scattering of wet apparatus throughout a building is to be avoided unless suitable precautions are taken. It must be remembered that encroachment on spaces that are otherwise usable can be charged against the system as an operating cost.

In general the apparatus should be arranged to have straight line air flow. This is desirable but not always possible. Each change in direction is the source of air resistance, and in addition may cause eddy currents resulting in stratification. The usual order of equipment, beginning at the outside air intake is: outside air screen, outside air louvers, maximum and minimum outside air dampers, preheaters, return air connection, filters, conditioner, by-pass connection with or without reheaters, reheaters, fan and distributing ductwork. See Figs. 1, 2 and 3 for typical arrangements. Use of one or more of the methods of zoning may require a modification of this order but usually only after the dehumidifier or conditioner.

Outside air screens prevent the entry of large foreign matter, birds, etc. The use of louvers or a hood at the outside air intake prevents the entry of rain and snow. Both of these should be used on all systems. As pointed out earlier, the louvers and screens should be of sufficient size to permit the passage of the entire conditioned air quantity.

The minimum outside air damper usually covers the entire face of the preheater, which in turn is selected for the minimum outside air quantity. The maximum outside air damper is designed for the difference between the dehumidified air quantity and the minimum outside air. Its size can be such that its air resistance when open will equal the air resistance of the open minimum outside air damper plus that of the preheater if used. Where the spaces conditioned are very tight against air leakage, some

type of relief or positive exhaust may be necessary when all outdoor air is introduced to the system to provide some means of egress for the air. Such reliefs require back-draft dampers to prevent infiltration when all outdoor air is not being used. A return fan properly dampered as indicated later is sometimes used.

The return air from the conditioned spaces usually is brought into the apparatus between the preheater and the filter. Just as it is necessary to make provision for using all outdoor air it is necessary to make provision for using all return air. Heating an unoccupied building or cooling it after a shut-down is easier if all the air used is return air. Consequently the return connections should be ample for this. In many cases higher velocities can be carried in the return ducts during such times, since the fan will exert a pull at the return connection nearly equal to the resistance of the outside air connection and the return ducts need not be larger than for normal return air quantities at normal velocities. In other cases it is possible that a reduced quantity of air due to the increased resistance of the return duct system may be satisfactory during the starting period. Where the return air system is extensive or complicated a return air fan is desirable. This fan can serve as a combination exhaust and return fan by arranging dampers in its discharge so that the necessary return air can be delivered back into the system and the remainder discharged outdoors. When all outdoor air is passed through the conditioner the entire return air quantity is discharged outdoors.

The by-pass connection normally connects the return air duct system into the apparatus casing between the conditioner and fan. Usually it is sized to handle about 50 per cent of the fan capacity where a variable by-pass is used though extreme load variations may require a greater amount. It is at times good design to locate the reheater in the by-pass connection using a certain amount of by-pass air when heating is required. Since the relatively high resistance of the conditioner is to be balanced by the heating coil and by-pass connection, enough heating surface can be provided to raise the temperature of the by-pass air to the point where the mixture of by-pass air and conditioned air will have the required temperature. When a variable by-pass is used a damper working in opposition to the by-pass damper should be placed across the face of the dehumidifier, for unless the resistances of the two are most carefully balanced at all operating points the proper mixtures of air will not be obtained. The avoidance of by-passing outside air is again stressed. Where the by-pass is made a part of the dehumidifier or conditioner and located on the top or side of it, the return air connection should be made in such a way that stratification of return air is insured, baffles being provided to accomplish this purpose if necessary. Where return air and by-pass 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 by-pass connection, if the return duct system is extensive and the connections simple. In this instance, when the by-pass damper is at maximum opening it may be much easier for outside air to pass through the return damper, into the return duct connection and through the by-pass than for return air to pass through the by-pass connection into the fan. Air always takes the easiest path and if the dehumidifier resistance is high, and the return duct resistances low, this situation is apt to occur unless precautions are taken. A return 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.

The location and arrangement of the dehumidifier, humidifier, or conditioner with reference to each apparatus assembly are more or less standardized. In general, the outside air intake, preheaters, and return air connections precede the conditioner while the by-pass, reheaters and fan follow the dehumidifier. In the cases of the *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 unless the proper precautions are taken.

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 restrictions with regard to location other than those of pumping power, working pressures, line costs, etc. Where open 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, 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.

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 all or 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 bellying or vibration under pulsations in air flow but to withstand the abuse of normal usage. Casings should be adequately braced wherever access doors are installed and all large panels should be adequately reinforced by angle iron.

Each apparatus layout is to be made with accessibility in mind. Where cooling coils are used space for removing and repairing or replacing the coils should be provided. Adequate space is to 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. Provisions should be made for the complete removal and replacement of any part of the system that is subject to wear, deterioration or damage, whether it may be filter, fan wheel, motor, rotor, pump impeller or heat transfer surface.

DESIGN PROCEDURE

The customary design procedure is outlined herewith. For simplification the procedure is set up on the basis of a year 'round system. For summer only or winter only systems, the unrelated parts are to be omitted.

1. Selection of design conditions (inside and outside).
   a. Summer.
   b. Winter.
2. Determination of outside air requirements.

3. Determination of cooling load.
   a. Room sensible heat gain.
   b. Room latent heat gain.
   c. Room total heat gain.
   d. Grand total heat gain.
4. Determination of heating load.
   a. Room sensible heat loss.
   b. Room moisture loss.
   c. Humidification requirement.
   d. Total heating requirement.
5. Determination of apparatus dew-point and dehumidified or humidified air quantity.
   a. Summer (full load and part load).
   b. Winter.
6. Supply air temperature difference and quantity.
   a. Summer.
   b. Winter.
7. Equipment selection.
8. Equipment layout.

The foregoing steps are merely typical. Many applications will require at least a preliminary investigation of some of the latter steps before proceeding with the earlier steps.

REFERENCES

1—Code of Minimum Requirements for Comfort Air Conditioning (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 27).

2—A Comparison of Methods for Controlling Evaporators and Compressors Used in Air Conditioning, by W. A. Grant (*Refrigerating Engineering*, April, May, 1945).

ABBREVIATIONS AND SYMBOLS USED IN CHAPTER

ρ = density of supply air, pounds per cubic foot.
c_p = specific heat of air at constant pressure, Btu per (pound) (Fahrenheit degree).
H_l = latent heat gain of room, Btu per hour.
H_s = sensible heat gain of room, Btu per hour.
h_r = enthalpy of air at room conditions; room wet-bulb, Btu per pound of dry air.
h_s = enthalpy of air at apparatus dew-point, Btu per pound of dry air.
h_v = heat of vaporization at apparatus dew-point temperature.
Q_m = air supply, cubic feet per minute.
t_s = apparatus dew-point, Fahrenheit degrees.
t_r = room dry-bulb temperature, Fahrenheit degrees.
W_r = moisture content of room air, pounds per pound of dry air.
W_s = moisture content of air, pounds per pound of dry air at apparatus dew-point.
SHF = Sensible Heat Factor.
LHF = Latent Heat Factor.
MF = Multiplier Factor.

CHAPTER 44

Owning and Operating Costs

Fixed Charges: Amortization, Interest, Taxes, Insurance, Rent; Maintenance Costs, Service Costs: Operating Refrigeration Equipment, Condenser Water, Heating

THE purpose of this chapter is to discuss the owning and operating costs of heating, ventilating, air conditioning and refrigeration systems for buildings from an economic standpoint so that owners or prospective purchasers may compare the operating economics of one system with another and evaluate properly the over-all costs of the systems instead of considering only the first cost.

There are cases in which it may be desirable to study the possibilities of installing a system for the purpose of obtaining a substantial increase in income or a better return on the investment due to: increased patronage in theaters, stores, or hospitals; increased occupancy in office buildings; improved efficiency of employees in offices or factories; or improvement in a manufactured product or a decrease in its cost of production.

Owning and Operating Costs may be grouped under three headings: (1) *Fixed charges*, (2) *Maintenance Costs*, and (3) *Service Costs*.

FIXED CHARGES

Fixed Charges, which are the costs of owning the system, include: (1) Amortization, (2) Interest, (3) Taxes, (4) Insurance, and (5) Rent.

Amortization

Amortization cost will depend on: (1) the total first cost, and (2) the amortization period.

The *total first cost* of an installation is the actual dollar outlay or capital expenditure required to buy and install the air conditioning, or heating and ventilating system ready for operation. It can be divided into two parts: (*a*) The first cost of the air conditioning or heating and ventilating system itself, and (*b*), other first costs incurred because of the installation of the air conditioning or heating and ventilating system.

The *first cost* of air conditioning or heating and ventilating systems includes the following:

1. Heat producing equipment including boilers, burners, 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 distributing equipment including ducts, outlets, grilles, etc.
5. Refrigerating equipment including piping, pumps, etc.
6. Water conservation devices including towers, evaporative condensers, etc.

The best procedure for establishing the first cost of any system is to select the various parts after thorough engineering study, and then to estimate the installed costs of same. When such detailed work is not warranted or when only rough comparisons are desired between several types, approximate unit costs are of value as time savers.

Approximate installed prices of the various parts are shown in Tables 1, 2, and 3 and, as indicated, vary with the size or capacity of the equipment. The apparatus and items included in each group are stated in the footnotes beneath the tables. By use of these three tables it is possible to obtain a reasonable approximation of the cost of heating or air conditioning a given space or building. In order to use the approximate values indicated in Tables 1, 2 and 3, a rough estimate of the load is required. If time does not permit a determination of the load and if extremely rough figures will suffice, Table 4 may be used as an indication of the price of various types of air conditioning applications.

Other first costs, incurred because of the installation of the air conditioning or heating and ventilating system, include costs of electrical work, plumbing, miscellaneous piping, building alterations, cutting, patching, remodeling, or redecorating after installation, consulting engineer's fees, licenses, permits, etc. These vary so widely that no approximations are possible and each case must be considered alone.

The length of the amortization period to be used depends upon: the type and remaining life of the building or space for which the system is to be used; the type of equipment to be employed as a part of the system; the character of the business; and the lease or ownership conditions. For small shops in rented quarters on short term leases, a period of 5 years or less may be proper, whereas for larger installations in buildings that are owned outright a period of 10 or 20 years or more may be used.

Depreciation due to deterioration and obsolescence must also be considered in arriving at the amortization period. Deterioration and

TABLE 1. TYPICAL INSTALLED COSTS OF HEAT PRODUCING AND HEAT DISTRIBUTING SYSTEMS

BTU PER HOUR (MILLIONS)	SQ FT EDR (STEAM) THOUSANDS	BOILER HORSE POWER	COST IN DOLLARS—PER MILLION BTU PER HOUR[a, b]			
			1	2	3	4
			BOILERS WATER TUBE	BOILERS FIRE-TUBE	HOT WATER SYSTEM FORCED CIRCULATION	DIRECT RADIATION SYSTEM STEAM
0.8	3.3	24		2100	8500	7850
1.2	5.0	36	2300	1750	8200	7600
1.6	6.6	48	2000	1570	8000	7350
2.0	8.3	60	1850	1460	7800	7200
3.0	12.5	90	1600	1230	7600	7000
4.0	16.6	119	1400	1100	7500	7000
5.0	20.8	149	1250	980	7500	7000
6.0	25.0	179	1160	920	7500	7000
7.0	29.2	209	1100	880	7500	7000
8.0	33.3	239	1050	850	7500	7000
10.0	41.6	299	1000		7500	7000
12.0	50.0	358	970		7500	7000
14.0	58.3	418	930		7500	7000
16.0	66.6	478	900		7500	7000
18.0	75.0	538	860		7500	7000

[a]Columns 1 and 2 include hand fired boiler erected, with shaking grates and standard trimmings. Column 1 includes brickwork and rotating soot blowers. Foundations and piping are not included.

Columns 3 and 4 include direct radiation, piping, valves, specialties, and insulation. Boilers, condensate and circulating pumps, boiler connections, and all building construction or alterations are not included.

[b]The approximate figures given above may vary as much as 30 per cent due to job conditions, labor rates, and locality. These figures represent the selling prices of the individual items indicated above based on costs encountered in the year 1940. It is suggested that correction be made according to the particular locality for increases in labor and material costs since that year.

TABLE 2. AVERAGE INSTALLED COSTS OF FORCED AIR HEATING AND AIR CONDITIONING APPARATUS

CFM SUPPLY AIR (THOUSANDS)	COST PER CFM—DOLLARS[d]			
	1	2	3	4
	AIR CONDITIONING EQUIPMENT[a]	HEATING AND VENTILATING EQUIPMENT[b]	DUCTS OFFICE BLDGS.[c]	DUCTS SPECIALTY STORES[c]
5	0.300	0.210		0.250
10	0.270	0.180		0.249
15	0.250	0.155		0.247
20	0.235	0.140		0.245
30	0.220	0.120		0.240
40	0.210	0.110	0.340	0.235
60	0.210	0.110	0.330	0.230
80	0.210	0.110	0.325	
100	0.210	0.110	0.320	
120	0.210	0.110	0.310	
140	0.210	0.110	0.305	
160	0.210	0.110	0.295	
180	0.210	0.110	0.290	
200	0.210	0.110	0.280	

[a] *Air conditioning equipment* includes fans and drives, filters, heaters, spray dehumidifiers or cooling coils, automatic controls, apparatus casings and insulation, and recirculating pumps.

[b] *Heating and ventilating equipment* includes fans and drives, filters, heaters, automatic controls and apparatus casings.

[c] Includes ducts, grilles, outlets, insulation where required and specialties.

[d] The approximate figure given above may vary as much as 35 per cent due to job conditions, labor rates and locality. Figures are based on conventional systems. Building alterations, piping, plumbing and wiring are not included. These figures represent the selling prices of the individual items indicated above based on costs encountered in the year 1940. It is suggested that correction be made according to the particular locality for increases in labor and material costs since that year.

TABLE 3. AVERAGE INSTALLED COSTS OF REFRIGERATION SYSTEMS AND WATER SAVING DEVICES

TONS REFRIGERATION	COST—DOLLARS PER TON[a]					
	1	2	3	4	5	6
	RECIPROCATING (WATER COOLING)[a]	CENTRIFUGAL (WATER COOLING)[a]	RECIPROCATING DIRECT EXPANSION[b]	EVAPORATIVE CONDENSER[c]	COOLING TOWER STEEL[d]	COOLING TOWER WOOD[d]
25	147		106	47		
50	132		86	40		
75	125		77	35		
100	120		72	32	47	40
150	115	137	68	30	45	38
200	113	124			44	36
250	110	115			42	35
300		107			40	33
400		97			38	31
500		93			36	29

[a] Columns 1 and 2 include compressors, evaporators, and water-cooled condensers; auxiliaries; electric motor, starter and drive; refrigeration piping, refrigerant and insulation of cooler and suction lines. Turbine driven centrifugal equipment may cost about 4 to 6 per cent more than motor driven, if condensing turbine and steam condenser, with supplementary equipment, are used.

[b] Column 3 includes same items as Column 2 except evaporator and auxiliaries are omitted.

[c] Column 4—These values are additive to Columns 1 and 3.

[d] Columns 5 and 6 include towers erected and a reasonable allowance for condenser water piping and pumps.

[e] The approximate figures given above may vary as much as 25 per cent due to job conditions, labor rates and locality. Building alterations, supply water and plumbing connections, wiring, chilled water piping and pumps are not included in these figures and may vary widely. Chilled water piping and pump costs may vary between $10 and $50 per ton of refrigeration effect. These figures represent the selling prices of the individual items indicated above based on costs encountered in the year 1940. It is suggested that correction be made according to the particular locality for increases in labor and material costs since that year.

TABLE 4. APPROXIMATE COSTS OF ALL-YEAR AIR CONDITIONING[a,b]

APPLICATION	SELLING PRICE PER UNIT INDICATED											
	SQ FT NET AREA			PERSON			TON REFRIGERATION			CFM SUPPLY AIR		
	LOW	AVG	HIGH	LOW	AVG	HIGH	LOW	AVG	HIGH	LOW	AVG	HIGH
Banks	$1.25	$2.00	$3.50	$80	$140	$320	$350	$480	$700	$0.95	$1.40	$2.75
Department Stores: Basement	0.60	1.00	1.15	25	40	100	250	325	400	0.60	0.80	1.00
Department Stores: Main Floor	0.60	1.05	1.25	25	40	80	250	325	400	0.60	0.75	0.90
Department Stores: Upper Floors	0.60	0.90	1.10	30	45	100	300	350	400	0.60	0.80	1.00
Hotels—Guest Rooms				$240[c]	$300[c]	$420[c]						
Office Buildings	$0.85	$1.25	$2.00	110	135	180	$300	$375	$500	$0.75	$1.05	$1.50
Offices—Small Suites	1.00	1.75	3.25	110	180	350	300	450	800	0.75	1.25	2.50
Restaurants	1.40	2.20	3.50	15	18	25	215	280	400			
Specialty Shops: Barber Shops and Beauty Parlors	$1.50	$2.50	$4.00	$50	$100	$150	$325	$400	$500	$0.70	$1.00	$1.50
Specialty Shops: Dress Shops	1.25	1.50	2.00	45	55	75	230	275	350			
Specialty Shops: Drug Stores	1.25	1.50	2.00	45	55	75	240	275	325			
Specialty Shops: Five-and-Ten	0.90	1.20	2.80	22	33	55	220	270	425	0.80	0.95	1.50
Specialty Shops: Shoe Stores	0.95	1.20	1.75	35	40	50	230	275	350			
Theaters				$9[d]	$15[d]	$23[d]	$215	$300	$450	$0.65	$1.00	$1.25

[a] Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne and W. A. Grant (*Pitman Publishing Corp.* 1940, p. 64).

[b] There is a wide variation in unit prices such as shown above, due to differences in load concentration and in completeness of physical layout. Prices cover air-conditioning work only, for year-round use (except the low column, which in general includes summer cooling only). Net area (square feet) includes corridors, but excludes elevators and service spaces. Excluded are the cost of plumbing and steam connections, electrical wiring, painting and redecorating, building construction and alterations, and boilers or burners. Low figures in general are simply direct expansion systems using city water for condensing. High figures are usually chilled water systems with cooling towers or evaporative condensers, and may be considered high quality systems as well.

[c] Price per room.

[d] Price per seat.

maintenance generally go hand in hand. If a long depreciation period is to be used, then the item for maintenance, repair and replacement of wearing parts must be greater than for a short depreciation period.

Obsolescence depends mainly on time required for the equipment to become out-moded. Air conditioning, particularly, would probably suffer more from obsolescence in small plants than in large establishments. In addition the obsolescence of the building or property in which the equipment is installed may have a similar effect upon the equipment.

An approximation of the useful life of various items of equipment and parts of systems is shown in Table 5. It should be noted that if an appropriate maintenance item is not established, the rate of equipment deterioration may be substantially increased.

TABLE 5. APPROXIMATE LIFE OF EQUIPMENT

(Including Obsolescence and Deterioration)[a]

	LIFE IN YEARS
1. HEAT PRODUCING EQUIPMENT	
(a) Boilers	15
(b) Stokers and burners	10
2. HEAT DISTRIBUTING EQUIPMENT	
(a) Piping—copper	20
(b) Piping—iron	10
(c) Radiation—concealed	12
(d) Radiation—direct	10
(e) Valves and specialties	5
3. AIR HANDLING EQUIPMENT	
(a) Filters—automatic	8
(b) Heating and cooling coils	10
(c) Spray humidifiers and dehumidifiers	10
(d) Fans	10
(e) Air conditioning units	10
(f) Motors	15
(g) Electrical starting equipment	8
(h) Pneumatic control systems	10
(i) Electric control systems	8
4. AIR DISTRIBUTING EQUIPMENT	
(a) Ductwork	20
(b) Outlets, grilles	20
(c) Duct insulation	10
(d) Painting	2
5. REFRIGERATING EQUIPMENT	
(a) Centrifugal refrigerating machines	15
(b) Reciprocating refrigerating machines	10
(c) Motors and starters (see Item 3 above)	
(d) Piping—copper	20
(e) Piping—steel	15
(f) Pumps	15
6. WATER SAVING DEVICES	
(a) Evaporative condensers	10
(b) Cooling towers	10
(c) Wells	Varies Widely

[a]Modern Air Conditioning, Heating and Ventilating, W. H. Carrier, R. E. Cherne and W. A. Grant (*Pitman Publishing Corp.* 1940, p. 66).

Interest

The interest chargeable may not represent the existing money rates. It may include an item to cover the diversion of capital or other items depending on existing tax laws which may make it necessary to charge interest due to diversion of capital as a cost item. It should be noted that interest may be based on an unamortized balance. As an example, a 15-year amortization period with a 4 per cent interest rate will approximate a 2.1 per cent average annual interest rate.

TABLE 6. OWNING AND OPERATING COST

FIRST COST		ANNUAL SERVICE COST	
Cost of mechanical system		Electric Power Costs	
Other costs		Fans	
First Cost (FC)—Total		Pumps—Chilled water	
		Pumps—Condenser water	
ANNUAL FIXED CHARGES		Pumps—Well water	
		Cooling tower fans	
Amortization—Depreciation period Y years		Cooling tower pumps	
Interest rate I%		Refrigeration machines	
Amortization and Depreciation $\frac{FC}{Y}$ =		Miscellaneous or other	
		Gas	
Interest: $\frac{Y+1}{2Y} \times I$ =		Coal	
Taxes		Oil—for boilers or Diesel engines	
Insurance		Steam	
Rent		For direct heating	
Annual Fixed Charges:		For Ventilation—preheaters	
(Total)		For Ventilation—reheaters	
		For Turbine driven equipment	
ANNUAL MAINTENANCE COSTS		For Engine driven equipment	
		Sewers	
Lubricating oil and grease		Charges for discharging well water into public drainage systems	
Painting for corrosion protection or other purposes		Annual Service Costs—TOTAL	
Replacement of worn parts			
Refrigerant		SUMMARY	
Wages of engineer or operator			
Annual Maintenance Cost—		Annual Fixed Charges	
		Annual Service Costs	
TOTAL		Annual Maintenance Costs	
		Annual Owning and Operating Costs—TOTAL	

Insurance

The rate for insurance may vary considerably depending on the type of structure in which the equipment is located and upon other governing factors. A rate of about $0.60 per $1,000 may be considered as being representative for normal installations.

Rent

If the equipment under consideration is to be located in rented or leased quarters it may be necessary to include an item for space rental.

An orderly arrangement of the various components of owning and operating costs, which will also serve as a check list to forestall inadvertent omissions, is illustrated in Table 6. The formulae for computing amortization and interest are given in the table. Interest should be computed on the undepreciated portion of investment only.

MAINTENANCE COSTS

Maintenance costs include replacement parts and the labor required for making repairs, replacing parts, cleaning, painting, etc. It should be noted that major overhauling or complete replacement may restore the capital value of certain items of equipment, and in such cases the costs incurred may not necessarily be charged as maintenance costs. Generally, routine labor requirements will be the function of an operating engineer or staff and the responsibility of this group may extend beyond the equipment being discussed here; hence, it is important to include only an equitable share of the time of this group. Extraordinary repairs involving special machinery will usually be covered by contract with equipment service divisions and should be accounted for on that basis.

Many of the items included in maintenance costs are highly variable and depend on the type and quality of the purchased equipment. For large air conditioning installations, using high quality equipment, some approximate costs per ton are given in Table 7.

In addition to the costs listed in Table 7, consideration should possibly be given to other items such as: water treatment for boilers; other boiler and heating plant cleaning and repairs; repair and replacement of heating plant valves, traps, and vents; water treatment for cooling tower or chilled spray water; drive belts, possible damage due to freezing weather; cleaning of air ducts; and repairs to insulation.

SERVICE COSTS

Service 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:

$$\text{Annual power cost} = \frac{0.746\ (\text{bhp})\ H\ R}{\eta} \qquad (1)$$

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.

Refrigerating Equipment Operating Cost

In an air conditioning system, the refrigerating equipment is usually the largest power-consuming item to be considered. Also, the prediction

TABLE 7. APPROXIMATE MAINTENANCE COST FOR LARGE AIR CONDITIONING INSTALLATIONS, USING HIGH QUALITY EQUIPMENT

	DOLLARS PER TON PER YEAR
Repairs for refrigeration machinery	0.60
Refrigerant	0.25
Oil and grease	0.07
Painting (Water boxes and dehumidifiers)	0.25
Filters, clean and recoil 4 times per year	0.85
Controls, outside service	0.15
Cleaning air conditioners	1.15

of operating cost is more complex, since the power required for summer cooling is affected by many factors which are of a variable nature. Among these are solar radiation, temperature difference between outside and inside, sensible and latent heat brought in with outside air, sensible and latent heat from people, heat released by electric lights, and, in some cases motor-driven equipment, cooking devices, and other equipment used in the conditioned spaces. Some of these factors vary with the weather while others are substantially independent of it. The relative proportion of each factor varies widely even among installations of the same application, depending on building layout and location, the size and quality of the establishment, the personal idiosyncracies of the owner, and other items.

In a strict sense, it is necessary to evaluate the effect of all such factors in order to predict the operating cost of a refrigeration plant. In most cases this procedure will prove to be too tedious, or it may not be possible because the exact breakdown of load data is unknown. The use of a simplified semi-rational formula, Equation 2, which takes into account for each application the number of hours open for business and the geographical location, will provide a value of H which may be used in Equation 3 to determine the *Season Power Cost.*

$$H_e = m\ (b + cf) \qquad (2)$$

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 which is due to external sources at maximum design conditions.
f = Ratio of the number of hours for a particular city, when the outside 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.

Table 8 was calculated from Equation 2. It should be pointed out that certain southern cities may have seasons longer than the 5-month period indicated in Table 8. 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 at relatively light load, since the table already includes the more severe part of the season.

TABLE 8. 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	PHILA-DELPHIA	OKLA-HOMA CITY	ST. LOUIS	WASH-INGTON 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
Specialty 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).

The season electrical power cost for refrigerating equipment is then given by the following equation:

$$\text{Season power cost} = \frac{0.746\,(\text{bhp}_t)\,T\,H_e\,R}{\eta} \qquad (3)$$

where

bhp_t = 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 = Tons of refrigeration at maximum design load.
H_e = Equivalent full load refrigeration operating hours (Table 8).
R = Power cost, dollars per kwhr, including demand and energy charges.
η = Motor efficiency at average load (decimal).

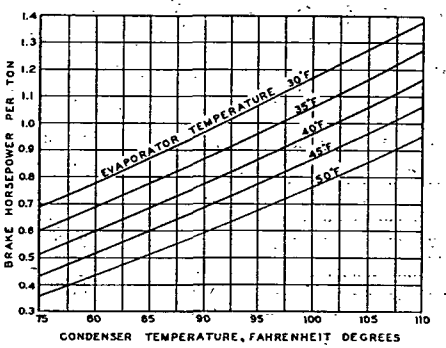


FIG. 1. TYPICAL BRAKE HORSE POWER REQUIREMENTS FOR REFRIGERATION[a]

[a]Values given are representative of "F-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 per cent to 0.5 per cent for each Fahrenheit degree the liquid temperature is lowered.

Table 9 will be useful in estimating the design load in the absence of detailed load estimates.

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 bhp-hr 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 and the gpm 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 8).
C = Water cost, dollars per 1000 gal.

The average gpm 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 gpm per ton obviously is equal to the design gpm 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.

TABLE 9. REPRESENTATIVE TONS PER SQUARE FOOT FOR VARIOUS APPLICATIONS

	Low	Avg	High
Department Store (Main Floor)	0.0050	0.0058	0.0067
Department Store (Upper Floors)	0.0029	0.0046	0.0054
Dress Shop	0.0029	0.0050	0.0071
Drug Store	0.0033	0.0054	0.0083
Lunch Room	0.0083	0.0108	0.0133
Office Bldg.	0.0021	0.0025	0.0038
Restaurant	0.0058	0.0075	0.0100
Shoe Shop	0.0029	0.0042	0.0063
Theater	0.064[a]	0.078[a]	0.093[a]

[a] Tons per Seat.

The average gpm 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 15. In lieu of this tedious work, the average gpm per ton may be taken as 80 per cent of design gpm per ton with reasonable accuracy, for the condition of variable temperature of entering water obtained from rivers and lakes.

When cooling towers or evaporative condensers are used, the windage and evaporation losses are usually between 3 per cent and 5 per cent of the water circulated.

Heating

The method of estimating fuel consumption to balance the building heat loss is given in Chapter 20. 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 outside 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

Modern Air Conditioning, Heating and Ventilating, by W. H. Carrier, R. E. Cherne, and W. A. Grant—Chapter IV (Pitman Publishing Corp., 1940).

Predicting Operating Hours of Refrigeration Equipment Used in Air Conditioning, by W. A. Grant (*Refrigerating Engineering*, July, 1941).

Cost of Operation of Refrigeration Used for Air Conditioning, by R. E. Cherne (*Refrigerating Engineering*, December, 1943).

CHAPTER 45

Industrial Air Conditioning

Atmospheric Conditions Required, General Requirements, Classification of Problems, Control of Regain, Moisture Content and Regain, Conditioning and Drying, Control of Rate of Chemical Reaction, Control of Rate of Biochemical Reactions, Control Rate of Crystallization, Elimination of Static Electricity

INDUSTRIAL air conditioning is primarily concerned with the atmospheric conditions required for the manufacture, processing, and preservation of material, equipment and commodities. The fundamental factors, one or more of which may govern these conditions are: humidity, temperature, air motion and air purity. The term air purity may have reference to the quantity of dust, bacteria, odors, or toxic gases present.

The optimum atmospheric conditions have been established for a large variety of industrial processes, but when the requirements are unknown it may be necessary to determine them by laboratory tests or by research. When the requirements are not definitely specified it is advisable to design the equipment to be flexible and to provide for future increase of facilities and capacity. Air conditioning requirements for human comfort are defined in Chapter 12. If the atmospheric conditions required for the process are not within the range of human comfort, separate consideration must be given to the maintenance of conditions which will not have a detrimental effect upon the health of the occupants.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity for processing depends upon the product and the nature of the process. As far as the behavior of the material and its desired final condition are concerned, each material and process presents a different problem. The desirable relative humidity may range from a low of 5 per cent, as in certain industrial applications, such as insulation winding processes, up to a condition approaching saturation, as in processes relating to textiles, tobacco and baking industries.

Similarly, the most favorable temperature will vary according to the specific material and particular process. Frequently a compromise between the known optimum condition for processing and that required for reasonable worker comfort is desirable. This is particularly true where unconfined processes are required in departments where people are working and their health, comfort and productive efficiency must be considered.

It is generally recognized that relative humidities of 50 per cent or less are on the dry side. Such conditions are conducive to low regains in hygroscopic materials, drying out, increased brittleness of fibrous materials, prevalence of increased static electricity and tendencies toward increased dust liberation from the product. Relative humidities higher than 50 per cent are considered to be on the damp side. These conditions are conducive to high regain, promote softness and pliability in materials, decrease static electricity and tendencies toward reduced generation of product dust which represents a loss in weight of the material in process.

In many processes, the optimum desired air conditions are variable according to the stage and progress of the processing cycle, from the raw material to the finished product. Some materials, such as cotton textiles, begin with a low relative humidity in the carding and picking rooms, and after passing through the various intermediate steps with a gradual increase of relative humidity, the product is subjected to relative humidities of from 75 to 85 per cent in the final stage of weaving. Other processes are encountered that require the reverse of this procedure, starting with a high relative humidity and finishing with a low relative humidity, as is the case when producing glue and gelatinous materials and making gelatine capsules.

In some cases the temperatures listed in Table 1 have no direct influence upon the product itself, except for effect upon efficiency of the employees which in turn affects workmanship, uniformity and the cost of production. In this category may be included the automobile assembly line. The time necessary to assemble the many parts into a complete unit is a factor recognized and associated with the worker's comfort, and the avoidance of fatigue with subsequent loss of efficiency.

Air conditioning contributes an important role during the processing, machining and honing of precision metal parts, instruments, tools, engines, guns, etc., which demand micrometric accuracy of dimensions, and which are affected by small temperature variations. Hence, some uniform condition is usually selected, both as to temperature and humidity to serve the demands of the worker's comfort and the exacting requirements of the process.

The temperatures and relative humidities listed in Table 1 should be analyzed in relation to the qualified requirements of the process. Conditions generally acceptable for *industrial processing* and for *general storage* are listed in these tables. While it is true that many storage requirements demand the control of some fixed air temperature and relative humidity condition, to hold and preserve the contents, it is not generally referred to as *processing*.

Logically many phases of drying may be included in the category of air conditioning for industrial processing, especially where temperature and humidity, by direct influence to product, bring about some definite change in physical characteristics as well as in weight. (See also Chapter 47.) As an illustration, refer to the conditions that are required to control the rate of crystallization in coating pans in which sugar syrup is applied to various forms of pills, nuts, gum, etc., in consecutive liquid doses, until a crystallized coating or jacket is built up to the required size and thickness. Here, the primary problem is one of drying which requires the supply of air at some fixed volume and velocity along with regulated control of both dry- and wet-bulb temperatures. The wet-bulb depression determines the rate of moisture pick-up or drying by the air and may be termed the *drying head*. Of equal importance is the uniformity at which the wet-bulb is maintained. If this is allowed to vary, poor results will follow due to checking and cracking of the unfinished coating. This is obvious when it is realized that continuous evaporation of moisture is taking place during the process and also that the temperature of the material corresponds to the air wet-bulb temperature and will vary accordingly. With undue expansion and contraction with every temperature change, the thin crystallized coatings which are not elastic will check and crack before the process is completed.

GENERAL REQUIREMENTS

Air conditioning apparatus for industrial purposes must be capable of absorbing heat from various sources such as machinery power, electric lights, people, sunlight and chemical reactions; of warming or cooling to any desired temperature; and of providing ample air supply. Refrigeration may or may not be required, depending upon natural conditions, the required relative humidity and the maximum permissible temperature. Washing, purifying and treating the air may be desirable. Good distribution is essential for the control of air motion and for the prevention of uneven conditions. Air velocity should not be so high that it will cause damage to the commodity or interfere with the industrial process. Accurate, sensitive and reliable automatic control of humidity or temperature is vital in most cases.

Outside weather conditions and the ventilation required for workers are of secondary importance in relation to the total work to be done by the air conditioning system. In extreme cases of high concentration of industrial heat from machinery and ovens the error of entirely omitting the heat gain through the building structure would not be serious. At the other extreme, where low temperatures must be produced with refrigeration and where comparatively little power is required by the machinery, the heat gain through the building structure will become the major factor in determining the size of equipment and in this case the ventilation requirement assumes importance.

Buildings which are to be air conditioned should therefore be designed with careful consideration of over-all cost and efficiency. Condensation resulting from high humidities must be prevented by suitable materials and construction, or else collected and drained to prevent loss of product or quick deterioration of the structure. Air leakage or filtration may add greatly to operating costs or make the maintenance of low humidities (relative or absolute) wholly impossible. Low temperatures require good insulation.

It is apparent that the subject of air conditioning for industrial processes is extensive and greatly involved, and that a detailed treatment is therefore beyond the scope of this chapter.

CLASSIFICATION OF PROBLEMS

Any industrial air conditioning problem may be listed under one or more of the following five classifications: (1) control of regain, (2) control of rate of chemical reactions, (3) control of rate of biochemical reactions, (4) control of rate of crystallization, and (5) elimination of static electricity.

Control of 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. This influence is due to the fact that 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.

In industries where the physical properties of a product affect its value, the percentage of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Standards of regain are firmly fixed in trade with fair

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING

Industry	Process	Temperature Fahrenheit Degrees	Relative Humidity Per Cent
Automobile	Assembly line	65 to 80	40 to 55
	Precision parts—honing—machining	75 to 80	40 to 55
Baking	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Dough retarding	32 to 40	76 to 85
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	90 to 95	80 to 90
	Storage of flour	65 to 75	55 to 65
	Storage of yeast	32 to 45	60 to 75
Biological Products	Vaccines	below 32	
	Antitoxins	38 to 42	
	Blood bank	38 to 42	60 to 65
	Penicillin (dehydrated)	36 to 40	
Brewing	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
Ceramic	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60 to 80	35 to 65
Chemical	General storage	60 to 80	35 to 50
Confectionery	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
Distillery	General manufacture	60 to 75	45 to 65
	Storage of grains	60	30 to 45
Drug	Deliquescent powder	75	35
	Effervescent granulations	80	40
	Liver extracts (powdered)	70	20 to 30
	Storage of powders and tablets	70 to 80	30 to 35
	Tablet compressing	70 to 80	40
	Packaging	80	40
Electrical	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
Food	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats (frozen)	0 to 5	85
	Storage of meats (above freezing)	36	85
	Storage of sugar	80	35
Fur	Drying of furs	110	
	Storage of furs	28 to 40	50 to 65

TABLE 1. TEMPERATURES AND HUMIDITIES APPLICABLE TO INDUSTRIAL AIR CONDITIONING—(Concluded)

Industry	Process	Temperature Fahrenheit Degrees	Relative Humidity Per Cent
Incubators	Chicken	99 to 102	55 to 75
Instruments	Repair and calibration	68	50 to 55
Laboratory	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
Leather	Drying of hides	90	
	Mulling	95 to 100	95
Library	Book storage	65 to 70	38 to 50
Linoleum	Printing	80	40
Matches	Manufacturing	72 to 74	50
	Storage of matches	60	
Munitions	Fuse loading	70	55
Paint	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	
	Air drying of oil paints	60 to 90	25 to 50
Paper	Binding, cutting, drying, folding, gluing	60 to 80	40 to 60
	Storage of paper	75 to 80	40 to 60
	Testing Laboratory	60 to 80	55 to 65
Photographic	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
Printing	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	75 to 80	50 to 60
	Storage of rollers	70 to 90	50 to 55
Rubber	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
	Cementing	80	25 to 30
Soap	Drying	110	70
Textile	Cotton— carding	75 to 80	50 to 55
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	50 to 70
	weaving	68 to 75	85
	Rayon— spinning	70	85
	throwing	70	60
	weaving	75 to 88	60 to 75
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
	Testing Laboratory	70	65
Tobacco	Cigar and cigarette making	70 to 75	55 to 75
	Softening	90	85
	Stemming or stripping	75 to 85	70

penalties for excesses. Deficiencies result in loss of revenue to seller and loss of desirable quality to buyer.

Manufacturing economy therefore requires that the moisture content be maintained at a percentage 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.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight, it is proper that they contain a normal or standard moisture content.

Moisture Content and Regain

The terms *moisture content* and *regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or 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 grains is dried to a bone-dry weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

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. During the processing of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb 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 as a basis in the calculations to determine the regain.

The moisture content of a hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire various percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying varies with the nature of the material, its thickness and density.

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 relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 F has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

The regain or moisture content affects the physical properties of textiles to a marked degree, changing the strength, pliability and elasticity.

The fact that the regain of textiles will come into equilibrium with the

TABLE 2. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75° F

Classification	Material	Description	Relative Humidity—Per Cent									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	18.8	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 Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fiber	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	U. S. B. of S.
	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	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
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 tire	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
	Sulphuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

conditions of the surrounding air and vary with its temperature and relative humidity is the fundamental basis for the control of physical qualities during manufacture. During the preparation processes in a cotton mill, the cotton fibers should be in a condition to be easily carded.

These preliminary processes are carried out best in a relative humidity

of 50 to 55 per cent. As the cotton fiber comes to the spinning operation, more flexibility is needed and the relative humidity is increased in this department. For many years, 65 per cent relative humidity was considered the optimum. To offset the extra work performed on the fiber as the spindle speed is increased, many cotton mills now carry 70 per cent relative humidity in the spinning rooms[1]. Winding, warping and weaving are all processes calling for great flexibility and a consequent need for higher humidity.

Other textile fibers, due to their different natural characteristics, are processed under relative humidities and temperatures applicable to each.

Rayons, on account of great loss of strength with the higher regains, should be processed in a relative humidity of 55 to 70 per cent. Acetate silk, another chemical fiber, with approximately 50 per cent of the regain of rayon, may be processed between 60 and 65 per cent relative humidity.

All hygroscopic materials when in the state of absorbing moisture from the surrounding air produce a sensible heat rise to the air equivalent to the latent heat released by air to the material. This adiabatic conversion may account for a small percentage of the total heat load of the conditioned space.

Conditioning and Drying

In general, the exposure of materials to desirable conditions for treatment may be coincidental 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 purpose of conditioning or drying is usually 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. 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 controlled atmospheric conditions.

Control of Rate of Chemical Reactions

A typical example of the second general classification, the 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 obtains a solution of known strength throughout 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 film homogeneous throughout. Desirable temperatures for *drying* varnish vary with the quality. A relative humidity of 65 per cent is beneficial for obtaining the best processing results.

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 65 per cent is maintained so as to hold the surface of the dough open to allow the carbon dioxide gases formed by the fermentation to pass through and produce a loaf of bread, when baked, of even, fine texture without large voids.

Another example of a similar process is found in the curing of macaroni. The flour and water mixture is fermented and dried. As it is necessary to have a definite amount of water present to carry on a fermentation process, the moisture must be removed in a relatively short period to stop fermentation and prevent souring and in such a manner as to avoid setting up internal strains in the mixture. Best results are obtained with the correct cycles of both temperature and humidity.

The curing of fruits, such as bananas and lemons, also comes under this classification. Bananas are treated somewhat differently and to accomplish the required results, a cycle of temperatures and relative humidities is used. The starches in the pulp of the fruit must be changed and the skin cured and colored, after which the fruit is cooled to maintain as low a rate of metabolism as possible. Ideal conditions range between 55 to 57 F and in no case should the temperature go below 49 F, as the starches then would become fixed and be indigestible.

The curing of lemons is an entirely different problem. Bananas are cured for a quick market, while lemons are held for a future market. The process, therefore, varies in the temperature used. Temperatures from 54 to 59 F have been found to be best suited for this process. A high relative humidity of 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so it will be sufficiently tough to permit handling.

Tobacco from the field to the finished cigar, cigarette, plug or pipe tobacco, offers another interesting example of what may be done by industrial air conditioning in the control of color, texture and flavor. In the processing of tobacco, the first three classifications of air conditioning are involved, and only through close atmospheric control can the best quality of the leaf be developed.

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.

Elimination of Static Electricity

The presence of static electricity is very detrimental to the satisfactory and economical processing of many light materials, such as textile fibers, paper, etc. It is also extremely dangerous where explosive atmospheres or materials are present. Fortunately, this hazard is easily eliminated by increasing the relative humidity.

In attempting to eliminate static electricity, 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 50 per cent 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 be necessary to maintain a room relative humidity of 65 per cent, or even more, before the desired results can be obtained.

CALCULATIONS

The methods for determining the proper heating and cooling loads for the various industrial processes are similar to those outlined in Chapters 14 and 15. Because of the large number of motors and heat producing units usually prevalent 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.

REFERENCE

1—The Present Status of Textile Regain Data, by A. E. Stacey, Jr. (*National Association of Cotton Manufacturers*, 1927).

BIBLIOGRAPHY

Food

Air Conditioning for Sausage Manufacturing Plants, by M. G. Harbula (A.S.H.V.E. Transactions, Vol. 28, 1922, p. 343).

Air Conditioning in the Bakery, by W. L. Fleisher (A.S.H.V.E. Transactions, Vol. 37, 1931, p. 141).

Proper Air Conditions for the Manufacturing of Confections, by A. E. Stacey, Jr. (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, October, 1937, p. 640).

Air Conditioning and Refrigerating Large Bakeries, by W. L. Fleisher (*Heating, Piping and Air Conditioning*, December, 1929, p. 621; January, 1930, p. 24).

Pre-Cooling Fruits and Vegetables with Circulating Air, by C. E. Baker (*Heating, Piping and Air Conditioning*, January, 1932, p. 42).

Air Conditioning Maintains Quality of Fruits and Vegetables, by C. E. Baker (*Heating, Piping and Air Conditioning*, August, 1935, p. 369).

Air Condition the Bakery Throughout, by W. W. Reece (*Heating, Piping and Air Conditioning*, August, 1936, p. 149).

Banana Ripening Manual, Circular No. 14, Equipment Department, Fruit Dispatch Co., New York, N. Y.

The Commercial Storage of Fruits, Vegetables and Florists' Stocks, by D. H. Rose, R. C. Wright and T. M. Whiteman (*U. S. Department of Agriculture, Circular* No. 278).

Temperature Studies of Some Tomato Pathogens, by Alice A. Nightingale and G. W. Ramsey (*U. S. Department of Agriculture Technical Bulletin* No. 520, August, 1936).

Refrigeration of Oranges, by H. M. Hendrickson (Application Data Section No. 17, *Refrigerating Engineering*, September, 1940).

Refrigeration of Lemons and Grapefruit, by H. M. Hendrickson (Application Data Section No. 18, *Refrigerating Engineering*, October, 1940).

Printing

Hygro-Xylometric Control of Air Conditioning, by H. H. Tunis (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, November, 1937, p. 637).

Air Conditioning of Press Rooms, by I. C. Baker (*Heating, Piping and Air Conditioning*, July, 1931, p. 553).

Newspaper Plant Air Conditioned for Efficient Production, by J. A. Osborn (*Heating, Piping and Air Conditioning*, January, 1932, p. 40).

Air Conditioning Frees Rotogravure Plant from Weather Hazards, by H. E. Jacobsen (*Heating, Piping and Air Conditioning*, September, 1935, p. 423).

Bureau of Standards Studies Determine Press Room Conditioning Requirements, by C. G. Weber (*Heating, Piping and Air Conditioning*, March, 1936, p. 137).

Silica Gel Air Conditioning System Serves Rotogravure Printing Plant, by H. E. Ryerson (*Heating, Piping and Air Conditioning*, August, 1937, p. 497).

Moisture Control by Liquid Absorption Offers a Useful Air Conditioning Tool, by F. M. Johnson (*Heating, Piping and Air Conditioning*, December, 1938, p. 782).

Meeting the Air Requirements in a Bindery and Printing Plant, by N. W. Downes (*Heating and Ventilating*, February, 1921, p. 25).

Heating and Ventilating the Bureau of Engraving and Printing (*Heating and Ventilating*, February, 1923, p. 55).

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).

Air Conditioning in the Paper Industry (*Heating and Ventilating*, October, 1935, p. 23).

Air Conditioning the Newspaper Plant, by R. T. Williams (*Heating and Ventilating*, September, 1937, p. 63).

Air Conditioning Requirements of Multicolor Offset Printing, by C. G. Weber (*Refrigerating Engineering*, December, 1936, p. 6).

A Study of the Removal of Sulphur Dioxide from Library Air, by A. E. Kimberly and A. L. Emley (*U. S. Bureau of Standards Miscellaneous Publication*, No. 142, October 17, 1933).

Reactions of Lithographic Papers to Variations in Humidity and Temperature, by C. G. Weber and L. W. Snyder (*U. S. Bureau of Standards Journal Research*, January, 1934).

The Treatment of Offset Papers for Optimum Register, by C. G. Weber and M. N. V. Geib (*U. S. Bureau of Standards Journal Research*, February, 1936).

Summary Report of National Bureau of Standards Research on Preservation of Records, by A. E. Kimberly and B. W. Scribner (*U. S. Bureau of Standards Miscellaneous Publication*, No. 154, March 16, 1937).

Miscellaneous

Effect of Air Conditioning upon Munitions, by J. I. Lyle (A.S.H.V.E. Transactions, Vol. 23, 1917, p. 383).

Air Conditioning of Blackout Plants, by W. A. Grant (A.S.H.V.E. Transactions, Vol. 48, 1942, p. 459).

Air Conditioning in Industrial Processes (Match Factory), by W. L. Fleisher (*Heating, Piping and Air Conditioning*, March, 1931, p. 196).

Air Conditioning as Applied in Theatres and Film Laboratories, by D. C. Lindsay (*Transactions Society of Motion Picture Engineers*, April, 1927, Vol. XI, No. 30, p. 335-365).

Air Conditioning for Textile Plants Making and Using Synthetic Yarns, by L. L. Lewis (*Rayon Textile Monthly*, July, August, September, 1930).

Relation of Air Conditions to Tobacco Curing, by J. Johnson and W. B. Ogden (*Wisconsin Agricultural Research Bureau* 110: 1-48, 1931).

An Investigation of the Bacterial Contamination of the Air of Textile Mills with Special Reference to the Influence of Artificial Humidification, by W. F. Wells and E. C. Riley (*The Journal of Industrial Hygiene and Toxicology*, Vol. 19, No. 10, December, 1937).

Air Conditioning for Clothing Research Laboratory (*Heating and Ventilating*, July, 1943, p. 69).

Controlled Air Supply for Supercharger, Carburetor and Engine Testing, by C. S. Leopold (*Refrigerating Engineering*, August, 1943, p. 85).

Air Conditioning Aids in Artillery Optical Equipment (*Heating and Ventilating*, August, 1943, p. 46).

Air Processing Needs of Plywood Plane Parts, by F. O. Jordan (*Heating and Ventilating*, April, 1944, p. 67).

Close Machine Tolerances Possible Through Temperature Control, by R. P. Dewey (*Heating and Ventilating*, April, 1944, p. 60).

Design of Low Temperature Test Cabinets, by F. Crampton Frost and T. J. Lopiccolo (*Heating and Ventilating*, July, 1944, p. 57).

Industrial Cooling as a Production Aid, by J. Partington, Jr. (*Heating and Ventilating*, April, 1944, p. 47).

Penicillin, by Lester T. Avery (*Heating and Ventilating*, March, 1944, p. 45).

Penicillin Production Depends Upon Control of Air Conditions (*Heating, Piping and Air Conditioning*, April, 1944, p. 188).

Heating, Air Conditioning and Insulation for Penicillin Production, by J. C. Siegesmund (*Heating, Piping and Air Conditioning*, August, 1944, p. 475).

Refrigeration Insures Quality of Clothing for the Army, by A. J. Mallinckrodt (*Heating and Ventilating*, March, 1944, p. 79).

CHAPTER 46

Exhausting and Conveying Systems

Classification of Systems, Hood Design Principles, Hood Suction and Indraft Velocities, Duct System Design, Air Cleaning Equipment, Resistance of Systems, Air Flow Producing Equipment, Protection Against Corrosion

IN many industrial plants some type of exhaust system designed to collect and remove dusts, fumes, mists, vapors and gases is essential in order to promote efficiency, economy, and safety of operation. Definitions of these various contaminants are included in Chapter 10, Air Contaminants.

The theory of air flow in an exhaust system in the following paragraphs *A* to *D* will be found in a publication [1] of the *American Foundrymen's Association.*

A. When the air flow producing equipment of an exhaust system is properly operated it will produce a negative pressure (below atmospheric) in the exhaust side of the system sufficient to overcome all resistance and to sustain the desired air velocity; and, further, will overcome all resistances on the discharge or positive pressure side of the system so that the air drawn through exhaust inlets will be discharged against atmospheric pressure.

B. An exhaust system is entirely dependent on a *sufficient volume* of air flowing into the exhaust inlets to catch the matter to be exhausted before such matter has an opportunity to diffuse into the general atmosphere of the work place or room.

C. The velocity of the air flowing into an exhaust inlet is usually of secondary importance and becomes an essential factor only when a certain velocity is required to overcome some force acting on the matter to be caught. The velocity within an exhaust system is only important to the extent that it shall be sufficient to convey the entrained matter and prevent it from settling or dropping out in the piping system. Velocity in terms of velocity pressure is most essential in designing a system because it is the basis upon which all calculations are made. In testing and checking a system the velocity as determined from the velocity pressure reading obtained by means of a Pitot tube is the only true indication of the exact air flow in a pipe or system.

D. The total pressure within an exhaust system is only of importance in determining the power required to operate the system. Total pressure tests do not indicate the proper functioning of an exhaust system as related to the volume and velocity of the air flowing into an exhaust inlet.

General design information is included in this chapter which is intended to relate primarily to industrial exhaust systems.

CLASSIFICATION OF SYSTEMS

In general there are two basic layouts of exhaust systems, the central and the multiple unit system. In the central system a fan is located near the center of operations with a piping system radiating to the various machines to be served. In the multiple unit system, which is sometimes employed where the machines to be served are widely scattered, or where the operations are apt to be independent or intermittent, small individual exhaust fans are located at the center of the machine groups or at each machine. The unit arrangement has the advantage of flexibility.

Exhaust systems may be classified with respect to the nature of the material to be handled by them as (*a*) those handling dust and (*b*) those handling fumes, mists, vapors and gases. There is a marked difference in design details of systems serving dust producing operations and those exhausting the more vapor-like matter even though the same basic

theories govern both classes. The type of hood and exhauster, the adaption of general ventilation, the conveying velocities and duct construction differ substantially.

Exhaust systems are also classified by the means employed to collect the material. The dust or refuse may be collected and controlled by enclosed hoods or open hoods with positive inward air movement or by diluting or exhausting the general air of the room. With some classes of machinery it is not feasible to hood the machines closely and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

Open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator, and should be placed so that the operator is in no case in the path of the exhausted material. When the hood must be placed at some distance above the machine it should be large enough to cover a large area as diffusion is usually quite rapid.

Some consideration should be given to the natural movement of the fumes. For those that are lighter than air, the hood may be over or above the machine; and where a heavy vapor, or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are sometimes preferable. In many cases there are convection currents and other atmospheric disturbances in the work room which should be given consideration. These disturbances diminish the tendency of dust and fumes to settle from the room air because of their density.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, and the removal of some dust from the machine or enclosure may be merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is required to handle only the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating, and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a current of clean air is drawn across the work in such a manner as to carry the dust or fume away from the operator and out of the work space. Another method of accomplishing the control of this type of installation, and one usually applied in the case of gases and fumes, is the dilution method. In this case sufficient clean air is introduced generally into the work space to dilute the contamination to a safe level.

HOOD DESIGN PRINCIPLES [2, 3]

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. The fan speed must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements of an efficient exhaust and collecting system are:

1. Hoods, ducts, fans, motors and collectors should be of adequate size and type.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should be placed so as not to interfere with the operation of a machine or any working part.
4. The system should do the required work with a minimum power consumption.
5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.
6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.
7. The exhaust system should be readily accessible for inspection and care.

HOOD SUCTION AND INDRAFT VELOCITIES

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry it into a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood must be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible. This will reduce the size of equipment, the power required by the system, and also the heating load requirements in winter.

Air Flow from Static Readings

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. Where the hood coefficient is known the volume of air flow through any hood may be determined from the equation:

$$Q = 4005\, f\, A \sqrt{h_t} \qquad (1)$$

where

Q = volume of air flow, cubic feet per minute.
A = area of connecting duct, square feet.
h_t = static suction measured 3 diameters from throat of hood, inches of water.
f = orifice or restriction coefficient which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. The factor f is determined from the equation:

$$f = \sqrt{\frac{h_v}{h_t}} \qquad (2)$$

where h_v is the velocity head in the connecting duct.

The *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with

TABLE 1. RATES OF FLOW THROUGH BRANCH PIPES WOODWORKING MACHINES

Pipe Diameter, In.	Air Volume, Cfm	Pipe Diameter, In.	Air Volume, Cfm
3	200	6	800
4	350	7	1100
5	550	8	1400

TABLE 2. BRANCH PIPE SIZE FOR WOODWORKING MACHINE HOODS

Type of Machine	Size, In.		No. of Branches	Minimum Diameter, In.		
	Min.	Max.		Bottom Branch	Top Branch	Others
Self feed table saw			2	5	4	
Other single saws		18	1		4	
	18		1		5	
Saws with Dado Head			1		5	
Band saws		2	2	4	4	
	2	3	2	5	4	
	3	6	2	5	5	
Disc sanders		18	1	4		
	18	28	1	5		
	26	32	2	4	4	
	32	38	2	5	4	
	38	48	3	5	4	4
Triple drum sanders		30	1	7		
	30	36	1	8		
	36	42	1	9		
	42	48	1	10		
Single drum sanders: (area in sq in.)		350	1	4[a]		
	350	700		5		
	700	1400		6		
	1400	2800		7		
Horizontal belt sanders		9	2	5	4	
	9	14	2	6	4	
Vertical belt sanders		6	1	4		
	6	9	1	5		
	9	14	1	6		
Jointers		8	1	4		
	8	20	1	5		
Single planers		20	1	5		
	20	26	1	6		
	26	36	1	7		
Tenoner			2	5	5	

[a]Not over 10 in. diameter.

TABLE 3. RATES OF FLOW THROUGH BRANCH PIPES GRINDING AND BUFFING WHEELS

Pipe Diameter, In.	Air Volume, Cfm	Pipe Diameter, In.	Air Volume, Cfm
3	225	6	900
4	400	7	1200
5	600		

TABLE 4. BRANCH PIPE SIZES FOR GRINDING AND BUFFING HOODS

Type of Wheel	Wheel Size Diameter, In.		Maximum		Branch Pipe Minimum Diameter, In.
	Min.	Max.	Width In.	Area Sq In.	
Grinding	---	9	1	30	3
	9	18	3	175	4
	18	24	4	300	5
	24	30	5	500	6
	30	36	6	700	7
Disc Grinding	---	20	---	300	4
	20	30	---	---	5
Buffing, Polishing and Scratch Brushing	---	8	2	50	3½
	8	16	3	150	4
	16	24	4	300	5
	24	30	6	600	6

respect to the hood are known. This is clearly indicated by Equation 3 which shows that the velocity at any point along the axis varies approximately inversely as the square of the distance. This formula coupled with Equation 1 should serve to indicate the velocity conditions to be expected when operations are conducted externally to the hood opening.

Design Based on Total Air Flow

Where the foregoing factors are not known, the usual method of designing an exhaust system is to base the air flow through the system on rates of flow through each hood which have been found by experience to provide adequate control. For woodworking systems the rates of flow given in Table 1, calculated on the basis of a branch velocity of 4000 fpm, are adequate for control. Using these air flow rates, Table 2 gives the size of pipe connections to be used with the more common woodworking machines. Properly designed grinding and buffing wheel hoods have been found to be adequately controlled when the rates of air flow given in Table 3, calculated on the basis of a branch velocity of 4500 fpm, are used. Table 4 gives the minimum branch pipe sizes to be used on the more common sizes of grinding and buffing wheels.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. (See Standards, Chapter 51.) The static suction requirements, which range from 1½ to 5 in. water displacement in a *U*-tube, must be followed in such states although in several instances they appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm past the lower edge of the wheel.

Controlling Air Velocities

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations velocities should not be less than 200 fpm at the point of origin. Recommended air velocities through hood openings for various processes are given in Table 5. For granite dust generated by pneumatic devices, velocities from 150 to

200 fpm, depending on the type of hood used, are recommended as sufficient for safe control [5]. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given in Equation 3.

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dust and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many

TABLE 5. RECOMMENDED AIR VELOCITIES THROUGH OPENINGS IN HOODS ENCLOSING OPERATIONS OR LOCATED OVER ZONES OF GENERATION OF DUSTS, FUMES, VAPORS AND GASES RELEASED IN CERTAIN MANUFACTURING PROCESSES

CONDITION OF GENERATION OF CONTAMINANT	MINIMUM AIR 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 conveyor transfer points, rotating mixtures, barrel filling.
Active generation	200–500	Foundry shakeout, high speed conveyor transfer points, crushers, screens.
Released with great force	500–2000	Grinding, tumbling mills, abrasive cleaning.

dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, Equation 3 may be used to determine the air velocity at any point along the axis [6]:

$$V = \frac{0.1\,Q}{x^2 + 0.1\,A} \qquad (3)$$

where V = velocity at point, feet per minute.
Q = volume of air handled, cubic feet per minute.
x = distance along axis, feet.
A = area of opening, square feet.

Velocity Contours

It is possible by use of a specially constructed Pitot tube [7] to map contours of equal velocity in any axial plane located in the field of influence. It has been found that the positions of these contours for any hood can be expressed as percentages of the velocity at the hood opening and are purely functions of the shape of the hood [8].

Further, the velocity contours are identical for similar hood shapes when the hoods are reduced to the same basis of comparison. These facts are applicable to all hood problems so that when the velocity contour distribution is known, the air flow required can be determined. Fig. 1 shows the contour distribution in two axial planes perpendicular to the sides of a rectangular hood with a side ratio of one-half. The distribution shown is identical for all openings with a similar side ratio provided the mapping is as shown in the figure. The contours, of course, are expressed as percentages of the velocity at the opening.

Low Velocity Systems

On multiple installations of the same operation it is often possible to institute a great saving in power cost by designing an exhaust system

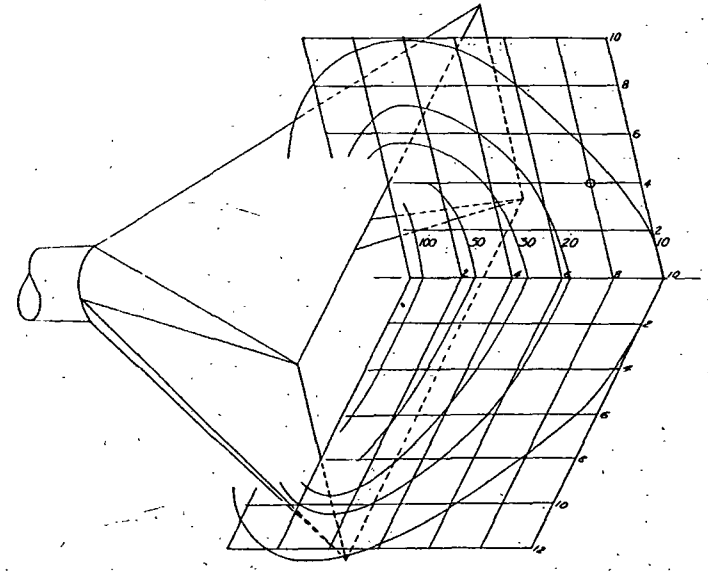


FIG. 1. VELOCITY CONTOURS FOR A RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

using low velocities in the main ducts. Such a system for use in grinding and shaping porcelain has been described [9]. In these operations, the separate machines are grouped around a central plenum chamber and exhausted by means of a low pressure fan connected to the plenum. In this case a power saving of over 90 per cent was obtained. A similar design technique [10] has been described for use in ventilating plating tanks.

Large Open Hoods

Large hoods, such as may be used for electroplating and pickling tanks, should be sub-divided so the area of the connecting duct is not less than one-fifteenth of the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distri-

bution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12 in. elevation, and wherever possible they should have side and rear aprons so as to prevent short circuiting of air from spaces not directly over the vats or tanks. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the equation:

$$Q = 1.4\,PDV \qquad (4)$$

where

Q = total volume of air handled by hood, cubic feet per minute.
P = perimeter of the tank, feet.
D = distance between tank and hood opening, feet.
V = air velocity desired along edges and surface of tank, feet per minute.

Lateral Exhaust Systems

The lateral exhaust method, as developed for chromium plating [11], is applicable in many instances in preference to the canopy type hoods. The method makes use of drawing air and fumes laterally across the top of vats or tanks into slotted ducts at the top and extending fully along one or more sides of the tanks. The slots are 1 in. wide and for effective ventilation a 2000 fpm exhaust air velocity at the slot face is advisable. In addition, the duct should not be required to draw the air laterally for a distance of more than 18 in. and the level of the solution should be kept 6 to 8 in. below the top of the tanks. If width of tank is over 18 in. a double lateral exhaust may be used with slots on both sides.

It has also been determined that a similar control may be used for tanks wider than 3 ft when the same velocity (2000 fpm) is maintained through a slot which is increased ¼ in. for every foot of width greater than 3 ft. When these slots must be extended more than 6 ft in length, some method of spreading the flow is necessary to provide even air flow distribution through the entire slot length. This can be accomplished by tapering the slot, which incidentally will add to the resistance of the system. A more economical approach is to place properly spaced vanes in the side ducts, or to branch the side ducts [12].

The flexible exhaust tube method may be advantageously used for removing dust or fumes. Flexible tubes having one end connected to an exhaust system and a slotted hood attached to the other end may be shaped at will to fit in with industrial processes without affecting the ease of operation. Efficient dust or fume removal may be had with use of relatively small exhaust volumes. This type of system may be used on swing grinders, portable grinding wheels, soldering operations, stone cutting, rock drilling, etc.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances spraying operations can be performed mechanically in wholly enclosed booths, the volatile vapors may reach injurious or explosive concentrations. At all times the concentrations of these vapors, and particularly those containing benzol, should be kept well below 100 parts per million in the breathing zone of the worker. Spray booth vapors are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located at the end of the booth opposite the opening. In front of this duct should be placed baffle plates which will cause a uniform air velocity distribution across the frontal area. The air volume should be sufficient to maintain a velocity of not less than 100 fpm over the open area of the booth (150 fpm is preferable where benzol or lead is present in the paint) and the vapors should be discharged through a suitable stack to permit dilution. It is good practice to pass the fumes or vapors through baffle type washers or scrubbers designed for efficient spray removal.

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should exceed 50 fpm when the window is fully open.

Kitchen Hoods

The length and width of kitchen hoods should be such as to extend beyond the extreme projection of the ranges, broilers, etc., over which they are installed. The minimum projection or overlap should be 12 in. Where space conditions permit, range hoods should be about 2 ft high so as to provide a reservoir to confine momentary bursts of smoke and steam until the exhaust system can evacuate the hood. Range hoods should be located as low as possible to increase their effectiveness.

A steel plate placed horizontally within a kitchen hood and located sufficiently high above the level of the bottom of the hood to permit use of a row of lights within the hood along the edge has been found effective in improving the operation. The area between the edges of the plate and the edges of the hood should be selected according to the exhaust volume and velocity required.

In general the amount of air to be exhausted from restaurant range hoods is at the rate of 100 fpm per square foot of face area. Thus, a hood 4.5 ft wide by 30 ft long has a face area of 135 sq ft, which multiplied by 100 fpm velocity results in a total air quantity to be exhausted of 13,500 cfm. In some cases, where the application is principally frying and where it is not practical to install a hood 2 ft high, it is recommended that the face velocity be increased from 100 to 150 fpm, depending on peak load conditions in the kitchen. Exhaust connections to range hoods should always be made at the top and back of hoods, and should be spaced preferably not more than 6 ft apart and be rectangular in shape with the long side parallel to the back of the hood. Exhaust openings into range hoods should be designed to maintain a velocity of 1500 to 1800 fpm.

An approved fire damper with fusible link should be (and is required by code in many states) installed in the main exhaust duct or branch adjacent to the range hood. Should there be more than one hood connected to a common duct, then the branch duct to each hood should be provided with a fire damper. Access doors should be provided at the fire

damper for purpose of inspection, cleaning, or for renewal of fusible link. All exhaust piping to range hoods, commonly called grease ducts, should be provided with tight fitting cleanout doors of adequate size to permit easy removal of grease. Some engineers use filters to advantage in hoods which are subject to grease conditions.

Hoods over steam tables should be of similar construction to range hoods. It is good practice to design such hoods with a face velocity of 60 to 70 fpm. Hoods over dishwashing machines are usually relatively small and generally 1500 to 2000 cfm per hood is allowed, which is equivalent to a velocity of approximately 100 fpm per square foot of face area. Range hoods in diet kitchens are constructed the same as restaurant range hoods but with less exhaust air per square foot of face area, depending upon the nature of the food cooked.

Hoods are not often used in private residences unless they are quite large and the consideration of expense is not important. For such residences the hoods should be designed on the same basis as diet kitchens. Most all residence kitchens can be effectively and economically ventilated by the installation of a built-in kitchen ventilator, which should be located in an outside wall and in close proximity to the kitchen range. It has been found that the capacity of the built-in kitchen ventilator should be at least 350 cfm regardless of the size of kitchen. This can be justified on the basis that the smaller the kitchen the more concentrated the heat will be thus requiring a more rapid rate of air change. Standard size built-in kitchen ventilators are generally available in three sizes, namely 350, 500 and 800 cfm. The proper size to use will depend on design conditions and available wall space.

DUCT SYSTEM DESIGN

In designing a duct system it is necessary to recognize a few fundamental principles (see also Chapter 41). Knowing the quantity of air required, the size of the duct may be computed from Equation 5:

$$A = \frac{Q}{V} \tag{5}$$

where

A = cross-section area of duct, square feet.
Q = air quantity to be handled by the duct, cubic feet per minute.
V = velocity of air, feet per minute.

Air Velocities in Ducts

Where it is necessary to transport the particulate material collected in an exhaust system, minimum carrying velocities must be maintained in the ducts preceding the collector. It has been found that good results are obtained when design air velocities in horizontal runs are not less than 2000 fpm or not greater than 5000 fpm. When the dust being carried is organic and other than wood flour, or similar material, a velocity of 2500 fpm is adequate. Approximate required conveying velocities are given in Table 6.

For duct systems wherein the air has no dust or solid load, a lower velocity is desirable, which may range from 1200 to 2000 fpm. In view of the fact that the horsepower required by a system depends directly on the resistance and the resistance is a function of the velocity, economical design requires velocities of this magnitude.

The equal friction method is generally used for designing a duct system as this insures equal resistance to air flow in all branches throughout the system (see Chapter 41). Long main ducts do not generally provide the most economical layout. Where it is necessary to ventilate a large number of machines, or machines which are widely separated, it is desirable to locate the fan at approximately the center of the system. With this arrangement it is possible to choose a fan which will deliver the required air quantity against a lower resistance pressure, and this will generally result in a horsepower saving.

When a system carrying dust is designed with an oversize main duct to allow for future extension, the air velocity may be found to be too low to carry the dust, and serious plugging may occur. In this case it is desirable to install an orifice in the end of the pipe to allow for the lower air quantity.

Construction

The interior of all ducts should be smooth and free from obstructions at joints and soldered air-tight. Other sealing mediums are permissible where soldering is impracticable.

Ducts should be constructed of galvanized sheet metal except when the presence of corrosive fumes or gases, temperatures above 400 F, or other factors would make galvanized material impractical. For the usual exhaust systems the metal thicknesses shown on Table 7 are recommended. Elbows and angles should be a minimum of two gages heavier than straight lengths of equal diameter. Hoods should be a minimum of two gages heavier than straight sections of a connecting branch.

Longitudinal joints of ducts should be lapped and riveted or spot-welded on 3-in. centers maximum. Girth joints or ducts should be made with lap in direction of air flow, with 1 in. lap for duct diameters through 19 in. and 1¼ in. lap for diameters over 19 in. Elbows and angles should have an inside or throat radius of two pipe diameters whenever possible. Large radii are recommended for heavy concentrations of highly abrasive dusts. Elbows 6 in. or less in diameter should be constructed of at least 5 sections and, if over 6 in. in diameter, of 7 sections, with angles pieced proportionally. Hoods should be free of sharp edges or burrs and reinforced to provide necessary stiffness. Transitions in mains and submains should be tapered with a taper 5 in. long for each 1 in. change in diameter whenever possible. All branches should enter the main at the large end of the transition at an angle not to exceed 45 deg or preferably 30 deg. Branches should be connected only to the top or sides of mains, with no two branches entering diametrically opposite to each other. Dead end caps should be provided within 6 in. from last branch of all mains and sub-mains. Cleanouts should be provided every 10 ft and near each elbow, angle, or duct junction in horizontal sections. Ducts should be supported sufficiently to place no loads on connected equipment and to carry weight of a system plugged with material. The

TABLE 6. 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

maximum distance between supports should be 12 ft for 8 in. or smaller ducts and 20 ft for larger ducts. Six in. minimum clearance should be provided between ducts and the ceiling, wall or floor. Blast gates for adjustment of the system should be placed near the connection of a branch to the main and means of locking gates after the adjustments have been made should be included. Rectangular ducts should be used only when clearances prevent the use of round construction. Rectangular ducts should be as nearly square as possible. The weight of metal and the lap, and other construction details, should be the equal of round duct construction having a diameter equal to the longest side. All pipes passing through roofs should be equipped with collars so arranged as to prevent water from leaking into the building.

The main trunks and branch pipes should be as short and straight as possible.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Every pipe should be kept open and unobstructed throughout its entire length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely. The passing of pipes through fire-walls should be avoided wherever possible, and floor sweep connections should be so arranged that foreign material cannot be easily introduced into them.

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always good practice and is frequently done at the expense of a reduced air velocity, it is often done where future expansion of the exhaust system is contemplated.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 2, Chapter 41. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 100 to 300 per cent of pipe diameter, the loss may be estimated from Table 8. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a center line radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately

TABLE 7. GAGES OF METALS FOR EXHAUST SYSTEMS[a]

DIAMETER OF ROUND PIPE OR GREATEST DIMENSION OF RECTANGULAR PIPE, INCHES	THICKNESS OF DUCT MATERIAL U. S. GAGE NUMBER	
	For Highly Abrasive Matter	For Other Matter
Up to 8 inclusive	20	22
Over 8 to 18 inclusive	18	20
Over 18 to 30 inclusive	16	18
Over 30	14	16

[a]Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems (*American Foundrymen's Association*, p. 53).

10 diameters long, while with a center line radius 1½ times the diameter, the resistance is apparently the same as that of seven diameters of straight pipe.

AIR CLEANING EQUIPMENT

Primary dust separators depend for their air cleaning action on centrifuging out the dust particles or on imparting to the transporting air stream a quick change in direction, accompanied by a sudden reduction in velocity. They may be applied to exhaust systems from polishing and buffing wheels and, in some cases, as a preliminary separator ahead of a final dust arrester.

Dust particles which cannot be caught in inertia type separators are usually removed by one of the following methods:

1. *Filtration.* Of the filter arresters the cloth type, either of the screen, envelope, or bag construction, is most generally used for industrial exhaust systems. Cloth arresters properly engineered for this job and handling dust for which they are suited operate at a good efficiency.

2. *Wet Collectors.* Wet collectors have been in use for many years, but only recently have been introduced for collection of industrial dust from exhaust systems. They are particularly applicable where moisture is present in the air being exhausted. The resistance is relatively low in wet collectors suitable for dust removal and remains constant for a given exhaust system.

3. *High-tension Electrical Precipitation.* High-tension electrical precipitation has been applied to removal of dust, fumes, and mists from air and other gases. (See section on Electric Precipitators in Chapter 33.)

4. *Dynamic Precipitators.* This equipment effects removal of entrained dusts by means of a specially designed fan in which the rotating element imparts a centrifugal force to the dust particles. Originally these precipitators were intended for the collection of dry dust, but recently the construction has been modified to permit the introduction of water into the dirty air intake, thus increasing its application to more kinds of dust. Consideration must be given to the type of dust handled[13].

For additional information on dust and cinders, see Chapter 33, Air Cleaning Devices.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the duct system.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. Where possible the resistance of the particular collector to be used should be obtained from the manufacturer.

TABLE 8. LOSS THROUGH 90-DEG ELBOWS

ELBOW CENTER LINE RADIUS IN PER CENT OF PIPE DIAMETER OR DEPTH	APPROXIMATE LOSS IN PER CENT OF VELOCITY HEAD
50	75
100	26
150	17
200–300	14

TABLE 9. CORROSION RESISTING MATERIALS FOR EXHAUST SYSTEMS[a]

MATERIAL	ACID[b]							
	ACETIC	CHROMIC	HYDRO-CHLORIC	HYDRO-FLUORIC	NITRIC	PHOS-PHORIC	SUL-PHUROUS	SUL-PHURIC
METALS	Dil. | Conc.	Dil. | Conc.	Dil. | Conc.	Dil. | Conc.	Dil. | Conc.	Dil. | Conc.	Dil. | Conc.	Dil. | Conc.
Aluminum	Good	Fair	Poor	No Data	Poor | Good	Poor	Poor	Poor
Magnesium and Alloys	No Data	Good | Poor	No Data	Poor | Good	No Data	No Data	No Data	No Data
Lead and Lead-Coated	Poor	Good	Poor	Poor	Poor	Poor	Good	Good | Poor
Moly Alloy (60 *Ni*—20*Mo*—20 *Fe*)	Good	No Data	Fair	No Data	Poor	Poor	No Data	Good
Monel Metal	Fair	Poor	Fair | Poor	Good[d]	Fair | Poor	Fair	Poor	Good[e] Poor
Bronze	Poor						Good	
Silicon Iron	Fair | Good	No Data	Fair	Poor	Good	Good	No Data	Good
Stainless Steel[c] (18 *Cr*-8 *Ni*)	Good	Good	Poor	No Data	Good	Poor	Good	Poor Good[e]
Enameled Steel	No Data	No Data	Good	Poor	Good	Poor	No Data	Good
MISCELLANEOUS								
Asbestos Comp			Good except against strong acids and alkalies					
Wood	Some woods are decomposed or softened faster than others.							
Rubber					Poor			Poor
Plastics	In general plastics resist weak acids and are decomposed by concentrated acid.							

[a]Standard Practice Sheet No. 115 (Division of Industrial Hygiene, New York State Labor Department).
[b]Acid mists in air are more corrosive than as liquid in storage tank. Galvanized iron not resistant to acid.
[c]Stainless steel of (24 *Cr*—10 *Ni*) fairly resistant at low temperature for *HCl* and H_3PO_4.
[d]Under most conditions.
[e]At room temperatures.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. The velocities should be found in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 8 in this chapter and Fig. 2, Chapter 41. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

EFFICIENCY OF EXHAUST SYSTEMS

The efficiency of an exhaust system depends upon its effectiveness in reducing the concentration of dusts, fumes, vapors and gases below the safe or threshold limits [14].

Too much emphasis cannot be placed on the necessity of testing exhaust systems frequently by determining the concentration of atmospheric contamination at the worker's breathing level [15]. Commonly accepted values of threshold limits for usual atmospheric contaminants will be found in Tables 3, 4 and 5, Chapter 10.

AIR FLOW PRODUCING EQUIPMENT

In any type of exhaust system some form of air flow producing equipment should be required to create the pressure necessary to cause the air

to flow through the system to the discharge stack. Natural draft should not be depended on to remove harmful or dangerous gases, fumes, mists, dusts, or other matter when it is imperative that such matter be removed from the work place or room atmosphere.

The principal types of air moving equipment are chimney exhausts, venturi ejectors, centrifugal exhaust fans, disc or propeller fans, and axial flow fans [16].

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a collector on the negative side of the fan or by lining the housing and blades with rubber.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 9. Hoods and ducts, when short, may frequently be constructed of wood and be quite effective. Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

REFERENCES

1—Tentative Recommended Good Practice Code and Handbook on the Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems, Page 21, Industrial Hygiene Codes Committee, *American Foundrymen's Association*.

2—How to Design Exhaust Hoods, by J. M. DallaValle (*Heating and Ventilating*, Series of 12 articles March, 1943 to February, 1944).

3—Industrial Exhaust Ventilation in Industrial Hygiene, by Allen D. Brandt (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 331).

4—For more detailed requirements refer to Fundamentals Relating to the Design and Operation of Exhaust Systems, Z9-1936 (*American Standards Association*). Industrial Code Bulletin Nos. 10 and 12 (New York State Labor Department). Principals of Exhaust Hood Design, by J. M. DallaValle (*U. S. Public Health Service*, 1939).

5—Control of the Silicosis Hazard in the Hard Rock Industries. I. A Laboratory Study of the Design of Dust Control Systems for Use with Pneumatic Granite Cutting Tools, by Theodore Hatch, Philip Drinker and Sarah P. Choate. (*Journal of Industrial Hygiene*, Vol. XII, No. 3, March, 1930).

6—The Control of Industrial Dust, by J. M. DallaValle (*Mechanical Engineering*, Vol. 55, No. 10, October, 1933).

7—Studies in the Design of Local Exhaust Hoods, by J. M. DallaValle and Theodore Hatch (*A.S.M.E. Transactions*, Vol. 54, 1932).

8—Velocity Characteristics of Hoods under Suction, by J. M. DallaValle (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 387).

9—Low Velocity Exhaust Systems, by Theodore Hatch (*Heating and Ventilating*, October, 1940, p. 27).

10—Tank Ventilating Power Costs Cut by Low Velocity Systems, by William B. Harris (*Heating and Ventilating*, July, 1942, p. 42).

11 Health Hazards in Chromium Plating, by J. J. Bloomfield and William Blum (*U. S. Public Health Report*, Vol. 43, No. 26, September 7, 1928).

12—New Data for Practical Design of Ventilation for Electroplating, by W. P. Battista, Theodore Hatch and Leonard Greenburg (*Heating, Piping and Air Conditioning*, February, 1941, p. 81). Ventilation of Plating Tanks, by Allen D. Brandt (*Heating, Piping and Air Conditioning*, July, 1941, p. 434).

13—The National Silicosis Conference Report (*Bulletin No. 15*, U. S. Department of Labor, February 3, 1937).

14—Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 353).

15—Keeping Dust Under Control, by John M. Kane (presented to the 31st National Safety Congress, October 28, 1942, and reprinted in part in *National Safety News*, January, 1943). The Determination and Control of Industrial Dust, by J. J. Bloomfield and J. M. DallaValle (*Public Health Bulletin 217*, 1935). Engineering Control of Air Contamination of the Working Environment, by A. D. Brandt (In Manual of Industrial Hygiene, *U. S. Public Health Service*, 1943, p. 198-266).

16—The Axial Flow Fan and Its Place in Ventilation, by W. R. Heath and A. E. Criqui (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 197).

CHAPTER 47

Drying Systems

Drying Methods, Radiant-Heat Drying, Conduction or Direct Contact Drying, Convection or Air Drying, Mechanism of Drying, Omissions in the Cycle, General Rules for Drying, Dryer Calculations, Humidity Chart, Design, Estimating Methods

DRYING, in its broader sense, refers to the removal of water, or other volatile liquid from either a gaseous, liquid, or solid material. In practice, the process of direct drying gaseous material is referred to generally as dehumidifying, or condensing, and in some cases chemicals are used in the adsorption or absorption of moisture. The subject of dehumidification is treated in Chapter 38. Drying a liquid is called evaporation or distillation. The common usage of the word *drying* refers to the removal of water or other liquid, such as a solvent, by evaporation from a solid material.

When the solid to be dried contains large amounts of free water, the actual drying process is frequently preceded by the removal of part of the water by some mechanical means, such as filtration, settling, pressing or centrifuging. Removal of as much water as possible by such methods is usually advisable, as the cost of these operations, per pound of water removed, is generally much less than by evaporation. In some drying processes the evaporation of the liquid is accompanied by a chemical change, as in the drying of paint and varnish.

DRYING METHODS

Heat must be supplied in order to dry a solid by evaporation. Since this latent heat is large compared with the specific heat of the materials, drying becomes largely a problem in heat transfer. Hence drying methods are often classified according to the method of heat transfer used, as follows:

1. Radiant-heat drying.
2. Drying by direct contact, and conduction.
3. Convection or air drying.

Radiant-Heat Drying

Drying by sun heat is still practiced where danger of rain is slight, atmospheric pollution is negligible, and sufficient time can be allowed.

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 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.

Conduction or Direct Contact Drying

Drying rolls or drums, 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.

Greatest resistance to heat transfer occurs on the air side of the material being dried. The rate of heat transfer from the surface of the heated material to the air, and hence the rate of drying, may be increased by: (a) forced convection or air circulation and (b) vacuum operation to lower the boiling point of the liquid being evaporated.

Convection or Air Drying

The circulation of heated air or other gases over the material being dried is termed convection drying. Some convection drying occurs in

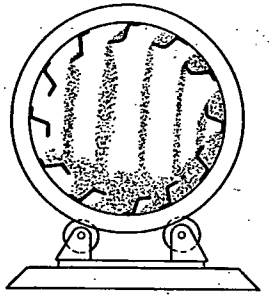


FIG. 1. SECTION OF ROTARY DRYER

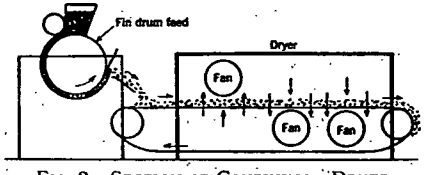


FIG. 2. SECTION OF CONTINUOUS DRYER, BLOW-THROUGH TYPE

practically all types of dryers, but if the main source of heat is from the air or gases, the dryer may be called a convection dryer. Typical forms or examples may be enumerated:

1. Rotary drum dryers, (Fig. 1).
2. Tunnel or oven dryers, batch type.
3. Tunnel dryers, conveyor type, (Figs. 3 and 4).
4. Tower or column dryers.
5. Through-circulation dryers, (Fig. 2).

In any type of convection dryer the heat transfer and hence the drying depends primarily upon the surface area of material exposed to the air and the velocity of the air over the surfaces (see Dryer Calculations, later).

A general classification of several types of dryers is given in Table 1. The chief basis of classification in this table is that of intermittent or batch operation as opposed to continuous operation. Another important basis of classification would be the method of handling the material to be dried. In an effort to secure maximum contact of the air with the product, as well as uniformity of heating, the effectiveness of cascading the material in an inclined drum or of blowing heated air through a bed of granular or pre-formed material is at once apparent, and where continuous drying at high capacity is required, these types are preferred. Extensive experimental studies on both types are available, (see References).

Simple *drying ovens* are often used for drying smaller quantities of material.

The heat and humidity supply for low temperature work up to 250 F is often steam; steam coils, either in the oven or outside, heat the air used for drying. Circulation of heated oil is used to a limited extent, but the danger of leaks is serious, for if the oil is hotter than the flash point, a fire may start if the oil is released to the atmosphere. In many cases where steam is not available, direct or indirect-fired heaters are used with gas or oil as fuel. Indirect heaters should be carefully selected from a standpoint of long life and efficiency. The heat exchange surface should be adequate in area and easily accessible for cleaning and removal. For extremely high temperatures, alloy surface may be used. With direct-fired equipment care must be used in the selection of burners and sufficient combustion space allowed to insure complete combustion of fuel. Humidity can be obtained in dryers by the use of steam spray, humidifiers, or recirculation.

For low temperature work up to 200 F ovens and dryers are commonly built of two thicknesses of insulating board (fireproof preferred), with air

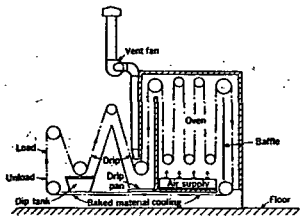


FIG. 3. SMALL PART MULTIPLE PASS OVEN

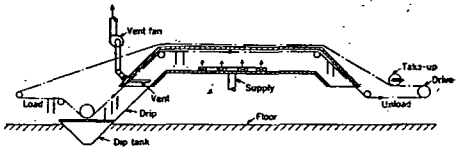


FIG. 4. INCLINED END ENAMELING OVEN

space between. As the temperature increases, materials better able to withstand the heat must be used. Metal lined ovens are easy to keep clean, and many high temperature dryers up to 1000 F are made of metal panels with insulation between. Care should be taken to avoid *through metal* (metal extending through the oven wall from inside to out). Batch type ovens are entirely closed while in use and control of air leakage is easily taken care of. In the continuous dryer where the ends are open, heat and air leakage becomes important. Warm air leaking out of the ends of ovens means a heat loss, and often the temperature and humidity outside the oven becomes unbearable. For this reason, inclined or bottom entry ovens are used, as the warm air leakage can be more easily controlled. See Figs. 3 and 4.

MECHANISM OF DRYING

The modern theory of drying may be summed up as follows: Assuming uniform velocity and distribution of air at a constant temperature and humidity over the surface to be dried, the drying cycle will be divided into two distinct stages:

1. Constant rate period.
2. Falling rate period.

The *constant rate period* occurs while the material being dried is still very wet, and continues as long as the water in the material comes to the

TABLE 1. DRYERS FOR EVAPORATION OF WATER

TYPE	KIND	MATERIALS HANDLED	MEANS OF HANDLING	TEMP. RANGE F DEG	HEAT SUPPLY	USES AND REMARKS
Batch or Intermittent	Compartment	Paper, Leather, Yarns, Lumber, Foodstuffs	Suspended, Truck, Tray	80 to 180	Steam Coils, Air, Electricity	When production does not warrant continuous drier
	Agitated	Chemicals too sticky for Rotary Drier	Shoveled into Drum or Pan	100 to 330	Water, Steam Jacketed, may have Vacuum on top	Where dust must be saved
	Vacuum	Chemicals, Explosives, Pharmaceuticals, Food Products	Tray, Basket, Tumbling Drum	80 to 300	Water, Steam	Cost of operation high, for expensive materials
	Tunnel	Ceramics, Chemicals, Lumber, Food Products	Truck, Tray, Belt	100 to 350	Steam Coils, Air, Electricity, Products of Combustion	For high production
Continuous	Rotary	Bulk	Cascades through	80 to 500	Air, Steam, Products of Combustion	Where material will stand rough handling and is not subject to balling up
	Drum	Liquids, Slurries	Flowed on Drum, Dry Material Scraped off	to 310	Steam, may have Vacuum on Top	Hygroscopic materials dried with vacuum, and packed immediately
	Cylinder	Paper, Textiles, Chemicals	Continuous Sheets, Endless Chain Belt	to 350	Steam inside of Drum	Where material comes in sheets or rolls, and will stand direct contact with heating surface
	Festoon	Paper, Chemicals	Continuous Sheets, Suspended on Metal Screens	to 200	Air, Steam Coils	Where one side cannot come in contact with supports until dry
	Tower or Column	Grains, Sand	Falls through by Gravity	125 to 250	Air, Steam Coils	Where headroom is available
	Spray	Solutions over 30% Solids	Sprayed into Chamber	120 to 350	Air, Products of Combustion	Drying is almost instantaneous
	Induction	Metals, for removal of traces of Water	Placed in High Frequency Field	to 400	Electricity	Where heating of metal from inside out is important

surface so rapidly that the surface remains thoroughly wet, and evaporation proceeds at a constant rate, precisely as from a free water surface. The material tends to assume a temperature corresponding to the wet-bulb temperature of the surrounding air. But the actual temperature is often slightly higher, due to radiation and conduction from dry surfaces adjoining the material. The constant rate period continues until a time is reached when the moisture no longer comes to the surface as fast as it is evaporated. This point is called the *critical moisture content* in the drying process.

As the drying proceeds, a period of *uniform falling rate* is entered. During this period, the surface of the material is gradually drying out, and

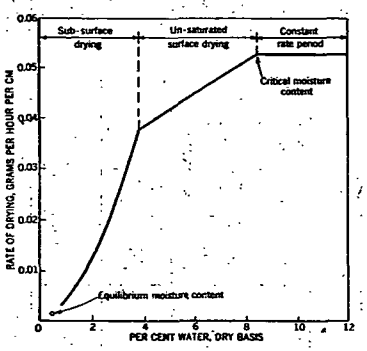


FIG. 5. RATE OF DRYING OF WHITING SLAB

the rate of drying falls as the remaining wet surface decreases in area. This period is also known as unsaturated surface drying.

As drying continues, the surface is completely dry and the water from the interior evaporates and comes through the surface as vapor. As the plane of water recedes, the diffusion of the vapor becomes more difficult and hence the period is known as *varying falling rate period*, or sub-surface drying.

Drying ceases when the *equilibrium moisture content* has been reached. The final moisture content of the product depends on the relative humidity of the air in contact with it. Equilibrium is established when the vapor pressure of the moisture in the air and the vapor pressure of the moisture in the material are equal. The equilibrium moisture content varies with the hygroscopic properties of the material, (see table of Regain of Hygroscopic Materials, Chapter 45).

The drying of a slab of whiting is shown in Fig. 5 and illustrates the principles referred to previously. The factors affecting the variations of drying rates during the periods mentioned are outlined in Table 2.

Omissions in the Cycle

Many solids, such as lumber, are so dry at the beginning of the drying operation that the constant rate period of free surface evaporation does not occur. Frequently the surface of the material is dry enough so that no surface drying can take place, in which case only the final stage of sub-surface drying is involved. In other instances, the critical moisture content of a wet solid is sufficiently low that sub-surface drying starts almost immediately after the conclusion of the constant rate period. Thus the intermediate state of unsaturated surface drying does not occur and the drying is of the sub-surface type during practically the whole of the falling rate period. With other kinds of material, particularly thin sheets, such as newsprint paper, sub-surface drying may occur at such a low moisture content that it is not encountered in commercial work, the falling rate period being confined in practice to unsaturated surface drying.

TABLE 2. FACTORS INFLUENCING DRYING

FACTOR	DRYING PERIOD	
	Constant Rate, Unsaturated Surface	Sub-Surface
Temperature	Increase in temperature increases drying rate	Increase in temperature increases drying rate, because with decreased viscosity, capillary flow is increased.
Humidity	Drying rate increases as humidity is decreased	No effect until equilibrium content is reached; drying then ceases
Air Velocity	Drying rate varies approximately as the 0.6 power of the velocity	No effect
Air Direction	Drying rate increases, the more nearly the air blows perpendicular to surface; for dead air film becomes thinner	No effect
Thickness of Material	Drying rate is not affected by the thickness	Drying rate varies inversely as the square of the thickness

GENERAL RULES FOR DRYING

Temperature

The highest temperature possible should be used because of faster drying and smaller requirements for ventilation. The amount of moisture that can be carried by a pound of air increases rapidly with rise in temperature as shown in the humidity chart of Fig. 6. Too high a temperature may cause spoilage of materials; many materials calcine or change their chemical properties if heated too hot; gypsum and glauber salts lose some of the chemically combined water, fall apart, and change their chemical properties. Too high or rapid rise in temperatures in drying lumber or ceramics may create a liquid vapor tension within the material so high that the cells explode, causing permanent injury to the fiber. If too high a temperature is used on some chemicals, they begin to react exothermally; a temperature rise and chemical action from within will burn the materials, *e.g.*, bakelite products, gunpowder, etc. During the constant rate period of drying, the material heats only to the wet-bulb temperature of the surrounding air, consequently high temperatures will not injure the material in this stage.

Humidity

Moisture in the drying air may be very important. Many materials tend to case-harden, dry on the outside, forming a skin which retards the moisture flow from the inside to the surface, or stops it completely, and so

increases the drying time very much or causes a change of the physical properties of the material. It is often necessary to add humidity to the air in the initial stage of drying. Lumber case-hardens, cracks, and warps if the outside is dried too fast. Ceramics crack if not heated through before drying commences. Elastic materials warp or crack if not evenly dried. Many paints case-harden if not dried under high humidity.

On the other hand, in the case of those materials whose physical or chemical properties require that they be dried at relatively low temperatures, high humidity tends to retard drying in the first stage and may even stop it altogether in the final stage. Where drying temperatures below 120 to 140 F are used, the drying rate may be highly dependent on atmospheric humidity conditions. In such instances it is often desirable to dehumidify the air entering the dryer during periods of high atmospheric humidity; where a high degree of uniformity is required, it is often possible to secure complete independence of atmospheric conditions by recirculating the air in a closed system which includes a suitable dehumidifier. For this purpose absorptive dehumidifying systems have the advantage of accomplishing the desired reduction of humidity without appreciably elevating or lowering the dry-bulb temperature of the air; for this reason after-cooling is not required, and reheating is reduced to a minimum. Dehumidifying systems are described in Chapter 38.

Air Circulation

As noted under Mechanism of Drying, air velocity is more important in the first two stages of drying than in the last, and for this reason zone drying in continuous dryers is frequently considered. It permits accurate regulation of temperature, humidity, and velocity in the different zones. High velocity results in more rapid drying, more even distribution of temperature and consequently more even drying in the first period. Too high a velocity may be detrimental because of excessive power needed for creating it, or because the material may blow away if it is light and fluffy. In the drying of paints, varnishes, and enamels, high velocity or improper distribution of the air, even with the use of filters, may cause dust already in the dryer to be blown against the material, ruining the finish.

DRYER CALCULATIONS

The fundamental calculations for the design and performance of dryers are based on the thermodynamics of air and water mixtures treated in Chapter 3, and the fundamentals of heat transfer treated in Chapter 5. For the humidity calculations a high-temperature psychrometric chart is given in Fig. 6. In addition to the fundamental heat transfer calculations of radiation, conduction and convection, the heat losses through the walls of the dryer will be computed by the methods illustrated in Chapter 6 and Chapter 41. Data on radiation calculations are given in Chapter 31.

Where products of combustion are used directly in a dryer, a knowledge of the properties of fuels and combustion products is important. Data on fuels and combustion are given in Chapter 16. For determining the heat available in products of combustion, a specific heat of 0.25 Btu per (pound) (Fahrenheit degree) may be used.

The calculations for drying during the constant-rate period are different from those applying to the falling-rate period.

Constant-Rate Period

The rate of drying by air passing over a wet surface is directly proportional to the vapor pressure difference, and also proportional to the

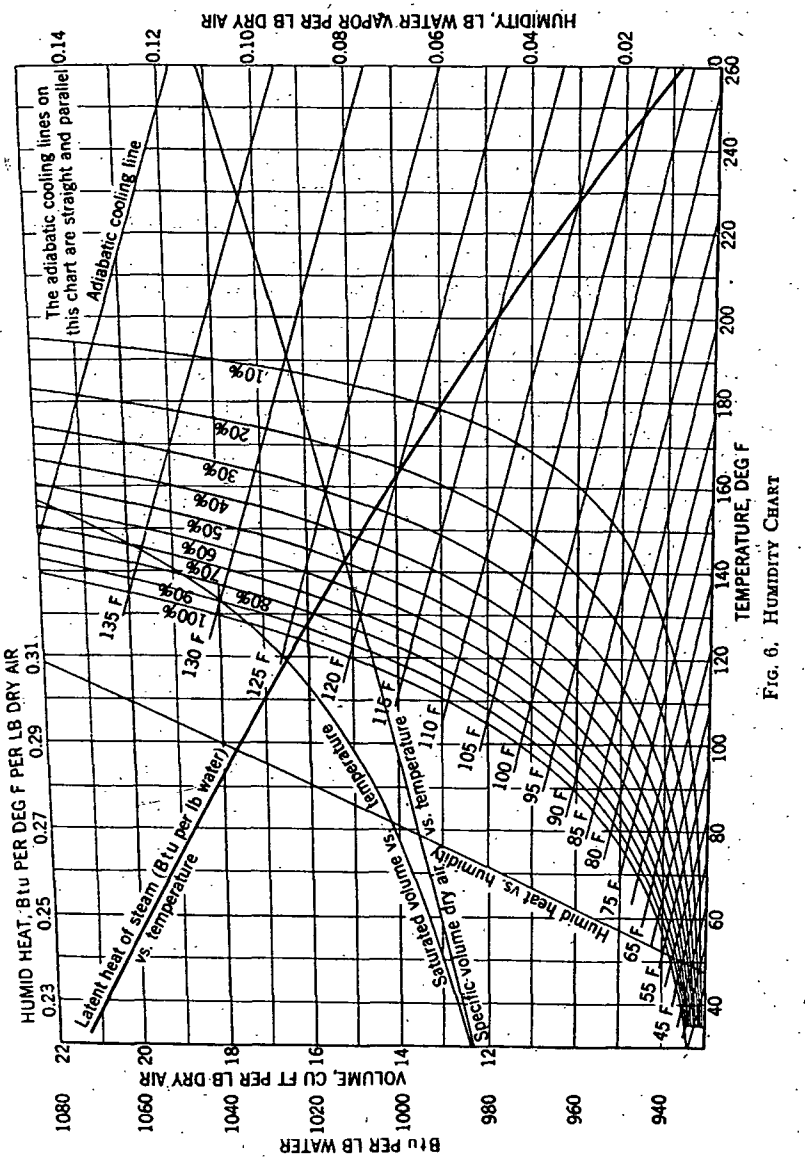


FIG. 6. HUMIDITY CHART

0.8 power of the air velocity. For practical calculations the wet surface is assumed to attain the wet-bulb temperature of the air passing over it, and evaporation takes place at constant rate under equilibrium conditions. The equation may then be expressed in three forms:

$$R = C\,A\,V^{0.8}\,(\Delta P) \quad (1)$$

$$R = C'\,A\,V^{0.8}\,(\Delta H) \quad (2)$$

$$R = C''\,A\,V^{0.8}\,(\Delta T) \quad (3)$$

where

R = rate of drying during constant-rate period, pounds of moisture per hour.
A = area of bed or material in contact with air, square feet.
V = air velocity over material, feet per minute.
ΔP = difference between vapor pressure at wet-bulb (surface) temperature and at dew-point of air.
ΔH = difference between humidity ratio of saturated air, at the surface temperature, and the actual humidity ratio of the air stream, pounds of water per pound of dry air.
ΔT = difference between dry-bulb and wet-bulb temperatures of air, *i.e.*, the wet-bulb depression.
C, C', C'' = proportionality constants (for numerical values consult references).

These equations are useful mainly for computing the effects of changes in operating conditions, such as changes in air velocity, air temperature, humidity and surface area. The equations assume that the material is in equilibrium at the wet-bulb temperature of the air. If equilibrium has not been reached, or if heat is being added to the charge by radiation or conduction, such conditions must be taken into account. For large tray dryers or continuous surfaces, the logarithmic mean difference should be substituted for the simple difference in ΔP, ΔH and ΔT.

When the constant of proportionality is known for a given set of conditions, Equations 1, 2 or 3 may be applied for basic design, as illustrated in *Example 1*.

Example 1. Compute the rate of drying of a granular material initially 35 per cent moisture (dry basis), if the material is spread in trays and is to be dried by blowing air horizontally over the surface at 1000 fpm. The air is 140 F dry-bulb, 90 F wet-bulb. Density of the dry material is 85 lb per cubic foot, and the drying constant C'', in Equation 3, has been found to be about 1/25,000. Find the size of dryer for a capacity of one ton per hour (dry basis), and the time required for drying each batch from 35 to 10 per cent moisture content, if the material is spread in trays, in a layer one inch thick. (The critical moisture content of the material is below 10 per cent, hence the drying is at constant rate.)

Solution: Assume that the surface of the material attains the wet-bulb temperature of the air, then $\Delta T = 140 - 90 = 50$ F. The rate of drying by Equation 3 is:

$$R = C''\,A\,V^{0.8}\,\Delta T = \frac{1000^{0.8} \times 50}{25{,}000} = 0.50 \text{ lb of water per (hour) (square foot of surface)}.$$

The total water evaporated per square foot of surface is:

$$W = \frac{85}{12}\,(0.35 - 0.10) = 1.77 \text{ lb (per batch)}.$$

Then the time required per batch is:

$$T = \frac{1.77}{0.50} = 3.54 \text{ hr.}$$

The size of dryer required to dry the material at the rate of one ton of dried material per operating hour will be:

$$A = \frac{2000 \times 3.54}{85/12} = 1000 \text{ sq ft, total area of trays.}$$

Falling-Rate Period

The *critical moisture content* marks the end of the constant-rate period and the beginning of the falling-rate period. This falling rate may be due to the fact that the surface is no longer completely wetted, or it may result from a condition in which the moisture cannot reach the surface as fast as it can be evaporated. When this high resistance to capillary flow and diffusion is the governing factor in the drying process, the time of drying increases rapidly with the thickness of the material. During constant-rate drying the time required is directly proportional to the thickness of the bed (see *Example 1*), while the time required for drying during the falling-rate period is often proportional to the square of the thickness of the material.

Actual calculations of drying during the falling-rate period are not highly satisfactory because of the number of variables. It has been demonstrated empirically that the rate of drying is approximately proportional to the free water content of the material.

An approximate value of the critical moisture content which marks the beginning of the falling-rate period may be obtained from Table 3.

TABLE 3. CRITICAL MOISTURE CONTENT FOR VARIOUS MATERIALS[a]

MATERIAL	CRITICAL MOISTURE CONTENT, PER CENT
Sand	3 to 20
Clay; soil; pigments	5 to 40
Paper, book or coated	30 to 40
Paper, newsprint	60 to 70
Paper pulp and pulp-boards (wallboard)	75 to 150
Leather (heavy)	90 to 120

[a]The critical moisture content is expressed as weight of water in per cent of weight of dry material.

DESIGN

In all drying problems, data regarding temperatures, time, and humidity must be obtained by experiment or previous experience. Experiments are best performed at the actual temperatures, humidities, and velocities to be used in the full sized dryer, and with full size samples.

The following nomenclature and explanation of terms will be used in the discussion of design calculations:

H = humidity ratio of air, pounds of water vapor per pound of dry air.
G = pounds of dry air supplied to the dryer per unit of time.
S = pounds of stock dried per unit of time in a continuous dryer.
S' = pounds of stock charged per batch to a discontinuous dryer.
Θ = time.
Q = total heat supplied to the dryer.
t = air temperature.
t' = stock temperature.
t'' = average stock temperature over short time interval, in a batch dryer.
t_w = wet-bulb temperature.
s' = specific heat of the stock.
B = total radiation and conduction losses per unit time.
w = pounds of water per pound of dry stock.

r = heat of evaporation of water.

s = humid heat of air, *i.e.*, heat necessary to raise 1 lb of dry air + H lb of steam 1 F.

Subscript (1) designates conditions at the point where the material in question (air or stock) enters, and (2) where it leaves the dryer.

Air dryers may be divided into two classes, those in which *all moisture* evaporated from the stock *leaves the dryer as vapor* in the effluent air, and those in which *part or all* of the moisture *is condensed* from the air *in the drying equipment itself.* In any continuously operating dryer of the first type the relation between moisture content of the stock and quantity of air required for the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S(w_1 - w_2) \tag{4}$$

In discontinuous dryers, *e.g.*, compartment dryers, the drying operation is given by the equation:

$$G\,(H_2 - H_1) = S' \frac{dw}{d\Theta} \tag{4a}$$

In the continuous dryer, the heat consumption per unit time is:

$$\frac{Q}{\Theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)\,(H_2 - H_1) + S(t'_2 - t'_1)\,(s' + w_1) + B \tag{5}$$

Equation 5 assumes continuity of operation. For charge or batch operations, the total time of the drying cycle may be broken up into a number of periods, sufficiently short so that over each period average values of t, t' and H may be employed provided the third term of the right hand member of the equation is modified to read:

$$S'\,(t''_2 - t''_1)\,(s' - w_1)$$

and in the second term t'_2 be replaced by

$$\frac{t'_1 + t''_2}{2}$$

Theoretically these periods should be very short and the equation integrated. Practically, the error introduced by using a small number of long periods and employing average values of the variables over each is not serious. The evaluation of Equation 4a may be approximated in a similar manner.

The first term of the right hand member of Equation 5 represents heat lost as sensible heat in the effluent air. In many drying operations this becomes excessive. Each pound of air supplied should remove the maximum amount of moisture. This is best accomplished by bringing the air into contact with the stock with sufficient intimacy so that the air leaving the dryer is saturated, or nearly so. Counter-current as against parallel flow of air and stock gives rise to optimum operating conditions, resulting in a minimum quantity of air required (G), and a corresponding minimum loss, as sensible heat, in the exit air. Similarly, continuous operation is superior to intermittent operation.

Despite the fact that the sensible heat loss increases with the rise in temperature of the air, the percentage of heat lost from this source decreases, provided the increase in moisture carrying capacity of the air, due to high temperature, is actually utilized. To secure maximum thermal efficiency in drying, a high drying temperature and high saturation of the outlet air are imperative.

Ventilation Phase

The technique of attack of the *ventilation phase* of a drying problem is best made clear by an illustration. Assume that a material containing 40 per cent moisture is to be dried until this quantity of moisture is reduced to 5 per cent by weight. The material will stand an air temperature of 150 F and it is possible to provide sufficiently good contact between the material and the drying air so that the effluent air can be brought up to 50 per cent humidity at 150 F. The dryer is to use room air, the temperature and humidity of which may be assumed to average 70 F and 50 per cent. A counter-current dryer will be employed and the air in this dryer will be kept at a substantially constant temperature of 150 F by heaters thermostatically controlled. The stock enters at 70 F, rises quickly to the wet-bulb temperature of the air, with which it is in contact, and, as has been found experimentally, remains at this wet-bulb temperature until the moisture content has fallen to 20 per cent. From this point its temperature rises progressively as it dries. In this range

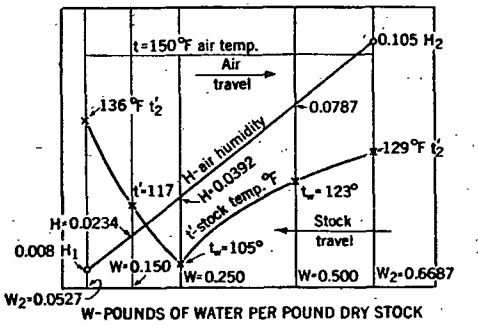


FIG. 7. TEMPERATURE HUMIDITY RELATIONS IN A DRYER

the difference in temperature between stock and air, divided by the wet-bulb depression, may be assumed proportional to the moisture content.

The moisture content of the entering stock, in the units here employed, is:

$$w_1 = \frac{40 \text{ per cent water}}{60 \text{ per cent dry stock}} = 0.6667;\ w_2 = \frac{5 \text{ per cent water}}{95 \text{ per cent dry stock}} = 0.0527$$

$w_1 - w_2 = \Delta w = 0.614$ lb water evaporated per pound of dry stock. Since the air leaving the dryer is 50 per cent saturated at 150 F from Fig. 6, $H_2 = 0.105$. Similarly, $H_1 = 0.008$, corresponding to 50 per cent humidity at 70 F. Consequently $H_2 - H_1 = \Delta H = 0.097$ lb water evaporated per pound dry air.

An analysis of Equation 4 shows that (H) is linear in w. Hence, one can construct on Fig. 7, the line marked (H) being drawn connecting the initial and final points just computed.

Since the air leaving the dryer has a temperature of 150 F and a humidity of 0.105, Fig. 6 shows that its wet-bulb temperature is 129 F. This is plotted at the right hand side of Fig. 7. Since the stock maintains a wet-bulb temperature down to 20 per cent moisture, where $w = 0.25$, the corresponding humidity can be computed by the use of Equation 4 or by reading directly from the diagram, the value being 0.0392. Fig. 6 shows that the corresponding wet-bulb temperature is 105 F. Any

intermediate point on the wet-bulb temperature curve can be calculated similarly. The points for $w = 0.5$ are shown in Fig. 7.

Below the point, $w = 0.25$, the temperature of the stock begins to rise appreciably above the wet-bulb temperature. Its temperature at any given point in this range, for example at $w = 0.15$, may be computed as follows: At this point, $H = 0.0234$ (from Equation 4) and from Fig. 6, $t_w = 95$ F. Hence the wet-bulb depression, $t - t_w = 150 - 95 = 55$ F.

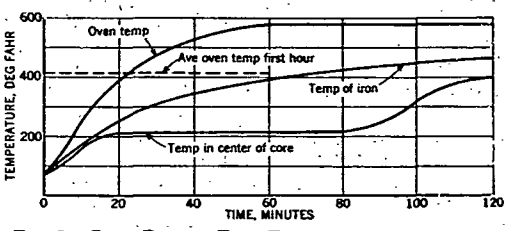


FIG. 8. CORE DRYING TIME TEMPERATURE RELATIONS

The assumption made regarding the relation between stock temperature and moisture content in this range may be formulated:

$$\frac{\Delta t'}{t - t_w} = \frac{w}{0.25}$$

At the point $w = 0.15$, $\Delta t' = 33$ F, $t' = 117$ F. The temperature of the stock leaving the dryer, similarly computed, is 136 F.

Fig. 7 thus computed gives in graphical form the information as to the temperature humidity relationships in the dryer. The air requirements can be computed by Equation 4. Thus, per 100 lb of dry stock, it is necessary to supply 633 lb of dry air. Furthermore, since from Fig. 6 it is seen that the volume of 50 per cent saturated air at 70 F is 13.55 cu ft per pound, 8580 cu ft of room air must be supplied per 100 lb dry stock. Similarly, since the volume of 50 per cent saturated air at 150 F is 18.0 cu ft per pound, the volume of hot wet air discharged from the dryer is

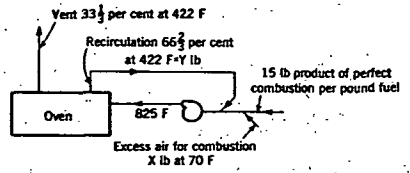


FIG. 9. CORE DRYING DIAGRAM OF COMBUSTION PRODUCTS AND AIR

11,400 cu ft per 100 lb of dry stock. Finally, the heat required for the dryer, as a whole or to any section of it, may be computed from Equation 5.

High Temperature Dryer

In the design of a high temperature dryer unit a method of approach to the necessary calculations involved is outlined as follows:

Example 2. Cores 4 and 5 in. thick are to be dried by heating to a temperature at 400 F. An intermittent type box oven is to be used, size 12 x 14 x 10 ft with 856 sq ft surface, having an average heat transfer of 0.3 Btu per (square foot) (degree) (hour).

Drying time as determined by test is 2 hr (Fig. 8). Cores weighing 6 tons, and 15-ton steel plates, trucks etc. are delivered to the dryer at 70 F. The oven is heated by an external heater; the products of combustion and 66⅔ per cent recirculated air will be delivered to the oven at 825 F. The fuel oil used has 19,980 Btu gross and 18,830 Btu per pound net heating value, weighing 6.75 lb per gallon and having 15 lb product per pound fuel for perfect combustion. Cores consist of 91 per cent sand, 3 per cent oil binder, and 6 per cent water.

Solution. Heat required per ton of cores:

	Lb Material × Temp. Rise × Sp. Ht.	=	Btu
Sand	0.91 × 2,000 × (400 − 70) × 0.2	=	120,120
Binder	0.03 × 2,000 × (400 − 70) × 0.4	=	7,920
Water heating	0.06 × 2,000 × (212 − 70) × 1.0	=	17,040
Water evaporation	0.06 × 2,000 × 970 (Fig. 6)	=	116,520
Water superheating (approx. 50 per cent reaches 575 F) = 0.5 × 0.06 × 2,000 × (575 − 212) × 0.45		=	9,800
Total Heat			271,400 Btu

Heat in 1 lb fuel oil	=	18,830 Btu
Heater Loss (10 per cent)	= 1883	
Duct Loss (5 per cent)	= 942	2,825 Btu
Btu available to heat oven		16,005 Btu

Heat content of gases in 1 lb fuel oil at 825 F is 205 Btu

Sensible heat in products of perfect combustion = (15 lb + 205) = 3,075 Btu

Btu to heat air X and Y (Fig. 9). = 12,930 Btu

$$Y (S_{825} - S_{422}) + X (S_{825} - S_{70}) = 12,930 \qquad (6)$$

$Y = 2 (X + 15)$ for 66.7 per cent recirculation

where

S = heat content of air at temperature noted taken from Fig. 6.

(Recirculation and exhaust contain water vapor, products of combustion, and a greater portion of air. Heat capacities of all vary so little that they have all been assumed to be air).

$S_{825} - S_{422} = 190 - 91 = 99$

$S_{825} - S_{70} = 190 - 8.6 = 181.4$

Substituting values of Y, H, etc. in Equation 6,

$(2 X + 30)\ 99 + 181.4\ X = 12,930$

$X = 26.3$ lb excess air.

$Y = 82.6$ lb recirculating air.

Total = 26.3 + 82.6 + 15 = 123.9 lb air and products of combustion circulated per pound fuel burned.

Heat in air exhausted from oven at 422 F per pound fuel burned = $0.333 \times 123.9 \times (S_{422} - S_{70}) = 41.3\ (91 - 8.6) = 3,400$ Btu.

Btu available for heating material = 16,005 − 3,400 = 12,605 Btu per pound fuel.

Fuel used in first hour = 2,180,470 ÷ 12,605 = 173 lb = 25.6 gal.

During the second hour the heater capacity will be much greater than required. If an automatic oven temperature control operates on the oil supply, the delivery temperature of the air entering the oven and the quantity of oil burned will decrease, the air supply being constant.

Heat in air exhausted = $41.3\ (S_{825} - S_{70}) = 41.3\ (127 - 8.6) = 4,880$ Btu per pound fuel.

Heat available for heating material = 16,005 − 4,880 = 11,125 Btu.

Fuel used in second hour = 1,072,008 ÷ 11,125 = 96.5 lb oil = 14.3 gal.

Total oil used per load = 25.6 + 14.3 = 39.9 gal.

HEATING LOAD FIRST HOUR

	HEATED TO		BTU
Sand	212 F	$\frac{142}{330} \times 120{,}120$ =	51,688
Binder[a]	212 F	$\frac{142}{330} \times 7{,}920$ =	3,408
Water	212 F	=	17,040
Evaporation	66.7%	$0.667 \times 116{,}520$ =	77,680
Superheat	66.7%	$0.667 \times 9{,}800$ =	6,530
Total Per Ton			156,346
For 6 ton		$6 \times 156{,}346$ =	938,076
Steel plates	390 F	$320 \times 30{,}000 \times 0.12$ =	1,152,000
Radiation[b]	422 F Avg	$352 \times 856 \times 0.30$ =	90,394
Total			2,180,470

HEATING LOAD SECOND HOUR

Sand	400 F	$\frac{188}{330} \times 120{,}120$ =	68,432
Binder[a]	400 F	$\frac{188}{330} \times 7{,}920$ =	4,512
Water			
Evaporation	33.3%	$0.333 \times 116{,}520$ =	38,840
Superheat	33.3%	$0.333 \times 9{,}800$ =	3,270
Total Per Ton			115,054
For 6 ton		$6 \times 115{,}054$ =	690,324
Steel plates	460	$70 \times 30{,}000 \times 0.12$ =	252,000
Radiation[b]	575	$505 \times 856 \times 0.30$ =	129,684
Total			1,072,008

[a] Binder oxidizes and liberates heat, which is neglected in this calculation.

[b] Average value of coefficient is less than 0.3 because oven is not up to 575 F. This is neglected. 422 F is arrived at by taking area under curve as compared to area under 575 F ordinate.

ESTIMATING METHODS

Values based on practical experience are available for rough estimating of drying problems. The temperature will drop approximately 8.5 F per (grain of water evaporated) (cubic foot of air, measured at 70 F) or approximately 0.62 F per pound of air at any temperature. Air will drop 55 F per cubic foot for each Btu extracted. Generally air will absorb from 2 grains to 5 grains per cubic foot of air in one passage through an air dryer, depending on the temperature and the degree of contact with the material. The amount of steam required to evaporate a pound of water will vary from 1.5 lb to a more usual figure of from 2.5 to 3 lb of steam per pound of water evaporated.

BIBLIOGRAPHY

Drying Apparatus, by H. C. Russell (A.S.H.V.E. TRANSACTIONS, Vol. 18, 1912, p. 76).

Commercial Drying Apparatus, by L. P. Dwyer (A.S.H.V.E. TRANSACTIONS, Vol. 22, 1916, p. 479).

Artificial Drying with Special Reference to the Use of Gas, by G. C. Shadwell (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 231).

Drying by Evaporation, by F. R. Still (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917, p. 255).

High Temperature Drying, by Burt S. Harrison (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 7).

The Temperature of Evaporation, by W. H. Carrier (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 25).

Address on Dehydration, by H. C. Gore (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 323).

Food Dryers and the Use of School Houses for Drying, by W. L. Fleisher (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 339).

Memorandum Concerning Fruit and Vegetable Dryers, by Alfred S. Kellogg (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918, p. 359).

Progress in the Dehydration Industry, by C. E. Mangels (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 99).

Dehydration, by Ralph H. McKee (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 105).

Commercial Dehydration, by J. E. Whitley (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 551).

Drying of Fruits and Vegetables, by Ray Powers (A.S.H.V.E. TRANSACTIONS, Vol. 27, 1921, p. 241).

Drying as an Air Conditioning Problem, by A. W. Lissauer (A.S.H.V.E. TRANSACTIONS, Vol. 27, 1921, p. 251).

A Chronological Survey of Drying and Driers, by J. E. Bolling (A.S.H.V.E. JOURNAL, Vol. 27, October, 1921, p. 715).

Dehydration and Freshening of Codfish, by Henry W. Banks, 3D (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 143).

Construction of Farm Dehydrators in California, by A. W. Christie and G. B. Ridley (A.S.H.V.E. JOURNAL, Vol. 29, 1923, p. 687).

Study of Air Velocity and Temperature in Vegetable Dehydration, by A. W. Christie and K. Matsumoto (A.S.H.V.E. JOURNAL, Vol. 33, June, 1927, p. 381).

Air Conditioning and Food Dehydration (*Heating and Ventilating*, December, 1942, p. 35).

Adiabatic Drying of Hygroscopic Solids, by A. M. McCready and W. L. McCale (*Transactions, American Institute Chemical Engineers*, 1933).

Limitations of Diffusion Equations in Drying, by O. A. Hougen, H. J. McCauley and W. R. Marshall (*Transactions, American Institute of Chemical Engineers*, 1940).

Factors Influencing the Performance of Rotary Dryers, by C. F. Prutton and C. O. Miller (*Transactions, American Institute of Chemical Engineers*, Part 1, February, 1942; Part 2, August, 1942).

Drying of Solids by Through Circulation, by W. R. Marshall and O. A. Hougen (*Transactions, American Institute of Chemical Engineers*, 1942).

Factors that Influence Drier Performance, by A. Weisselberg (*Chemical and Metallurgical Engineering*, August, 1932).

Typical Dryer Calculations, by O. A. Hougen (*Chemical and Metallurgical Engineering*, January and March, 1940).

Symposium on Drying, Articles by W. K. Lewis, W. H. Carrier, A. E. Stacey, Jr. and R. S. Fleming, R. G. Metz, G. B. Ridley, C. O. Lavett, D. J. Van Marle (*Journal Industrial Engineering Chemistry*, May, 1921).

Principles of Drying Lumber and Humidity Diagram, by H. D. Tiemann (*Forest Service Bulletin* 104, 1912).

The Drying of Solids, by T. K. Sherwood (*Bulletin Massachusetts Institute of Technology*, Nos. 237, 247 and 258).

Air Conditioning and Engineering, American Blower Co., 1935.

Drying in Industrial Plants, by J. O. Ross.

Elements of Chemical Engineering, by Badger and McCabe (McGraw Hill Co., 1931).

Fan Engineering, Buffalo Forge Co.

Die Trockentechnik, by M. Hirsch (Julius Springer, Berlin, 1932).

Drying (*Kent's Mechanical Engineers Handbook*).

Drying, by W. H. Carrier (*Marks' Mechanical Engineers Handbook*).

Drying (*Perry's Chemical Engineers Handbook*, 1941).

Kiln Drying of Lumber, by A. Koehler and R. Thelen, New York, 1926.

Kiln Drying of Lumber, by H. D. Tiemann (Lippencott, 1920).

Modern Drying Machinery, by H. B. Grenshaw, London, 1926.

Principles of Chemical Engineering, by Walker, Lewis, McAdams (Chapters on Evaporation, Humidity and Drying, McGraw Hill Co.).

CHAPTER 48

Transportation Air Conditioning

Railway Passenger Car Ventilation, Method of Air Distribution, Air Cleaning, Winter and Summer Air Conditioning, Humidity and Temperature Control, Summer Air Conditioning for Buses and Automobiles, Airplane Air Conditioning

THE principles of air conditioning used in connection with stationary applications such as stores, restaurants, hospitals, theaters, and homes are in general applicable to such mobile applications as railway passenger cars, passenger buses, automobiles, and ships. However, the equipment used for these mobile applications, differs from that used for stationary purposes in that it must meet additional requirements. Especially important are the features of compactness with the retention of ready accessibility for quick inspection and servicing, and low weight. Freedom from vibration which could be transmitted to the supporting vehicle and thus to the passengers is essential. (For Air Conditioning of Ships see Chapter 49.)

RAILWAY PASSENGER CAR VENTILATION

In non-air-conditioned cars, ventilation is accomplished by exhaust fans, roof ventilators, and open doors and windows. This practice provides an ample supply of outside air but does not prevent the entrance of smoke, cinders, and other dirt.

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 the average car from 2000 to 2500 cfm should be circulated by the air conditioning unit. Some of this air may be recirculated, but a portion of it should always be brought in from the outside. The amount of outside air required depends upon the type of car, number of passengers, air temperature, humidity, odors, and whether or not occupants are smoking, and will vary from 15 to 90 per cent of the total air circulated.

Careful attention must be exercised in specifying the rate of outside air taken in so as to fit the type of service adequately and yet not to supply more ventilation than is necessary. Conditioning this outside air is a major factor in determining the size of both summer and winter conditioning equipment. With present average ventilation requirements, about 30 per cent of the cooling equipment and sometimes as high as 50 per cent of the heating equipment is necessary to handle only the outside air load.

For normal conditions, 10 cfm of outside air per passenger is sufficient. When smoking is permitted, at least 15 cfm should be admitted. In some of the dining cars and de luxe sleeping cars, outside air rates as high as 20 and 30 cfm per occupant are used.

Method of Air Distribution

The fact that the amount of space devoted to railway passengers may be as low as 60 cu ft per person (ranging as high as 190 cu ft per person) and the high air flow rates made necessary by severe ventilation and sun

loads make the problems of air distribution and air delivery in railway cars critical.

Various methods may be used to distribute the air delivered to the interior of the car by the circulating fan or blower. The methods commonly used are:

1. A duct lengthwise along the center of the car.
2. One or two side ducts built on the outside of monitor-roofed cars, or on the inside of turtle-backed or arch-roofed cars.
3. A ceiling which is perforated for the emission of air into the car; the air being carried to the ceiling by an overhead duct. From one-half to the full area of the ceiling is used for the distribution of the air although practical considerations generally limit the available area to about two-thirds. As large a part of the ceiling area as possible should be used for air distribution.

Delivery grilles and plaques are used, and are often designed to give considerable entrainment and mixing to avoid cool drafts.

Smoking rooms present a special problem. The cloud of smoke that usually hangs near the ceiling can be broken up by having the incoming air directed along the ceiling in all directions at a velocity somewhat higher than that used for the rest of the car. The air should be exhausted from the room by a fan or through a grille to the washroom or lavatory, and then outside by a fan in a ventilator.

For compartments an adjustable supply duct outlet grille of suitable size and design should be provided and provisions made in the door or partition for the removal of the air to be recirculated.

Lower berths in sleeping cars and office cars should be provided with an adjustable air outlet which will discharge the amount of air desired at low velocity in any direction so that the occupant can regulate the ventilation to meet his own requirements.

In cars containing but one or two rooms or compartments, satisfactory results may be obtained by discharging the air directly from the conditioning unit into the upper part of the car. Care must be taken to have a proper discharge velocity. If the velocity is too low, the air will drop before reaching the end of the car and if too high it will discharge against the end bulkhead and be reflected back. Care must be exercised to secure proper circulation, otherwise objectionable drafts will be experienced.

The recirculating air grilles are usually of the straight flow type, and should be located so that objectionable drafts will not be created by the return air. The outside air intakes, located in the car vestibule, on the side of the car, or on the roof of the car, depending upon the location of the cooling coils, should be of ample size to permit the entrance of sufficient outside air. On many of the recently air-conditioned cars, there are no dampers or shutters at the outside air intakes, the percentage of outside air being controlled by adjusting the flow through the recirculating grille.

Air Cleaning

All of the air circulated by the blower is filtered before passing over the cooling coils. In some cars the outside and recirculated air is filtered separately before mixing, while in others the air from the two sources is mixed before passing through a common filter. Filters in use are made 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, retreated, and returned to service while other types are discarded when dirty.

RAILWAY PASSENGER CAR WINTER AIR CONDITIONING

The majority of cars in service use steam from the locomotive or from a head-end, oil-fired boiler as a source of energy for winter heating. In some instances electrical energy from either a head-end generating set or motive power supply is utilized for resistance heating. In still other cases electrical energy and waste heat from individual car engine-generator sets are employed. The peak heating loads which depend largely upon the amount of insulation used in the car, the type of windows (whether single or double glazed), and the ventilation rate, may vary from 150,000 to 250,000 Btu per hour.

In order to temper the cold outside air, about 30 to 50 per cent of the total heat energy required is distributed by means of finned coils or resistance heaters located in the outside air duct. The remainder is usually transmitted to the car air by finned tubing located along the sides of the car near the floor, thus preventing cold convection currents falling from the car windows from reaching the feet of the passengers.

RAILWAY PASSENGER CAR SUMMER AIR CONDITIONING

Three general types of cooling or refrigerating equipment are being used in 13,362 (May, 1944) air conditioned railway cars and 31 air conditioned rail motor cars in the United States. Of these, 30 per cent are ice activated, 15 per cent use steam jet systems, and 50 per cent employ mechanical compression schemes. These systems which functionally are identical with those used for stationary applications (see Chapter 39) are modified in design to meet the requirements of mobile service. Contrasted with stationary applications of summer conditioning equipment, the use of water as a final means of heat disposal from condensers cannot be resorted to because water in such quantities cannot be transported economically. Accordingly, air cooled or evaporative condensers are always used, with the result that mobile cooling equipments operate at higher temperature, pressure, and power requirement levels than stationary equipment.

The maximum cooling and dehumidifying load which depends largely upon the amount of insulation, the type of windows, the ventilation rate, the sun intensity, and the number of passengers may vary from 60,000 to 96,000 Btu per hour.

An average ice-activated system for such capacities uses about 500 lb of ice and 1.2 kw per hour. The increase in car weight due to such a system is approximately 8500 lb.

The same service from a steam jet system is obtained with the expenditure of 180 lb of steam and 3.3 kw per hour, with an added weight per car of 8000 lb.

The mechanical compression systems, all of which use dichlorodifluoromethane as a refrigerant, may be classified by several types depending on the method of driving the compressor. The source of power for driving the compressor (approximately 10 hp) is complicated by the necessity of obtaining this power at all times whether the car is in motion or standing still on the right-of-way or in a terminal where auxiliary power plug-ins are available. In those cases where compressors are driven from car axles, additional refinements in the drive are necessary in order that a nearly constant cooling capacity may be obtained from a variable speed power source. Numerous combinations of electrical generating schemes for generating sufficient electrical energy from the car axle for lighting, ventilation, and summer air conditioning are in use, and their operation

is closely interlocked with compressor demands, need for pre-cooling, battery charging, etc. It is difficult therefore to state the additional weight imposed on a car because of such a compression air conditioning system, but it is probably in the vicinity of 6000 to 8000 lb. These systems, depending mostly upon the locomotive for supplying power for operation, impose a load, including the power required to pull the weight added by the equipment, of from 5 to 10 per cent of the total locomotive power.

Several schemes for relieving the locomotive of this compression load are used. Some of the articulated trains, which run as unit equipment—the same cars always in the same train—employ a head-end, engine-generator combination for supplying power to compressor motors. In other cases, especially in many of the new streamlined trains, propane fueled engine-driven generators on individual cars are used to supply power to motors, as well as to supply all power for car lighting and accessories. Gas engine compressor combinations on individual cars provide attractive low weight equipment where automatic engine operation is permissible under all circumstances. Diesel-powered generator units have been used experimentally on individual cars for supplying electrical energy and in some cases waste engine heat has been used either for modulating refrigeration with a reheat cycle or for car heating purposes.

RAILWAY PASSENGER CAR HUMIDITY AND TEMPERATURE CONTROL

The temperature to be maintained in a car depends upon the outside temperature and the humidity desired inside the car. With a low humidity it is necessary to maintain a higher temperature to establish a desirable comfort condition. Little humidity control has been attempted on cars up to the present time. A certain degree of automatic humidity control is secured with cooling, but the relative humidity obtained depends largely upon the temperature of the evaporator, which should be below the dew-point temperature of the air. With certain outside atmospheric conditions it may not be possible to operate the conventional equipment with a sufficiently low evaporator temperature to reduce the humidity without dropping the temperature too low. One method has been developed whereby the evaporator temperature is carried below the dew-point a sufficient amount to insure dehumidification and then the cold air is heated to the proper temperature by passing it over coils through which part of the high temperature liquid from the condenser is by-passed. Such a system is costly and has not been generally applied. The reheat cycle obtainable from waste engine heat may be used to good advantage in reducing the humidity without reducing the dry-bulb temperature.

During the heating season humidification is desirable from a comfort standpoint, but unless properly controlled, condensation will appear on the windows. A steam or water spray controlled by a humidistat will provide the necessary moisture for humidification. There are several cars with this feature now in use.

Temperature control for the most part obtained by rugged thermostats and relays capable of withstanding vibrations attendant with mobile service is usual equipment.

Manual zone control for varying outdoor conditions, as well as controls which regulate the car temperature automatically in accordance with outdoor conditions, are employed.

Simplified controls from the standpoint of operation by train crews and especially from the servicing viewpoint are very desirable. The control of summer temperatures is accomplished mainly by cycling the complete cooling system; however, modulation is being effected by using multiple evaporators in which a fixed portion may automatically be cut out of operation to suit the cooling capacity requirements and to keep the equipment in operation for longer periods.

With motor or engine driven compression equipment modulation of compressor capacity to suit the reduced evaporator capacity is accomplished by changing motor or engine speed, or by varying the number of compressor cylinders in operation.

Steam ejector equipment is provided with dampers which by-pass the air around the cooling coil during the off cycle to prevent re-evaporation of moisture from the coil during the off period.

For further information on controls, see Chapter 34.

PASSENGER BUS SUMMER AIR CONDITIONING AND VENTILATION

The highways in the United States are now traveled by about 3500 summer air-conditioned passenger buses. Many of the facts stressed in connection with the design and installation of summer conditioning equipment in railway cars are even more important in these newer vehicles. Weight and space limitations are more stringent, and the problem of circulating from 900 to 1200 cfm of air in coaches carrying from 25 to 40 passengers with about 35 cu ft of space per passenger without drafts is no easy one.

Some bulkhead delivery systems have been used and, while the overhead package racks have served to break up drafts to some extent, these installations are not gaining in popularity. Longitudinal ducts in the corners above the package racks are sometimes used to carry conditioned air to a series of outlet louvers along the top of the windows. Other designs provide for false spaces below the package racks which serve as ducts to distribute air to either entrainment grilles in the bottom of the racks or distributing slots at the edges of the package racks. Some coaches employ a false ceiling to provide a duct, with delivery taking place from numerous perforations in the ceiling.

Return air grilles and filters are usually located near the rear ceiling where the evaporator is placed. Outside air intakes and filters are located preferably near the front of the vehicle so as not to contaminate this supply with exhaust fumes and road dust. Of the 30 cfm circulated per person, about 8 to 10 cfm are outside air and the remainder is recirculated. Power for the motor driving the centrifugal fans is obtained from the bus battery.

More recently a coach design has been brought out which provides for a number of return air outlets below the seats; these permit return air to enter a longitudinal duct below the floor. The filters and evaporator are located in this duct near the front of the vehicle. In this instance a central heating coil utilizing waste heat from the coach engine is also located in this duct. Conditioned air is delivered through a pair of vertical ducts to a package rack distribution scheme.

Summer conditioning systems for these vehicles range in cooling capacity from 36,000 to 48,000 Btu per hour. Mechanical compression systems using dichlorodifluoromethane 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 a coach.

Sometimes an auxiliary generator driven by the air conditioning engine is used and serves to help charge the bus battery, thereby offsetting the 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 propellor 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 so that quick daily servicing of the engine is possible. In all cases fuel is obtained from the main bus tanks, and in some cases the main engine jacket water cooling system is used to cool the air conditioning engine.

In the de luxe equipment, after the driver has started the air conditioning engine by means of its own cranking motor, the engine speed is modulated automatically as the refrigeration demand is partially met, and if this demand is then fully met, the engine is stopped thermostatically. Restarting when the cooling thermostat is no longer satisfied is accomplished either automatically or manually. The various protective and automatic devices on the refrigerant and engine systems make some of the bus air conditioning control systems quite complicated.

AUTOMOBILE SUMMER AIR CONDITIONING

Recently summer air conditioning has been applied to automobiles. The average present day automobile with little insulation, large, single glazed window areas, and high infiltration and exfiltration losses requires about 15,000 Btu per hour of cooling capacity. One system utilizes a reciprocating compressor belted from the main engine fan shaft thus operating at varying speeds up to 3000 rpm. The resulting refrigeration capacity varies from about 6000 Btu per hour at idling speed to 24,000 Btu per hour at maximum car speed.

A dry air condenser is placed in front of the engine radiator, and the liquid and suction refrigerant lines run back under the car floor to the evaporator which is located in back of the rear seat. Conditioned air is delivered into the car just above the shelf near the back of the rear seat. A return grille is provided under the rear seat, and the recirculated air is filtered. Outdoor air is provided by infiltration. Power for the air circulating blowers is obtained from the car storage battery. Equipment of this nature increases the car weight approximately 200 lb.

AIRPLANE AIR CONDITIONING

Air conditioning of airplanes is closely associated with heating and ventilation of airplanes, but air conditioning of planes in flight has received little attention to date, probably because of added weight of equipment necessary. Under ordinary conditions the cleanliness and humidity desired are fixed by natural conditions and therefore the supply of heat and temperature control become the chief consideration. Passenger planes and military planes even at low level flying are at altitudes which require no cooling and at extreme altitudes the design temperature of the atmosphere is 60 deg below zero.

In the operation of internal combustion engines the heating value of fuel appears approximately, one-third as power, one-third transferred to cooling fluid, and one-third in the exhaust gases.

Ordinarily, the heat in the exhaust gases is ample for all heating needs, despite the excessive transmission loss due to high speed, low temperatures, and light wall construction. The maximum coefficient of heat transmission without insulation is found to be 2.28. The minimum coefficient with insulation is 0.33. For present day construction a value of 0.56 may be assumed. Naturally, the waste heat in the exhaust gases offers an attractive source for designers of heating equipment, who in the course of design have followed closely the development of heating as applied to land structures. In first attempts heat was obtained from an annular space surrounding the exhaust pipe, from which branches conducted the heated air to grilles or to seats of passengers. The speed of the plane eliminated the need for a fan or pump. In present practice danger of carbon monoxide contamination is avoided by the use of both primary and secondary heat exchangers.

Heating systems using hot air are in use for passenger and military planes. Steam systems have been used with boilers located in the path of exhaust gases but difficulty experienced in preventing freezing of the water has apparently rendered this system obsolete. A similar system using glycol is in use and requires special care because of the peculiar qualities of glycol. Thermostatic control, mixing dampers and air distribution vary little from standard practice. Controlling devices are built to withstand the extensive vibrations experienced. Humidity determinations under flying conditions show need for humidity control. Passenger planes or other planes requiring air conditioning when on the ground are satisfactorily serviced by portable air conditioning machines. High flying airplanes—above an elevation of 14,000 ft—require oxygen supply or pressurized cabins. The compression of the outside air by the supercharger raises the temperature of the cabin air from an extremely low to a comfortable high. The compression of the air also increases the humidity. Cleanliness at flying altitude is not a consideration, except under special conditions such as above deserts or upon encountering dust storms. In small planes carrying several passengers or crew, localized heating and ventilation are provided. Electrically heated clothing and oxygen masks meet requirements. At high altitudes pressurized cabins require no special oxygen supply.

Medium bombers at one time were equipped with separately fired heating units, but larger bombers, to meet the heating load imposed by crew and equipment, demand too much heat to be supplied in this manner. The recovery of heat from waste gases is therefore imperative. In airplane operations light weight is a more important requirement than in any other type of transportation. Noise should be reduced to a minimum and combined use may be made of material for insulating and noise reduction.

REFERENCES

Railway Passenger Cars

Summary Report on Air Conditioning of Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, November 24, 1936.

Engineering Report on Air Conditioning of Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, April 15, 1937.

Report on Performance and Cost of Operation of 1937 Internal Combustion Engine Mechanical Compression Equipment for Air Conditioning Railroad Passenger Cars, by Division of Equipment Research, *Association of American Railroads*, May 1, 1937.

Report on Relative Performance of Air Filters, by Mechanical Division, *Association of American Railroads*, January 15, 1938.

Air Conditioning of Railroad Passenger Cars, by L. W. Wallace and G. G. Early, Jr. (*A.S.M.E. Transactions*, November, 1937).

Improved Methods of Railroad Air Conditioning, by A. E. Voigt (*Refrigerating Engineering*, July, 1940, p. 16).

Comfort Cooling on Wheels, by F. H. Sahlmann (*Refrigerating Engineering*, May, 1934, p. 233).

Passenger Car Cooling Methods, by Kenneth Cartwright (*Refrigerating Engineering*, February, 1936, p. 83 and March, 1936, p. 158).

Diesel Drive for Passenger Air Conditioning, by J. R. Hornaday (*Refrigerating Engineering*, March, 1942, p. 139).

Railroad Air Conditioning, by Gordon T. Wilson (*Refrigerating Engineering*, May, 1943, p. 323).

Buses and Automobiles

Iso-Thermal Manifold and the Conditioned-Air Unit Used on Nash Cars, by E. Monson (*Society of Automotive Engineers Journal*, January, 1938).

Summer and Winter Air Conditioning of Automobiles and Buses, by L. W. Child (*Heating and Ventilating*, March, 1938, p. 38).

Air Conditioning Takes to the Highways (*Heating, Piping and Air Conditioning*, May, 1938, p. 314).

Fleet of Air Conditioned Buses West Carries Demountable Units (*Heating and Ventilating*, May, 1938, p. 30).

Conditioned Air for Automobiles (*Heating and Ventilating*, May, 1938, p. 44).

Air Conditioning of Automobiles and Buses, by L. W. Child (*Society of Automotive Engineers Journal*, June, 1938).

Bus Air Conditioning, by Jerry Hicke (*Heating, Piping and Air Conditioning*, October, 1938, p. 639).

Bus Air Conditioning, by A. J. Mallinckrodt and Lars Hanson (*Refrigerating Engineering*, June, 1939, p. 388).

Problems in Air Conditioning Automobiles, by F. J. Linsenmeyer (*Society of Automotive Engineers Journal*, July, 1939).

Packard Offers Automobile Air Conditioning System (*Heating, Piping and Air Conditioning*, December, 1939, p. 756).

Airplanes

Flight in the Sub-stratosphere, by J. C. Edgerton (*Society of Automotive Engineers Journal*, 39:484-6 December, 1936).

Steam Heating for Modern Air Transports, by Jack McGuire (*Heating and Ventilating*, February, 1937, p. 37).

Comfort Aloft (*Heating and Ventilating*, October, 1937, p. 40).

Portable Climate, American Buys 12 Plane Air Conditioning Units (*Aviation*, 36:59, September, 1937).

Steam Heating in the Sky (*Heating, Piping and Air Conditioning*, October, 1938, p. 631).

Altitude Conditioning (*Heating and Ventilating*, August, 1940, p. 16).

Comfort in High Altitude Flying, by D. W. Tomlinson (A.S.H.V.E. Transactions, Vol. 47, 1941, p. 57).

Heat Exchangers for Aircraft, by Arthur J. Hess (*Refrigerating Engineering*, September, 1944, p. 192).

CHAPTER 49

Marine Heating and Ventilation

General Considerations, Ship Construction Features, Factors Affecting Design, Requirement for Various Types of Space, Ship Insulation

THE importance of adequate shipboard heating and ventilation arrangements cannot be overemphasized. Installations must not only keep the passengers comfortable and the operating personnel physically fit and mentally capable of maintaining the ship as a profitable business investment, but must also satisfy the human element. The provision of satisfactory living and working conditions is one of the most economical means of keeping a high morale.

GENERAL CONSIDERATIONS

Unlike a shore establishment in connection with which personnel obtain their food, rest and recreation from outside sources, a ship must be self-sufficient and provide for all human needs. Accordingly, every vessel is a complete, independent community comprising, at least to some degree, the facilities available in the average urban development. These facilities must be contained in the minimum practicable space and weight to conserve dead weight and increase the pay load. The pay load may be expressed in terms of cargo carrying ability, passenger carrying capacity, fighting strength, or towing ability.

An atmosphere compatible with efficient operation must be maintained in working and machinery spaces, and temperatures must be such that materials or equipment will not be damaged. Conditions in living spaces, whether ashore or afloat, must permit adequate rest and comfort of occupants. Passenger accommodations must be treated to provide service and living conditions similar to those afforded by the various classes of hotels. Many of the expedients used ashore for this purpose are not applicable afloat. For instance, all living quarters aboard ship cannot be located at a distance from the power plant; but must often have boundaries in common with heat producing spaces. The thermal conductivity of shipbuilding materials such as steel, copper, brass, etc., is many times the value for building materials used ashore, and this, together with concentrated arrangement of equipment, produces a difficult heat transfer and insulation problem. Furthermore, the use of portholes, windows, skylights, and similar openings is greatly restricted in marine applications because of the necessity for strength, water-tightness and protection from the sea in foul weather. During wartime, the utility of portholes, windows, and skylights is greatly restricted because they must be fitted with light-excluding devices.

Much of the cooling and ventilation that is accomplished by natural means ashore must be handled by mechanical means afloat. Space does not exist for the large ventilation trunks required for natural ventilation. Also, because of the restricted space and shipbuilding materials the heat quantities that must be dissipated by ventilation air are often greater than would be required for a similar shore installation. Mechanical ventilation is not merely desirable, but is absolutely necessary in many shipboard spaces to enable the crew to operate the vessel. Furthermore there are numerous spaces which must be ventilated to prevent the accumula-

tion of objectionable, combustible and toxic gases. Shipboard equipment must also be reliable inasmuch as specialized servicing facilities are not available at sea, and failure during an emergency may jeopardize the vessel's safety. Experience has demonstrated that *simple* and *foolproof* heating and ventilating arrangements are essential for satisfactory service. Ventilation systems on ships must be designed to operate in heavy weather with little or no attention, and to resist the efforts of unauthorized personnel to readjust the heat or air distribution. Also, they must be laid out in such a manner that the watertight and fireproof integrity of the vessel is not impaired.

Contrary to the impression among some engineers and designers, that there is a basic difference between marine and shore installations in the application of heating, ventilating and air conditioning arrangements, the same fundamentals apply to both shore and shipboard applications. The air capacity handled by a system ashore will equal the capacity that an identical system will deliver afloat. The same laws of heat transfer apply, and the same heat balance must be maintained in the human body. The only variation between ship and shore applications is that emphasis is placed on different practical aspects of the design.

SHIP CONSTRUCTION FEATURES

The seaworthiness of the vessel is increased by subdividing the main watertight hull space into a number of watertight compartments. This is usually done by providing transverse watertight structures (bulkheads) so located along the ship's length that the vessel will not sink even when the shell is pierced in one or more watertight compartments, by collision, torpedoing, or other causes. It is obvious that the watertightness of these bulkheads is very important, particularly under war conditions. Every effort is made to lead ventilation ducts so that they do not pass through such structures. Where openings must be cut for ventilation or other purposes, special watertight closures are fitted to protect the watertightness of the bulkheads.

In addition to the obstructions caused by the main transverse bulkheads, the ship's interior is fitted with decks and flats (partial decks) that are spaced to suit the use to which the space is put. The deck height (floor height) in cargo spaces generally varies from 9 to 30 ft. This height in living quarters varies from 7½ to 9 ft. Usually in main machinery spaces, the entire depth is subdivided only by a few flats, with gratings provided where necessary to service the equipment.

Bulkheads, shell, and decks are designed to suit structural as well as watertight requirements of the vessel, and consequently, the designer must always contend with beams, stiffeners, stanchions and brackets.

Fig. 1 indicates the arrangement of a typical cargo vessel, and shows the main transverse watertight bulkheads, decks, structural members, and many other pertinent characteristics. Living quarters are usually located on and above the weather deck, this custom being based on the assumption that better natural air and light are available in this location. It is obvious that the upper 'tween deck and other spaces can be conveniently used for additional living and working spaces, but that natural ventilation is far less effective in this case.

The foregoing description mentions some of the limits and natural boundaries which affect the design and layout of ship ventilation and heating systems. It should be added that there are also equipment, wireways, and miscellaneous piping which must be considered. Since

there is very little unassigned space, fan rooms are usually quite congested and many studies must be made to ascertain the arrangement of equipment which will best suit the conditions. Cuts in structural members must be limited in number and size. Lack of headroom often causes poor aspect ratios, inaccessibility, and other undesirable conditions. It therefore follows that the design of a ship's heating and ventilating system is far more than a theoretical calculation. A good designer must be familiar with all phases of ship construction and operation in order to provide sufficient and efficient ventilation and heating, and to locate the equipment where it will interfere least with basic arrangements, major equipment, stability, and other important factors.

FACTORS AFFECTING DESIGN

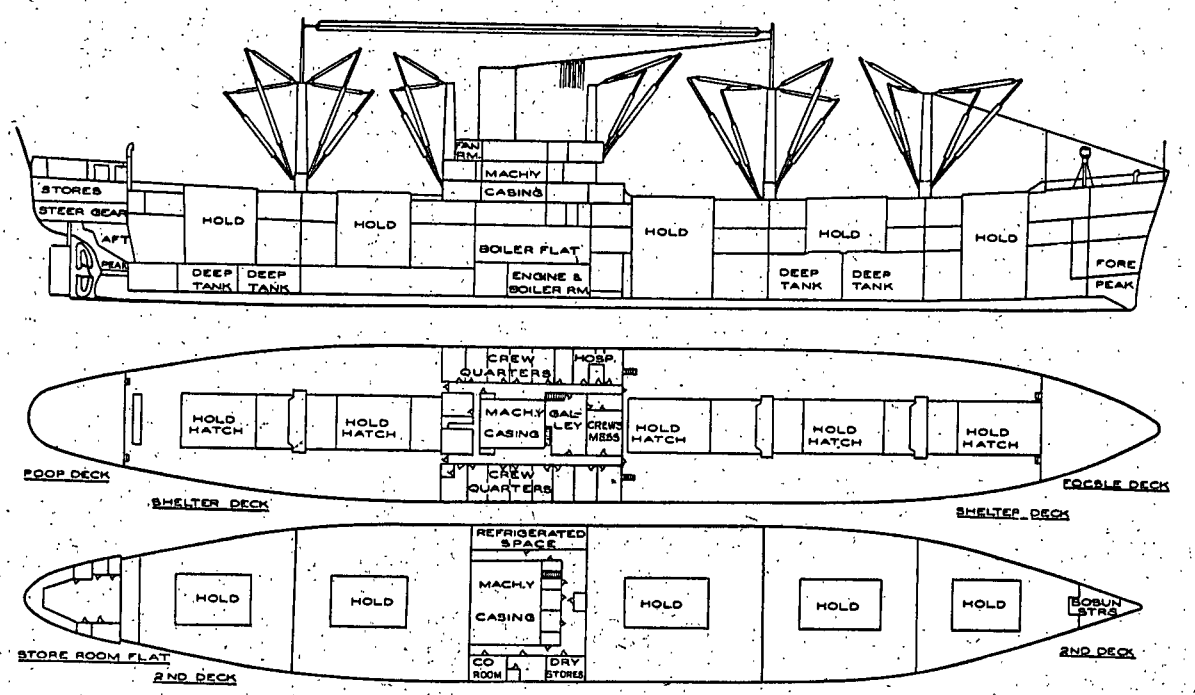
A ship may be favorably compared to a *floating hotel.* Its make-up includes living accommodations, dining rooms, shops, power plants, commissary spaces, etc., all of which must be so arranged and maintained as to make it self-sustaining for long periods of time. In contrast to a hotel, however, a ship may, on a single cruise, move from arctic to tropical climates. On the other hand, it may be built to operate only in one single locality. It is therefore necessary to assume maximum and minimum design air temperature conditions based on the intended use of the vessel so that adequate cooling and heating may be provided. For vessels designed to operate generally in all parts of the world, a maximum summer temperature may be taken as 88 F and a minimum winter temperature as 10 F. For other ships, those which spend long periods in port or in inland waters, the maximum and minimum temperatures may be taken as 90 and 0 F respectively.

Preliminary design of heating and ventilating installations aboard ship is simplified by the uniformity of the conditions which apply to all ships. Among such conditions are (1) the necessity for the vessel to supply its own power, (2) the availability of an unlimited supply of sea water, (3) the drastic restriction of shipbuilding materials because of strength requirements, corrosion resistance and fire hazard, and (4) the limitation of available space and permissible weight. On the other hand, shipboard space, weight, and power limitations require precise layouts with minimum design safety factors, and this necessitates unusual attention to design details. Particular attention should be given also to the adjustment of a system after installation, because designed air quantities are usually close to the minimum acceptable and therefore, improper balance will result in unsatisfactory conditions in some spaces.

Each space aboard ship must be treated in accordance with the particular requirements of its use. The ventilation of most spaces is determined by empirical calculations. The quantity of air supplied must be not less than that which satisfies (1) maximum allowable temperature rise, (2) minimum allowable fresh air per person, and (3) permissible air change, named in order of importance.

The present practice of various ship designers may be to supply the quantity of air which satisfies only one of these requirements, but the most satisfactory ship ventilation will result from the use of that quantity which is large enough to satisfy all three.

When heating and cooling loads are estimated, the cooling effect of the water through which the vessel is sailing must be considered if the space is at or below the waterline. Maximum and minimum water temperatures of 85 and 35 F are usually used for such calculations. This factor is



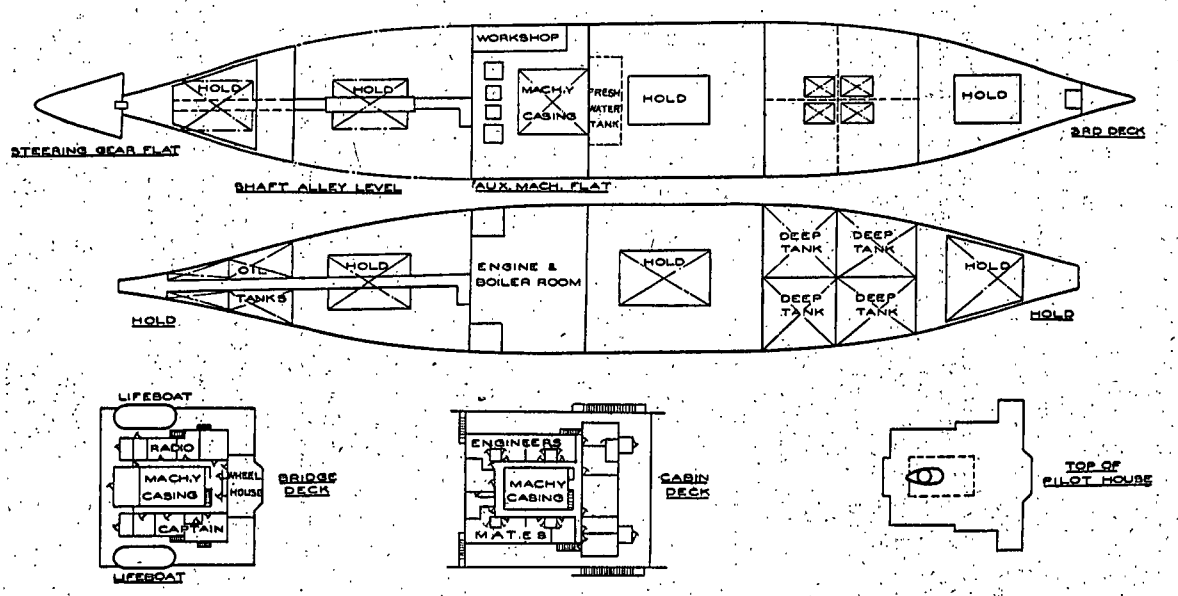


FIG. 1. ARRANGEMENT OF A TYPICAL CARGO VESSEL

important enough to justify the omission of ventilation from certain spaces which have only small heat gains and do not require fresh air for other purposes.

The relation between supply and exhaust ventilation depends on the type of space, and the space's relation to the weather. In any case, it should be remembered that positive means for supply and exhaust must always be provided. Where natural ventilation is anticipated, infiltration and exfiltration via open doors, windows, etc., cannot always be considered adequate for this purpose.

In addition to the ventilation requirements, but closely related to and usually forming an integral part of the study, is the manner in which heating shall be furnished for cold weather conditions. In all spaces which are provided with mechanical supply ventilation, where heat is also required, the simpler method and most economical from standpoint of weight is that in which duct type or blast heaters are fitted in the ventilating systems. Where this type of heating is furnished, convectors or radiators should be provided for spaces requiring heat but not furnished with mechanical supply ventilation. The most satisfactory arrangement for this method of heating, except where individual space control is essential, is zone heating. To accomplish this, the ventilation supply system, with the fan operating at slow speed (usually about 50 to 70 per cent delivery) passes the air over a preheater which raises the air to a temperature of about 40 to 60 F. This preheated air is then passed through the system and thence through one or more reheaters which are located in the ducts and so arranged that each serves a group of spaces which possess the same or similar heating requirements. It may be necessary to slightly revise the individual space air quantities, which have been determined on the basis of summer cooling, in order to bring them in line with the reheat or delivery temperature. In some cases central heating is used. By this arrangement all the air needed for heating is passed through the preheater and reheater fitted in series. This system has the disadvantage of affording poorer thermostatic control and cruder heating properties. In all installations of duct type heaters, however, care must be taken in the first stage of heating (preheater) to provide protection against freezing, either by heater design or compensating automatic valve control. Of course, many of the usual combinations or variations of duct type and convection or radiant heaters may be used with varying degrees of satisfaction, depending on basic design requirements such as weight and space limitations and economic justification for refinements.

REQUIREMENT FOR VARIOUS TYPES OF SPACE

In the paragraphs which follow the most important considerations are given for the treatment of the various spaces, but the resulting quantities of air should in all cases be checked with those shown in Table 1.

Machinery Spaces

The prime purpose of machinery space ventilation is to maintain a habitable temperature for the operating personnel. Of secondary importance is the necessity for preserving temperature conditions satisfactory for the successful operation of machinery, particularly of that designed for certain maximum ambient temperatures. High dissipation of heat, from high temperature steam equipment or from high powered electrical installations, make it more practicable to use *spot cooling* of working

TABLE 1. TYPICAL HEATING AND VENTILATION PRACTICES FOR SHIPS

SPACE	TYPE OF VENTILATION		TYPE OF HEATING	AIR CHANGE MINUTES	REMARKS
	Sup.	Exh.			
Crews Quarters	Mech.	Mech. or Nat.	H.B.[a] or D.R.[b]	5-6	
Toilets and Showers	Nat.	Mech.	D.R.	3-4	
Chart House	Mech.	Mech. or Nat.	H.B. or D.R.	5-6	
Passenger & Officers Qtrs.	Mech.	Mech. or Nat.	H.B. or D.R.	4-6	
Office & Similar Spaces	Mech.	Mech. or Nat.	H.B. or D.R.	5-6	
Mess Rooms	Mech.	Mech. or Nat.	H.B. or D.R.	4-6	May be exhausted through Galleys and Pantries.
Dining Salons	Mech.	Mech. or Nat.	H.B. or D.R.	3-6	May be exhausted through Galleys and Pantries.
Galley and Pantry	Mech. & Nat.	Mech.	H.B.	1-2	Tempered air Sup.
Deck Pantries (No cooking Equipment)	Mech. & Nat.	Mech.	H.B. or D.R.	3-6	
Stores Spaces (Adjacent to Living Quarters)	Mech. or Nat.	Nat. or Mech.	.....	15-20	
Stores Spaces (Isolated Locations)	Mech. or Nat.	Nat. or Mech.	.....	15-30	
Hospital Spaces	Mech.	Mech.	H.B. or D.R.	4	
Soiled Linens and Oilskin Lockers	Nat.	Mech.	.....	15-20	
Shops	Mech.	Mech. or Nat.	H.B. or D.R.	6-10	
Group Berthing Spaces	Mech.	Mech. or Nat.	H.B. or D.R.	3-6	
CO_2 Bottle Space	Nat.	Mech.	.....	10	
Battery Room	Nat.	Mech.	D.R.	2	
Laundry	Mech.	Mech.	H.B.	1-4	Tempered air Sup.
Dry Cargo Holds	Mech. or Nat.	Nat. or Mech.	.....	20-30	
Stearing Gear Space	Mech. or Nat.	Mech. or Nat.	H.B. or D.R.	2-6	Unit Heater if desired.
Dry Stores	Mech.	Nat. or Mech.	.....	6-10	
Gyro Room	Mech.	Nat. or Mech.	.....	3-6	
Theaters and Halls	Mech.	Nat. or Mech.	H.B. or D.R.	6-8	
Passageways	Nat.	Nat.	D.R.	.....	Exhausting or Supplying adjacent Quarters.
Engine & Boiler Rooms	Mech. or Nat.	Mech. or Nat.	.....	1-2	
Misc. Machinery	Mech. or Nat.	Mech. or Nat.	.....	2-4	
Resistor Spaces	Nat.	Mech.	.....	10	
Stair Wells	Nat.	Nat.	D.R.	.....	Same as passageways. Exh. at top.

[a] H. B.—Hot Blast System.
[b] D. R.—Direct Radiation.

areas, rather than to attempt to obtain uniform ambient temperature. The permissible temperature rise at working stations is usually about 15 F while the over-all temperature rise is usually between 30 and 50 F.

When the necessary quantity of air has been determined, the ventilation arrangements should be designed so that all of this air will be distributed to locations where the personnel are to be stationed under operating conditions, thus giving them the maximum benefit from the cool air so introduced. This consideration is of vital importance because it must be remembered that the physical well being and alertness of the crew are more important than the operating ambients of the machinery; and by such installation the personnel receives the air before it becomes heated to the temperatures satisfactory for the equipment.

In addition to the air supply, these spaces must be exhausted, preferably by mechanical means. Every attempt should be made to remove air at or close to heat sources so that the mixing of ambient air with that which has been heated to a higher temperature will be at a minimum. The quantity or capacity of the mechanical exhaust systems should take into consideration the expansion of the supply air because of heating and should be also sufficient to insure an indraft through access openings to the space. Generally, in order to accomplish this, the quantity of mechanical exhaust should be about 120 per cent of the supply. In many installations, the combustion air for diesel engines and boilers is taken directly from the machinery space and serves as a partial mechanical exhaust system.

Normally, heat is not required for machinery spaces except for those fitted with electrically operated equipment, which may remain inactive during periods while in port. Such spaces should be equipped so as to heat them to about 50 F during such periods.

Living Spaces

The minimum quantity of air required for any space which is fitted for sleeping or office work, including hospital spaces, should be that which will limit the temperature rise over the outside air conditions to not more than 10 F (a rise of 7 F is more satisfactory if practicable), or a minimum of 30 cfm per person, whichever is the greater. For spaces fitted for eating, recreation, or manual work the rise may be taken at 10 F with not less than 20 cfm per person.

It will be noted that the quantities thus obtained are minimum and should be supplied by mechanical means. This is especially important if the spaces are located inside of the ship and have no direct weather exposures with ample openings, such as portholes or windows, which may remain open even in inclement weather. The same requirement applies to the necessity for mechanical exhaust, although natural exhaust may be used where only a short run of duct is required.

If the air to these spaces is supplied by mechanical ventilation it should enter through registers or diffusing type terminals. The installation of bracket fans within these compartments at proper locations and of sufficient number will aid materially in providing proper distribution for comfort.

Heat should be furnished to these spaces to maintain the following temperatures when operating in the minimum winter conditions:

Staterooms, Berthing, Messing and Office spaces	70 F
Working spaces and Shops	60 F
Hospital spaces	75 to 78 F

Toilets, Washrooms, Showers and Baths

Spaces for these purposes should be fitted with mechanical exhaust ventilation for odor and steam removal. They should have one or more well-located exhaust terminals overhead; one will usually suffice for the average size space. The supply may be taken naturally. Generally, the surrounding living or working spaces are exhausted through them. The minimum total quantity exhausted should be that which would change the air within the compartment in 6 min, plus 25 cfm for each fixture, plus 50 cfm for each shower head. Air requirements on merchant vessels are commonly estimated on the basis of a 4-min rate of change.

Heating should be obtained by use of convection-type heaters which should maintain 70 to 80 F for bathing or washing spaces and 70 F for toilet spaces when the outside winter temperature is at the design condition.

Galleys, Bakeries, and Food Handling Spaces

The problem of the ventilation of spaces fitted for cooking and food preparation is primarily one of heat and smoke or fume removal. Mechanical exhaust is always required for this purpose. The supply may be natural, but the required exhaust capacity will be most satisfactory if arranged for 50 per cent mechanical and 50 per cent natural.

The exhaust quantities are generally large and may be predicated on restricting the ambient temperature rise to about 15 F over the outside summer design air conditions. It will generally be found that the resulting quantity will change the air in these spaces in about $\frac{3}{8}$ to 1 min, depending on concentration of equipment. All of the exhaust should be arranged so as to remove air from the space through hoods fitted over the cooking equipment and these hoods should contain grease filters over ranges and griddles and should be made readily accessible for frequent cleaning.

The mechanical supply when fitted should blow air directly on the personnel but away from the equipment and so as not to interfere with the flow of exhaust air to the hoods.

Generally, no heat is required for these spaces in winter except that the supply air temperature should not be too low. Usually a preheater delivering air at about 45 to 60 F should be provided.

Laundries

The problems in the ventilation of laundries are somewhat similar to those for galleys. Heat removal is necessary and best accomplished by the installation of mechanical exhaust through hoods fitted over the heat-producing equipment.

The supply for these spaces may be all mechanical, or part mechanical and part natural. The mechanical supply should be distributed through adjustable blast-type terminals at relatively high velocities directed to blow cool air on the torsos of the operating personnel. Experience indicates that this quantity should be sufficient to change the air in the spaces in from 1 to 4 min.

The exhaust should be at least one air change per minute and at least equal to 120 per cent of the total supply so as to insure an indraft of air through the access openings to the space. All exhaust openings within the space should be fitted to be readily accessible for frequent cleaning and lint removal.

No winter heating is required except that the supply air should be delivered at about 40 to 60 F.

Storerooms and Cargo Spaces

The ventilation of these spaces should be predicated upon the kind and type of stores or cargo to be carried.

For materials which would not be adversely affected by summer temperatures, no ventilation is required. Also, storerooms or cargo spaces below the water line, in which temperatures would not normally exceed 100 F with the sea water assumed as 85 F maximum, may not require ventilation for certain cargoes. As spaces in these two categories are frequently damp some means of moisture removal must be provided. Chemical dessicants are satisfactory where it is essential to prohibit openings through watertight structure. In other cases a supply of dry air is provided from a central silica gel dehumidification system and distributing ducts, with recirculation used to accelerate the drying process.

Where storage spaces must be ventilated to obtain a change of air in about 15 to 30 min, in some cases the ventilation is determined by the maximum temperature which the cargo or stores can withstand without damage.

Spaces in which inflammable liquids are carried, or where inflammable vapors may be generated, require special consideration. They should be fitted with mechanical exhaust with terminals so located as to remove explosive or combustible vapors. The supply to these spaces may be natural and arranged so that good distribution, free from pockets, is assured.

If stores and cargo, which require some special and constant temperature or humidity control, are carried, special air conditioning equipment must be installed to suit the particular requirements.

SHIP INSULATION

In order to properly limit one of the major ventilation heat loads which is that made necessary by heat transmission, and to prevent condensation, it is necessary to use insulation judiciously. The principal sources of heat in a ship are the power plant and sun load. The confinement or exclusion of this heat in the structure of a ship is not easy, principally because of the complex structural nature of the beams, stiffeners, bulkheads, decks and hull. The continuous metal paths offer easy means of heat flow throughout the structure.

Insulation like any other component hull part of the ship cannot be used indiscriminately because of weight and space limitations. Therefore higher heat transmission coefficients are accepted for insulated structures of certain classes of vessels, than would be considered satisfactory ashore. On passenger and cargo vessels, the structure is frequently covered with a metal sheathing in order to improve appearance. Such sheathing reduces the resistance to heat flow because the necessary supports form a metallic contact that by-passes the insulation.

Hull insulation may be either sheathed fill or blanket and board type and should possess certain desirable physical characteristics, namely:

1. *Fireproofness.* The material must be incombustible and when subjected to high temperatures by fires within compartments it must not give off smoke or harmful gases. If cements are used to secure the material they too must satisfy the same combustible restrictions. When the exposed surface of the insulation is to be finished with paint, the paint should be fire retardant. Insulation properly used will retard the spread of fires within ships. Government regulations govern the construction and insulation of bulkheads to prevent the spread of fire on vessels.

2. *Density.* A 6,000 ton warship may have from 13 to 25 tons of hull insulation, depending on the type of insulation. It is obviously desirable to minimize this dead weight commensurate with other considerations.

3. *Thermal Conductivity.* It is important that the conductivity of insulation used be 0.33 Btu per (hour) (square foot) (Fahrenheit degree per inch), or less.

4. *Ruggedness.* As any exposed or internally applied material is subjected to rough usage aboard ship it must be able to withstand much pounding from the seas and vibration from the ship's machinery.

5. *Verminproof.* For sanitary reasons it is essential that insulation harbor no vermin.

6. *Applicability.* Because of the necessity of speeding construction of vessels and minimizing costs, insulating materials must lend themselves to easy and ready application. Generally, when cements are used the application is slow and laborious.

For duct insulation mineral wool or spun glass are the most commonly used materials. Corrugated asbestos is not recommended because the presence of moisture tends to disintegrate it. Semi-rigid insulation is generally used because it is simplest to install. Blanket type insulations are applied only to round or curved surfaces, and are secured with twine and lagged with sewed-on canvas. Semi-rigid type insulation, secured by adhesive, flat wire bands, and corner clips, is lagged with canvas only where exposed. Blanket type insulation is at least 1 in. thick. The thickness of semi-rigid insulation is frequently selected in accordance with the values given in Table 2.

TABLE 2. DUCT INSULATION THICKNESS FOR USE IN SHIPS

APPLICATION	THICKNESS IN.
Tempered air ducts in unheated spaces where a temperature differential greater than 35 F exists	½
Cold air ducts, supply and exhaust passing through heated spaces	1
Exposed ducts in heated spaces, carrying reheated air	½
Concealed ducts carrying reheated air, adjacent structure exposed	1
Concealed ducts carrying reheated air, adjacent structure not exposed	½
Ducts serving only the space in which they are located	None
Exhaust ducts carrying hot gases (galley, forges, etc.), in living and working spaces	1
Supply ducts in machinery spaces and similar hot spaces, serving spaces other than the hot spaces through which they pass	1
Heaters	1
Ducts subject to excessive sweating	½

Fans are seldom insulated. Preheaters are frequently located close to the fresh air intake in order to conserve insulation, and for the same reason zone reheaters are located as close to the zone as possible. Where a reheater serves only one space, the heater is commonly located in the space.

BIBLIOGRAPHY

The Ventilation of Ships, by R. McDonald (*Journal of the Institution of Heating and Ventilating Engineers* (Br) October, 1939).

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).

Heating, Ventilating and Air Conditioning on Shipboard, by J. H. Clarke (*Heating, Piping and Air Conditioning*, August, p. 467; September, p. 529; October, p. 610, 1940).

Care of Cargo at Sea, by O. D. Colvin, W. H. E. Hahne and M. R. Colby (*Transactions*

of the Society of Naval Architects and Marine Engineers; Part I, Vol. 46, 1938, p. 109; Part II, Vol. 49, 1941, p. 208).

Ventilation of Ships, by J. Dawson (*Journal of The Institution of Heating and Ventilating Engineers*, London, Vol. 10, June-July, 1942, No. III, p. 89).

Modern Marine Engineers Manual, Vol. II, Sections 16 to 19, Incl. (Cornell Maritime Press, 1943).

Warship Ventilating, Heating and Air Conditioning, by Comdr. T. H. Urdahl, U.S.N.R., and W. C. Whittlesey (A.S.H.V.E. TRANSACTIONS, Vol. 49, 1943, p. 35).

Standardized Heating and Ventilating Equipment for Fighting Ships, by Comdr. T. H. Urdahl, U.S.N.R. and Lt. John Everetts, Jr., U.S.N.R. (*Heating, Piping & Air Conditioning*, July, 1943, p. 333).

Designing Warship Ventilation with Standardized Equipment, by Comdr. T. H. Urdahl, U.S.N.R. and W. C. Whittlesey (*Heating, Piping & Air Conditioning*, August, 1943, p. 419).

CHAPTER 50

Hot Water Supply

Hot Water Supply Piping, Storage Capacity and Heating Load, Methods of Heating Water, Computing Grate and Coil Surface Areas, Controls, Solar Water Heaters, Friction Loss

IN computing the total heating load for a building, it is important to allow ample boiler capacity for heating the hot water supply. The amount of warm water used in any large building is variable, depending on the type of structure, usage, occupancy and time of day. It is necessary to provide the piping, water heating and storage facilities of sufficient capacity to meet the peak demand without wasteful excess in equipment cost.

The determination of the amount of water used in residences has been well established over a period of time and as a result, reliable factors for water consumption are available. Tests have been made of the amount of water required by standard fixtures in normal use with water at ordinary pressures so that this information permits a fairly correct basis of design.

HOT WATER SUPPLY PIPING

As a result of investigations conducted at the *National Bureau of Standards*, basic design principles have been outlined for the design of the hot and cold water supply piping requirements in a plumbing system[1].

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. 1. 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 run-out 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. 1 (*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. 1 (*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.

STORAGE CAPACITY AND HEATING LOAD

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, office buildings, etc., 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 tempera-

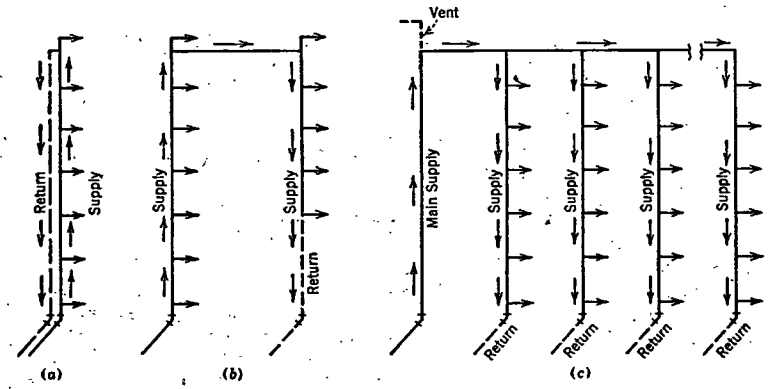


FIG. 1. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

ture. 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.

In estimating the hot water which can be drawn from a storage tank it should be borne in mind that only about 75 per cent of the volume of the tank is available, as 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 F rise. The actual requirement is $\frac{100 \times 8.33}{240} = 3.48$ sq ft per gallon of water heated 100 F. The value of 4 allows for transmission losses, etc.

There are two ways in common use of estimating the hot water requirements of a building; first, by the number of people and second, by the number of plumbing fixtures installed. Where the number of people to

be served is known or can be reasonably estimated, the data in Table 1 may be used.

Example 1. From Table 1, a residence housing five people would have a daily requirement of $5 \times 40 = 200$ gal per day, and a maximum hourly demand of $200 \times \frac{1}{7} = 28.5$ gal. The heater should have a storage capacity of $200 \times \frac{1}{5} = 40$ gal and a heating capacity of $200 \times \frac{1}{7} = 28.5$ gal per hour.

The conditions given in *Example 1* 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 2.*

Example 2. Assume an apartment house housing 200 people. From the data in Table 1: Daily requirements $= 200 \times 40 = 8000$ gal. Maximum hours demand $= 8000 \times \frac{1}{7} = 1140$ gal. Duration of peak load = 4 hours. Water required for 4-hour 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 hours $= 4560 - 750 = 3710$ gal. Heating capacity per hour $= \frac{3710}{4} = 930$ gal.

If instead of a 1000 gal tank, a 2500 gal tank had been installed, the required heating capacity per hour would be $\frac{4560 - (2500 \times 0.75)}{4} = 671$ gal.

TABLE 1. ESTIMATED HOT WATER DEMAND PER PERSON FOR VARIOUS TYPES OF BUILDINGS

TYPE OF BUILDING	HOT WATER REQUIRED AT 140 F	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 apts., hotels, etc.	40 gal per person per day	1/7	4	1/5	1/7
Office buildings	2 gal per person per day	1/5	2	1/5	1/6
Factory buildings	5 gal per person per day	1/8	1	3/5	1/8
Restaurants $0.50 meals $1.00 meals $1.50 meals	 1.5 gal per meal 2.5 gal per meal 4.5 gal per meal			1/10	1/10
Restaurants 3 meals per day		1/10	8	1/5	1/10
Restaurants 1 meal per day		1/5	2	3/5	1/6

Table 2 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 3* will illustrate the procedure.

Example 3. Determination of heater and storage tank size for an apartment building from number of fixtures.

60 lavatories	× 2 =	120 gal per hour
30 bath tubs	× 20 =	600 gal per hour
30 showers	× 75 =	2250 gal per hour
60 kitchen sinks	× 10 =	600 gal per hour
15 laundry tubs	× 20 =	300 gal per hour
Possible maximum demand		= 3870 gal per hour
Probable maximum demand	= 3870 × 0.30	= 1161 gal per hour
Heater or coil capacity		= 1161 gal per hour
Storage tank capacity	= 1161 × 1.25	= 1450 gal.

TABLE 2. HOT WATER DEMAND PER FIXTURE 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*	1.25	0.90	1.00	0.60	0.80	1.00	2.00	0.70	1.00	1.00

*Ratio of storage tank capacity to probable maximum demand per hour.

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 water surfaces of heat transfer are small, and if the water carries a heavy proportion of precipitable salts, the water passages may soon clog and then burn out. A familiar example of such trouble is the water back of the firebox in the kitchen stove or the pipe coil inserted into the firebox of a warm air furnace or small boiler. The critical water temperature, 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 with water temperatures lower than 140 F.

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 fire-brick lining at the base of the fire-pot to prevent too much chilling of the fuel. While mud and scale will eventually clog the water ways 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

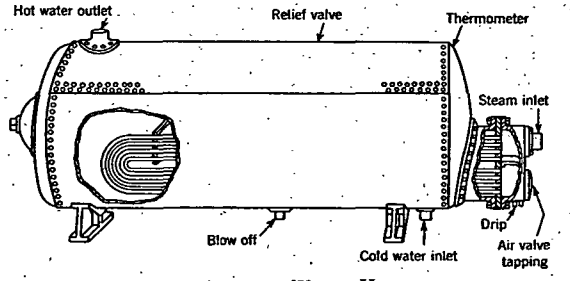


FIG. 2: INDIRECT WATER HEATER

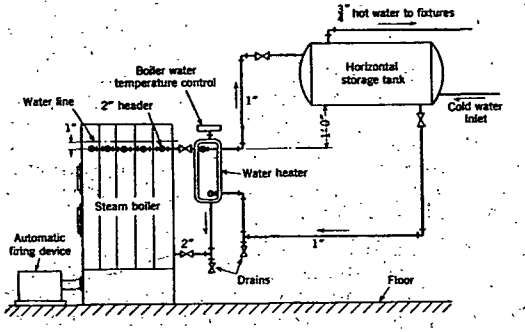


FIG. 3. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

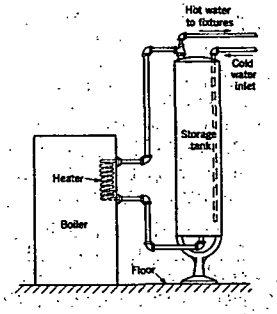


FIG. 4. INDIRECT WATER HEATER PLACED IN BOILER

sediment. Clean-out 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 then commensurate sized coal-burning heaters. They have the same tendency as coal boilers to *lime up*, and 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.

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. 2. The coils usually are of copper and are *U*-shaped to permit expansion and contraction. The shell may be of steel, copper, or with a special inside protective lining. Where straight heating tubes are used, one end of the tube is usually expanded into a *floating* head to take 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. 3. 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 bottom of the tank as far above the boiler as possible. Horizontal storage tanks smaller than 18 or 20 in. diameter are not recommended because of the difficulty of preventing the hot and cold water from mixing, and especially is this an important consideration when large quantities of water are withdrawn. In Fig. 4 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. 3. 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 located in the steam space above the water line of a steam boiler, it operates more satisfactorily when below the water line since clogging of the water tubes may thereby be delayed. This method is widely used without storage tanks since the intimate contact and efficient circulation of the water in this arrangement permit the utilization of the heat stored in the water of the boiler. 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.

COMPUTING AREA OF HEAT TRANSMITTING SURFACE

The area of the inside surface of a heating coil may be determined from Equation 1.

$$A = \frac{Q \times 8.33\,(t_2 - t_1)}{U \times t_m} \qquad (1)$$

where

A = surface area of coil, square feet.
Q = quantity of water heated, gallons per hour.
t_2 = hot water outlet temperature, degrees Fahrenheit.
t_1 = cold water inlet temperature, degrees Fahrenheit.
U = coefficient of heat transmission, Btu per (hour) (square foot) (degree Fahrenheit 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).
t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature and is *approximately*:

$$t_s - \left[\frac{(t_2 + t_1)}{2}\right]$$

t_s = temperature of the heating medium, degrees Fahrenheit.

Equation 1 may be used to check the heating coil ratings under temperatures other than those stated in the manufacturer's published ratings.

TABLE 3. COEFFICIENT OF HEAT TRANSFER OF INSTANTANEOUS WATER HEATERS
U = Btu per (hr) (sq ft) (degree Fahrenheit logarithmic mean temperature difference)

Boiler Water Temperature	210	200	180
U	225	175	150

Example 4. 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?

$$t_m = \left[220 - \frac{(180 + 40)}{2}\right] = 110 \qquad A = \frac{70 \times 8.33\,(180 - 40)}{100 \times 110} = 7.39 \text{ sq ft.}$$

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 3 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, as when there is a sudden demand followed by an inflow of cold water, the bottom of a steam heating 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 water line 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. 3, unless the boiler has large top nipple ports providing inter-sectional circulation. If the top nipples are entirely within the boiler steam space, no internal circulation occurs between sections and steaming may occur in sections not connected to the heater and further the unconnected sections will not deliver any heat to the heater.

COMPUTING GRATE AREA FOR COAL-FIRED HEATER

The grate area required for a small coal-fired water heater may be calculated by Equation 2.

$$G = \frac{W\,(t_2 - t_1) \times 100}{H \times E \times C} \qquad (2)$$

where

G = grate area, square feet.
W = weight of water, pounds per hour.
$t_2 - t_1$ = temperature difference between entering and leaving water, degrees Fahrenheit.
H = heating value of coal, Btu per pound.
C = weight of coal burned, pounds per (hour) (square foot of grate).
E = efficiency, per cent.

In a small heater 4.5 lb is a conservative value for C, and an efficiency of 60 per cent would represent excellent performance.

Example 5. What grate area is required for a coal-burning water heater warming 100 gal per hour of water from 50 to 180 F, when the combustion rate is 4.5 lb per hour per square foot of grate, if the heating value of the fuel is 12,500 Btu per pound, and the efficiency is 60 per cent?

$$\text{Substituting: } G = \frac{100 \times 8.3 \times (180 - 50) \times 100}{12{,}500 \times 60 \times 4.5} = 3.2 \text{ sq ft.}$$

The quantity of gas, oil, or other fuel required per hour for water heating may be calculated by Equation 3.

$$F = \frac{W\,(t_2 - t_1) \times 100}{H \times E} \qquad (3)$$

where

F = units of fuel (lb, cu ft, gal, etc).
H = heating value of fuel, Btu per unit.
W = weight of water, pounds per hour.
$t_2 - t_1$ = temperature difference between entering and leaving water, degrees Fahrenheit.
E = efficiency, per cent.

Efficiencies for oil and gas may be taken as 75 and 80 per cent respectively. The heating value of the fuel and the temperature rise should be determined to suit local conditions.

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 motor 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, immersion thermostat controls 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 all 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.

In buildings which have restaurants it is generally desirable to install two separate service hot water systems so that water at about 180 F minimum may be available for dish washing, while water at 140 F maximum may be used for lavatory and bath purposes.

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.

SOLAR WATER HEATERS

Solar heaters utilize the energy of the sun for heating hot 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 outside 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.

The storage tank must be able to store sufficient heated water for the night period of about 16 hours 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 per cent of the average daily usage is considered adequate. Since about 25 per cent 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:

$$S = \frac{Q \times 0.50}{0.75} = 0.666Q \qquad (4)$$

where

S = storage capacity of tank, gallons. Q = average daily usage, gallons.

Thus for a family of four persons using an average of 40 gal of hot water per person per 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 to this theoretical capacity 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,

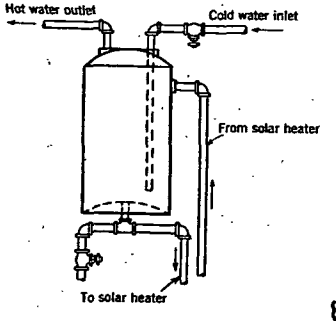


FIG. 5. SOLAR HEATER TANK CONNECTIONS

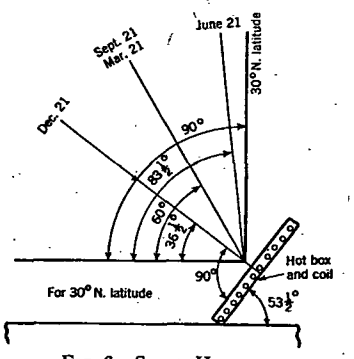


FIG. 6. SOLAR HEATING COIL INCLINATION

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 to the house, as shown in Fig. 5, is located at the top of the tank, which serves to vent the air from the tank through the hot water faucets as fast as it accumulates.

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 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. 6 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. Over the top of the box glass sash are placed and the box should be constructed as near air-tight as possible. 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 of a light gage copper plate on the bottom of the box to which the pipe of the coil is soldered, for good metallic contact, will add

TABLE 4. 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 per cent, 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 (A.S.H.V.E. TRANSACTIONS, Vol. 41, 1935, p. 131).

to the amount of heat received by the coil due to the fact that this plate will receive all of the sun rays which fail to directly strike the coil. The heat from this source is transmitted to the coil through the plate instead of by heating the air surrounding the coil and from which only part of the heat enters the coil, the balance being transmitted through the glass sash.

Design data given in Table 4 may be used with some 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 per person.

FRICTION LOSS IN WATER SUPPLY PIPING

Fig. 7, showing friction loss in *fairly rough* pipe, has been reproduced from Fig. 4 of Building Materials and Structures Report BMS 79, Water Distributing Systems for Buildings, by Roy B. Hunter, *National Bureau of Standards*. For friction loss in tubing, as well as in smooth and rough pipe the reader is referred to Figs. 2, 3 and 5 in the same publication.

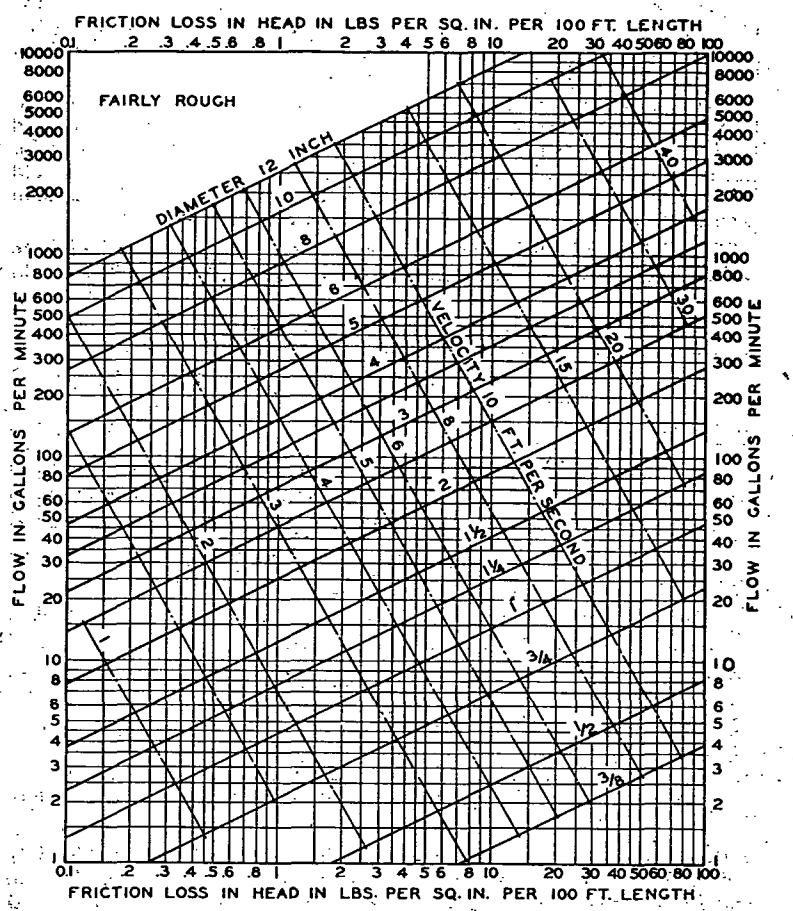


FIG. 7. FRICTION LOSS IN FAIRLY ROUGH PIPE

REFERENCES

1—Methods of Estimating Loads in Plumbing Systems, by R. B. Hunter (*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). Water-Distributing Systems for Buildings, by R. B. Hunter (*National Bureau of Standards*, Report BMS79, 1941).

BIBLIOGRAPHY

Laundry, Kitchen and Hospital Equipment, by H. C. Russell (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 45).

Water Consumption, Cost and Savings, by G. C. St. Laurent (*American Hotel Association, Hotel Engineering*, Vol. 1, 1940).

Plumbing Practice and Design, (1943) by Svend Plum (John Wiley & Sons, Inc.).

CHAPTER 51

Codes and Standards

THE State Codes listed in Table 1 have been compiled from lists supplied by the various states and revised in 1945. These codes can be obtained from the state department indicated.

The Codes and Standards listed in Table 2 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. STATE CODES, STANDARDS OR LAWS RELATING TO THE HEATING, VENTILATING OR AIR CONDITIONING OF BUILDINGS

ALABAMA: None.

ARIZONA: Law 56-117. Laundry—hours of labor—ventilation—penalty; Laws 65-218, 65-231, 65-235. Ventilation (with reference to mines). (Secretary of State, State House, Phoenix.)

ARKANSAS: Installation of all boilers. Dept. of Labor Safety Code No. 6 for Industrial Sanitation in Manufacturing Establishments, 3.3 Ventilation. (Dept. of Labor, State Capitol, Little Rock.)

CALIFORNIA: Reference to the ventilation laws may be found in the California Health and Safety Code under the following sections: Sec. 16800—Construction Requirements, Air Ducts; Sec. 16820-16835—Vent Shafts; Sec. 16900-16905—Gas Appliance Vents; Sec. 17080-17088—Garages; Sec. 16233-16235, 16270-16271—Rooms; Sec. 16300-16305—Stairways. (Supervisor of Documents, 214 State Capitol, Sacramento.)

COLORADO: State Boiler Inspection Code on high-pressure boilers—steel, all cast iron, 15 lb or over. (State Industrial Commission, State Annex Bldg., Denver.)

State-Wide Plumbing Code—Rules and Regulations of the Division of Public Health, regarding ventilation of toilet, shower and wash rooms; vents on gas or oil-fired instantaneous water heaters; construction equipment of sanitary features in all buildings in the state, including ventilation of all plumbing fixtures. (Div. of Public Health, State Office Bldg., Denver.)

Regulations regarding heating, ventilation and lighting. (State Coal Mine Inspector, Argonaut Hotel, Denver.)

CONNECTICUT: School Building Code (1941); Chapter 6—Structural and Mechanical, D. Heating and Ventilation (State Dept. of Education. Div. of Instruction, Hartford.)

Statutes: Section 2566 specifies the number of cubic feet of air for children and adults in tenement and boarding houses.
Section 2584 specifies the dimensions of windows and skylights in tenement house public halls.
Section 2585, amd. by 443 f, specifies ventilation required for bathrooms and toilets in tenement houses.
Section 2659 concerns heating of leased buildings.

Sanitary Code: Regulations 280 and 281 concern ventilation to be provided for industrial processes. Regulation 127 concerns ventilation of all new water closet compartments. (State Dept. of Health, Hartford.)

Labor Laws (1941): VIII. Industrial Safety, A. General, Sec. 2355— Lighting and sanitary condition of factories and round houses: IX. Industrial Health and Sanitation, Legislation, Sec. 2355—Lighting and sanitary condition of factories and round houses. (Dept. of Labor and Factory Inspection, Hartford.)

DELAWARE: Minimum Standards for School Buildings and Sites (1931). (State Board of Education, Dover.)

DISTRICT OF COLUMBIA: Building Code (1941), Section 505—Mechanical Ventilation; Section 702, Article 702-04—Construction of Air Conditioning Ducts. (Dept. of Printing and Binding, Govt. District of Columbia.)

FLORIDA: School Code Law; Laws Relating to Construction Hotels, Apartment Houses, and Public Eating Places. (Secy. of State, Tallahassee.)

GEORGIA: None.

IDAHO: None.

ILLINOIS: Laws Relating to Labor and Employment (1943) Health and Safety Act; Workmen's Occupational Diseases Act; Wash Rooms in Certain Employments; Work Under Compressed Air Act; Industrial Home Work. (Secy. of State, Springfield.)

Illinois School Law, Section 15, paragraph 20 is amplified in April, 1940, Educational Press Bulletin with sections on Ventilating and Heating. (Superintendent of Public Instruction, Springfield.)

Jurisdiction over lodging houses, boarding houses, taverns, inns and hotels, specifying the number of cu ft of air space for each person sleeping therein at the same time. (Dept. of Public Health, Springfield.)

General Mining Laws, 14. Ventilation—Quantity of Air. (Dept. of Mines and Minerals, Springfield.)

INDIANA: Rules and Regulations of the State Board of Health Governing the Construction, Equipment and Maintenance of Sanitary Features of Public and Parochial School Buildings (1943). (Division of Environmental Sanitation, State Board of Health, 1098 W. Michigan St., Indianapolis.)

Rules and Regulations of the Administrative Building Council—Article VIII, Sections 19-8-1, 19-8-2 and 19-8-3 Pertaining to Ventilation of Rooms and Plumbing Fixtures.

Sanitary Dwelling Law.

Tourist Camp License Law. (Ind. State Board of Health, Indianapolis.)

IOWA: Code of Iowa, Chapter 323, Requirements for Sanitation and Ventilation of Single Family and Multiple Dwellings. Code of Iowa, Chapter 73, Ventilation Requirements in Industrial Establishments Pertaining to Health and Safety. Department of Public Instruction Rules Relating to Heating and Ventilating of School Buildings. (Dept. of Health, Div. Public Health Engr. and Industrial Hygiene, Des Moines.)

KANSAS: Statutes of Kansas: G. S. 1935, 44-636—Leaves matter to judgment of Factory Inspector in safeguarding Safety and Health of Workers in factories under jurisdiction of State Labor Commissioner. (State Labor Department, 801 Harrison, Topeka.)

Statutes of Kansas: G. S. 1935, 49-204, 49-205, 49-216, 49-217. Ventilation of mines under jurisdiction of State Labor Commissioner.

G. S. 1935, 36-110; Ventilation in Hotels, Lodging Houses and Restaurants. (State Hotel and Restaurant Board, State House, Topeka.)

KENTUCKY: Statutes of Kentucky: Sections 101.010 to 101.990, relating to ventilation of tenement and apartment houses in cities of first class; Section 217.280 provides for the proper ventilation of food establishments; Sections 228.080, 228.090, 228.120, 228.140, 228.150 and 228.170, relating to heating and ventilating of dry cleaning and dyeing establishments; Sections 338.070, 338.080 and 338.090, relating to ventilation in rooms containing buffing or grinding machinery; Sections 352.020, 352.030, 352.040, 352.050, 352.080, 352.090, 352.120, 352.150, 352.190, 352.210, 352.320, 352.330, 352.340, 352.350, 352.360, 352.370, 352.380, 352.390 and 352.400; also, 352.570, 352.580, 352.590, 352.600 and 352.610 provide for the proper ventilation of mines. (Kentucky Statute Revision Commission, Frankfort.)

TABLE 1. STATE CODES, STANDARDS OR LAWS RELATING TO THE HEATING, VENTILATING OR AIR CONDITIONING OF BUILDINGS—(Concluded)

LOUISIANA: None.

MAINE: Regulations bearing on this subject in reference to the installation of plumbing fixtures in closed rooms are outlined in Sections 101 and 102 of the State Plumbing Code.

Rules and Regulations Relating to Sanitation of Factories and Mercantile Establishments include Sections on Ventilation. (Division of Sanitary Engineering. Dept. of Health and Welfare, State House, Augusta.)

MARYLAND: Standard for Maryland School Buildings, Revised, 1941. (State Dept. of Education, Baltimore.)

MASSACHUSETTS: Laws Relating to the Erection, Alteration, Inspection and Use of Buildings (Form A); Regulations Relative to the Inspection of Buildings Which Are Subject to the Provisions of Chapter 143, General Laws (Form B-1). Ventilation of Motion Picture Booths, Laws, Rules and Regulations Form C. (Dept. of Public Safety, Division of Inspection, 1010 Commonwealth Ave., Boston.)

MICHIGAN: Housing Law of Michigan (1939). (Secretary of State, Capitol Bldg., Lansing.)

MINNESOTA: Laws Relating to Sanitation, Ventilation and Toilets in Factories, etc. Also Requirements in regard to garages, spray rooms and spray booths.

General Orders on Dusts, Fumes, Vapors and Gases. (Industrial Commission, State Office Bldg., St. Paul.)

MISSISSIPPI: Sanitary and Safety Regulations for Industrial Establishments. (Mississippi State Board of Health, Jackson.)

MISSOURI: State Labor and Industrial Inspection Laws (1944). (Dept. of Labor and Industrial Inspection, State Office Bldg., Jefferson City.)

MONTANA: Revised Codes of Montana (1935), Section 1175 refers to ventilation of school buildings. (Secretary of State, Helena.)

NEBRASKA: Safety Codes (1937) and Labor Laws (1943). (Dept. of Labor, Lincoln.)

NEVADA: Nevada Compiled Laws (1929), Sections 5894 and 5715 pertain to School Buildings. Compiled Labor Laws (1937), Part 14, Section 4241 Ventilation of Mines; Part 8, Sec. 2817 Ventilation Bunk Houses. (Secretary of State, Carson City.)

NEW HAMPSHIRE: Under the provisions of Chapter 215 of the Revised Laws, Safety and Health of Employees, recommendations are made in mills, factories, workshops, commercial and mercantile establishments for proper ventilation. (Bureau of Labor, Concord.)

NEW JERSEY: Industrial Code Bulletin and Labor Laws; Sanitary and Engineering Industrial Standards; Abrasive Wheels, Use, Care and Protection; Safety Code for Work in Compressed Air; Foundry Code. (State Dept. of Labor, Wallach Bldg., Trenton.)

NEW MEXICO: Regulations Governing the Heating and Ventilation of Tourist Courts, Tourist Camps, Hotels and Lodging Houses (1939). (Dept. of Public Health, Santa Fe.)

NEW YORK: Commissioner of Education is authorized to determine the requirements for proper ventilation of school buildings. (Commissioner of Education, Albany.)

Codes are enforced by the Department of Labor which contain sections on ventilation requirements pertaining to a number of industrial processes, such as: No. 10 Foundry Code; No. 12 Dust, Fumes and Gases; No. 32 Automobile Spray; No. 33 Rock Drilling; No. 34 Stone Crushing; No. 35 Stone Cutting and Finishing; Nos. 9 and 16 Sanitary Code; No. 17 Mines and Quarries; No. 25 Removal of Toxic Gases in Mines, Tunnels and Shafts; No. 37 Military Pyrotechnics. (Secretary, Labor Dept., 80 Centre St., New York.)

NORTH CAROLINA: North Carolina Building Code (1941) Chapter 14, Heating and Mechanical Ventilation. (North Carolina Insurance Dept., Raleigh.)

NORTH DAKOTA: None.

OHIO: Ohio State Building Codes: No. 102 Theaters and Assembly Halls (1940); No. 103 School Buildings (1938); No. 105 Churches (1938); No. 106 Hospitals and Homes (1936); No. 107 Hotels and Apartments (1940); No. 108 Public Garages (1938); No. 109 (1941); Workshops, Factories, Mercantiles and Office Buildings. See also Rule 16, Ventilation (adopted 1932 by the Ohio Board of Building Standards). (Dept. of Industrial Relations, Div. of Factory and Building Inspection, Columbus.)

OKLAHOMA: Bureau of Factory Inspection Bulletin No. 7-A, containing the laws governing the inspection and regulation of factories or other places where labor is employed. Bureau of Factory Inspection Book No. 11-A—Petroleum Industry Safety Standards. (Dept. of Labor, Oklahoma City.)

OREGON: General Safety Manual; Logging Safety Code; Sawmill and Woodworking and Allied Industries Safety Code; Safety Code for Construction Work; Electrical and Communication Workers Safety Code; Safety Code for Arc Welding; Sandblasting and Spray Painting Operations. (Industrial Accident Commission, Salem.)

State laws and regulations relating to school buildings, sanitation, plumbing, heating and ventilation, including factory inspections. (State Board of Health, Oregon Bldg., Portland.)

PENNSYLVANIA: Regulations For: Abrasive and Polishing Wheels; Bedding and Upholstery; Brewing and Bottling; Canneries; Cereal Mills, Malt Houses and Grain Elevators; Construction and Repairs; Dry Cleaning; Dry Color Industry; The Storage, Handling and Use of Explosives; Electric Safety Code; Fire and Panic Regulations; Foundries; General Safety Act; Industrial Sanitation; Lead Manufacturing; Paint Grinding; Regulations for Labor Camps; Laundries; Lead Corroding and Lead Oxidising; Logging, Sawmill, Woodworking, Veneer and Cooperage Operations; Mines other than Coal Mines; Miscellaneous Hazards and Conditions of Employment; the Manufacture of Nitro and Amido Compounds; Plants Manufacturing or Using Explosives; Printing and Allied Industries; Spray Coating; Tunnel Construction and Work in Compressed Air; Textile Industries. (Dept. of Labor and Industry, Harrisburg.)

RHODE ISLAND: Rhode Island Labor Laws (1942). (Dept. of Labor, State House, Providence.)

SOUTH CAROLINA: None.

SOUTH DAKOTA: State Code Section 13.2018 Care of Steam Boilers; Section 27.1711 Ventilation of Hotels, Rooming Houses, Restaurants, Tourist Camps. (Office of State Engineer, Pierre.)

TENNESSEE: Williams Code of Tennessee, Sections 5342-5343, which deal with factory and workshop ventilation, places of amusement, etc. and the State Agencies charged with responsibility therefor. (Secy. of State, Nashville.)

TEXAS: School Building Law, Bulletin 382, February, 1938, Article No. 2920, No. 2921, No. 2922. (Dept. of Education, Austin.)

UTAH: 35-1-16. Sanitary Code for Public Buildings and Railway Cars; 35-1-17. Supervision of Bathing Places. (Dept. of Engineering, 439 State Capitol Bldg., Salt Lake City.)

VERMONT: Public Laws, Rules and Regulations Relating to Public Buildings, Sanitation, Plumbing, Heating and Ventilation (1941). (State Board of Health, Burlington.)

VIRGINIA: Mining Laws of State of Virginia have provisions regarding ventilation in mines. (Dept. of Labor and Industry, Finance Bldg., Richmond.)

WASHINGTON: General Safety Standards: Standard No. 49 Blower and Exhaust Systems; No. 50 Respirators, Helmets, etc.; No. 51 Carbon Monoxide Gas; No. 49 Ventilation; No. 210 Laundry Ventilation; No. 154 Dry Cleaning with Volatile, Inflammable or Explosive Liquids. Coal Mining Laws; Occupational Disease Code; Metal Mine Standards; Safety of Persons Employed in Tunnels, Quarries, Caissons or Subways; Construction Code. (Dept. of Labor and Industries, Olympia.)

State also has approved the *A.S.M.E.* Boiler Codes and Liquified Petroleum Gases Standards of the *N.B.F.U.*

WEST VIRGINIA: *A.S.M.E.* Boiler Code for Construction and annual certificates for operation required for all steam boilers operating at more than 15 psi. (W. Va. Dept. of Labor, Charleston.)

WISCONSIN: Heating, Ventilation and Air Conditioning Code (1939). Applies to public buildings and places of employment. General Orders on Spray Coating (1939). General Orders on Dusts, Fumes, Vapors and Gases (1941). (Industrial Commission of Wisconsin, Madison.)

WYOMING: None.

TABLE 2. CODES AND STANDARDS PREPARED AND ACCEPTED BY VARIOUS SOCIETIES AND ASSOCIATIONS

SUBJECT	TITLE	SPONSOR	REFERENCE
Air Conditioning	Code of Minimum Requirements for Comfort Air Conditioning (1938).	A.S.H.V.E. *A.S.R.E.*	A.S.H.V.E.
Air Conditioning (150,000 Btu/Hr or less)	Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (1945).	*N.W.A.H. & A.C.A.*	*N.W.A.H. & A.C.A.* No. 7
Air Conditioning (Above 150,000 Btu/Hr)	The Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems (1942).	*N.W.A.H. & A.C.A.*	*N.W.A.H. & A.C.A.*
Airplane	Aeronautical Recommended Practice for Heating and Ventilating Airplanes (1943).	*S.A.E.*	*S.A.E.* ARP 85
Airplane	Aeronautical Recommended Practice for Internal Combustion Type Airplane Heaters (1945).	*S.A.E.*	*S.A.E.* ARP 143A
Boilers	I=B=R Testing and Rating Codes for Low Pressure Heating Boilers (1945).	*I.B.R.*	*I.B.R.*
Boilers	Net Square Feet Radiation Loads in 70 Deg Fahr, Recommended for Low Pressure Heating Boilers (1943).	*H.P. & A.C.C.N.A.*	*H.P. & A.C.C.N.A.*
Boilers	Net Load Recommendations for Heating Boilers.	*H.P. & A.C.C.N.A.*	*H.P. & A.C.C.N.A.*
Boilers	Standard and Short Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2)-(1929).	A.S.H.V.E.	A.S.H.V.E.
Boilers	A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) (1929).	A.S.H.V.E.	A.S.H.V.E.
Boilers	A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel (1932).	A.S.H.V.E.	A.S.H.V.E.
Boilers	A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand Fired Boilers (Revised April 1930).	A.S.H.V.E.	A.S.H.V.E.
Boilers	A.S.H.V.E. Standard Code for Testing Stoker-Fired Steam-Heating Boilers (1938).	A.S.H.V.E.	A.S.H.V.E.
Boilers	*A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.	*A.S.M.E.*	*A.S.M.E.*
Boilers (Steel)	Steel Boiler Institute Rating Code for Commercial Steel Boilers and Residential Steel Boilers (1945).	*S.B.I.*	*S.B.I.*
Boilers (Steel)	Simplified Practice Recommendation for Steel Firebox Heating Boilers (1937).	*S.B.I.*	*B.S.* R157-37
Burners (Gas)	American Standard Testing Requirements for Gas Conversion Burners (1941).	*A.G.A.*	*A.S.A.* Z21.17-1940
Burners (Gas)	American Standard Requirements for Installation of Gas Burning Equipment in Power Boilers (1942).	*A.G.A.*	*A.S.A.* Z21.33-1942
Burners (Anthracite)	Commercial Standard for Domestic Burners for Pennsylvania Anthracite (Underfeed Type) (1940).	*B.S.* *A.I.L.*	*B.S.* CS48-40
Burners (Oil)	Commercial Standard for Mechanical-Draft Oil Burners Designed for Domestic Installations (1942).	*B.S.* *O.B.I.*	*B.S.* CS75-42
Cleaners (Air)	A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (1934).	A.S.H.V.E.	See A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 225
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1945).	*H.P. & A.C.C.N.A.*	*H.P. & A.C.C.N.A.* Engrg. Stds., Sec. 2, Part V
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1928).	*A.S.M.E.*	*A.S.A.* A13-1928
Coils	Proposed Commercial Standard for Rating and Testing Air Cooling Coils Using Non-Volatile Refrigerants (1945).	*B.C.M.I.* *B.S.*	*B.S.* T.S. 4044
Compressors	Tentative *A.S.R.E.* Standard Methods of Rating and Testing Refrigerant Compressors.	*A.S.R.E.* A.S.H.V.E. *A.C.R.M.A.*	*A.S.R.E.* Circular No. 23

TABLE 2. CODES AND STANDARDS PREPARED AND ACCEPTED BY VARIOUS SOCIETIES AND ASSOCIATIONS—(Continued)

SUBJECT	TITLE	SPONSOR	REFERENCE
Condensers	*A.S.R.E.* Standard Methods of Rating and Testing Evaporative Condensers.	*A.S.R.E.* A.S.H.V.E. *A.C. & R.M.A.*	*A.S.R.E.* Circular No. 20
Condensers	Tentative *A.S.R.E.* Standard Methods of Rating Water-Cooled Refrigerant Condensers.	*A.S.R.E.* A.S.H.V.E. *A.C. & R.M.A.*	*A.S.R.E.* Circular No. 22
Condensing Units	*A.S.R.E.* Standard Methods of Rating and Testing Mechanical Condensing Units (1940).	*A.S.R.E.* A.S.H.V.E. *A.C. & R.M.A.*	*A.S.R.E.* Circular No. 14-41
Condensing Units	Commercial Standard for Commercial Electric Refrigeration Condensing Units (1945).	*B.S.* *S.R.C.A.*	*B.S.* CS107-45
Conductivity	Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate (Tentative) (1942).	A.S.H.V.E. *A.S.R.E.* *A.S.T.M.* *N.B.S.*	A.S.H.V.E.
Convector	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) (1931).	A.S.H.V.E.	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367
Convector	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section) (1933).	A.S.H.V.E.	A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 237
Coolers (Air)	Proposed *A.S.R.E.* Standard Methods of Rating and Testing Forced Circulation and Natural Convection Air Coolers for Refrigeration (1945).	*A.S.R.E.* A.S.H.V.E. *A.C. & R.M.A.* *R.E.M.A.*	*A.S.R.E.* Circular No. 25-44
Coolers	Tentative *A.S.R.E.* Standard Methods of Rating and Testing Water and Brine Coolers.	*A.S.R.E.* A.S.H.V.E. *A.C. & R.M.A.*	*A.S.R.E.* Circular No. 24
Cooling Units	Standard Methods of Rating and Testing Self Contained Air Conditioning Units for Comfort Cooling (1940).	*A.S.R.E.* A.S.H.V.E. *R.M.A.* *N.E.M.A.* *A.C.M.A.*	*A.S.R.E.* Circular No. 16
Ducts and Fittings	Simplified Practice Recommendation for Pipes, Ducts and Fittings for Warm Air Heating and Air Conditioning (1945).	*Mfrs.* *B.S.*	*B.S.* R207-45
Exchangers (Heat)	Standards of *Tubular Exchanger Manufacturers Association* (1941).	*T.E.M.A.*	*T.E.M.A.*
Exhaust Systems	American Standard for Grinding, Polishing, and Buffing Equipment Sanitation (1941).	*A.F.A.*	*A.S.A.* Z43-1941
Exhaust Systems	Tentative Code of Recommended Practices for Testing and Measuring Air Flow in Exhaust Systems (1937).	*A.F.A.*	*A.F.A.* Preprint 36-27
Exhaust Systems	Tentative Recommended Good Practice Code and Handbook on the Fundamentals of Design, Construction, Operation and Maintenance of Exhaust Systems.	*A.F.A.*	*A.F.A.*
Fans	Standard Test Code for Centrifugal and Axial Fans (1938).	*N.A.F.M.* A.S.H.V.E.	*N.A.F.M.* Bulletin No. 103
Fans	*N.E.M.A.* Fan Standards (1944).	*N.E.M.A.*	*N.E.M.A.* Publ. 44-95
Fire Prevention	Building Code Recommended by the *National Board of Fire Underwriters* (1943).	*N.B.F.U.*	*N.B.F.U.*
Fire Prevention	National Fire Codes (1944).	*N.F.P.A.*	*N.F.P.A.*
Fire Prevention	National Fire Code for the Prevention of Dust Explosions (1943).	*N.F.P.A.*	*N.F.P.A.*
Furnaces (Duct)	American Standard Approval Requirements for Gas-Fired Duct Furnaces (1942).	*A.G.A.*	*A.S.A.* Z21.34-1942
Furnaces (Gas, Floor)	Commercial Standard for Gas Floor Furnaces—Gravity Circulating Type (1942).	*B.S.* *A.G.A.E.M.*	*B.S.* CS99-42
Furnaces (Gas)	American Standard Approval Requirements for Central Heating Gas Appliances (1943).	*A.G.A.*	*A.S.A.* Z21.13-1943
Furnaces (Forced Air, Solid-Fuel)	Commercial Standard for Solid-Fuel-Burning Forced Air Furnaces (1944).	*F.H.A.* *N.W.A.H. & A.C.A.* *A.I.L.*	*B.S.* CS109-44

TABLE 2. CODES AND STANDARDS PREPARED AND ACCEPTED BY VARIOUS SOCIETIES AND ASSOCIATIONS—(Continued)

SUBJECT	TITLE	SPONSOR	REFERENCE
Furnaces (Oil-Fired)	Commercial Standard for Warm Air Furnaces Equipped with Vaporizing Pot Type Oil Burners (1943).	*Mfrs.* *B.S.*	*B.S.* CS(E)104-43
Furnaces (Oil)	A Tentative Code for Testing Oil-Fired Furnaces.	*N.W.A.H. & A.C.A.*	*N.W.A.H. & A.C.A.*
Furnaces (Oil)	Commercial Standard for Oil Burning Floor Furnaces Equipped with Vaporizing Pot-Type Burners (1944).	*B.S.* *O.P.A.*	*B.S.* CS113-44
Garages	Code of Minimum Requirements for Heating and Ventilating Garages (1935).	A.S.H.V.E.	A.S.H.V.E.
Gases (Toxic) and Dust	American Standard Allowable Concentration of Harmful Gases:	*A.S.A.*	*A.S.A.*
	Carbon Monoxide		Z37.1-1941
	Hydrogen Sulfide		Z37.2-1941
	Carbon Di-sulfide		Z37.3-1941
	Benzene		Z37.4-1941
	Cadmium		Z37.5-1941
	Manganese		Z37.6-1942
	Chromic Acid and Chromates		Z37.7-1943
	Mercury		Z37.8-1943
	Metallic Arsenic and Arsenic Trioxide		Z37.9-1943
	Xylene		Z37.10-1943
	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
Mineral Wool	Commercial Standard for Mineral Wool: Blankets, Blocks, Insulating Cement, and Pipe Insulation for Heated Industrial Equipment (1944).	*B.S.* *I.M.W.I.*	*B.S.* CS117-44
Mineral Wool	Commercial Standard for Mineral Wool: Loose, Granulated, or Felted Form, in Low Temperature Installations (1943).	*B.S.* *I.W.M.I.*	*B.S.* CS105-43
Mineral Wool	Recommended Commercial Standard for Industrial Mineral Wool Products—All Types—Testing and Reporting.	*I.M.W.A.* *B.S.*	*B.S.* TS-4030
Piping	American Standard Code for Pressure Piping (1942).	*A.S.M.E.*	*A.S.A.* B31.1-1942
Pumps	Hydraulic Institute Test Code for Centrifugal Pumps. Hydraulic Institute Test Code for Rotary Pumps (1943).	*H.I.*	*H.I.* Section F
Radiators	Code for Testing Radiators (1927).	A.S.H.V.E.	A.S.H.V.E.
Radiators	Simplified Practice Recommendation for Cast Iron Radiators (1943).	*I.B.R.* *B.S.*	*B.S.* R174-43
Refrigeration (Mechanical)	American Standard Safety Code for Mechanical Refrigeration (1939).	*A.S.R.E.*	*A.S.R.E.* Circular No. 15
Refrigerators (Household)	American Standard Test Procedures for Household Electric Refrigerators (Mechanically Operated) (1944).	*A.S.R.E.* *U.S.D.A.*	*A.S.A.* B38.2-1944
Sound	Sound Measurement Test Code for Centrifugal and Axial Fans (1942).	*N.A.F.M.*	*N.A.F.M.* Bulletin No. 104
Sound	American Recommended Practice for the Calibration of Microphones (1938).	*A.S. of A.*	*A.S.A.* Z24.4-1938
Space Heaters	American Standard Approval Requirements for Gas Space Heaters (1942).	*A.G.A.*	*A.S.A.* Z21.11-1942
Space Heaters	Commercial Standard for Flue Connected Oil-Burning Space Heaters Equipped with Vaporizing Pot-Type Burners (1943).	*I.C. & H.A.M.*	*B.S.* CS101-43
Stokers	Code for Determination of Rated Capacities of Anthracite Underfeed Stokers (1944).	*S.M.A.*	*S.M.A.*
Stokers	Code for Determination of Rated Capacities of Bituminous Underfeed Stokers (1944).	*S.M.A.*	*S.M.A.*

TABLE 2. CODES AND STANDARDS PREPARED AND ACCEPTED BY VARIOUS SOCIETIES AND ASSOCIATIONS—(Concluded)

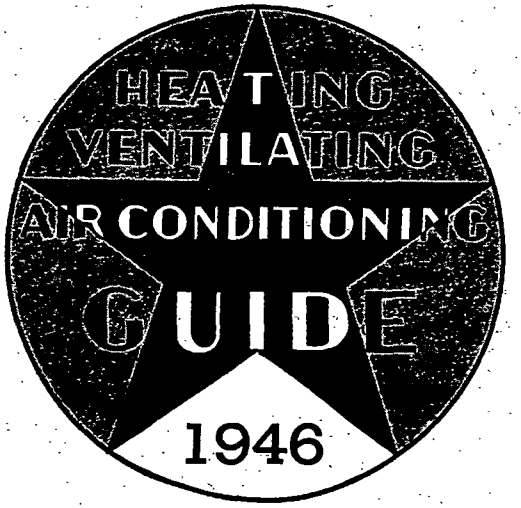
SUBJECT	TITLE	SPONSOR	REFERENCE
Stokers	Recommended Minimum Firebox Dimensions and Base Heights (1944).	*S.M.A.*	*S.M.A.*
Stokers	Recommended Standards Governing Minimum Setting Heights (1944).	*S.M.A.*	*S.M.A.*
Unit Heaters	Standard Code for Testing and Rating Steam Unit Heaters (1930).	A.S.H.V.E. *I.U.H.A.*	A.S.H.V.E. *I.U.H.A.*
Unit Heaters	Standard Code for Testing Hot Water Unit Heaters (1942).	*I.U.H.A.*	*I.U.H.A.*
Unit Heaters	American Standard Approval Requirements for Gas Unit Heaters (1940).	*A.G.A.*	*A.S.A.* Z21.16-1940
Unit Ventilators	A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators (1934).	A.S.H.V.E.	A.S.H.V.E.
Vacuum Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps (1934).	A.S.H.V.E.	A.S.H.V.E.
Warm Air (Gravity)	Gravity Code and Manual for the Design and Installation of Gravity Warm Air Heating Systems (1945).	*N.W.A.H. & A.C.A.*	*N.W.A.H. & A.C.A.* Section No. 5
Water Heaters	*N.E.M.A.* Standards for Electric Water Heaters (1945).	*N.E.M.A.*	*N.E.M.A.* 45-104
Water Heaters	American Standard Approval Requirements for Gas Water Heaters (1944).	*A.G.A.*	*A.S.A.* Z21.10-1944

ABBREVIATIONS

*A.C.M.A.**	Air Conditioning Manufacturers Association.
A.C. & R.M.A.	Air Conditioning and Refrigerating Machinery Association.
A.F.A.	American Foundrymen's Association.
A.G.A.	American Gas Association.
A.G.A.E.M.	Association of Gas Appliance and Equipment Manufacturers.
A.I.L.	Anthracite Industries Laboratory.
A.S.A.	American Standards Association.
A.S. of A.	Acoustical Society of America.
A.S.H.V.E.	American Society of Heating and Ventilating Engineers.
A.S.R.E.	American Society of Refrigerating Engineers.
A.S.T.M.	American Society for Testing Materials.
B.S.	National Bureau of Standards.
F.H.A.	Federal Housing Administration.
H.I.	Hydraulic Institute.
H.P. & A.C.C.N.A.	Heating, Piping and Air Conditioning Contractors National Association.
I.B.R.	Institute of Boiler and Radiator Manufacturers Association.
I.C.H.A.M.	Institute of Cooking and Heating Appliance Manufacturers.
I.M.W.I.	Industrial Mineral Wool Institute.
I.U.H.A.	Industrial Unit Heater Association.
N.A.F.M.	National Association of Fan Manufacturers.
N.E.M.A.	National Electrical Manufacturers Association.
N.F.P.A.	National Fire Prevention Association.
N.R.C.	National Research Council.
O.B.I.	Oil Burner Institute.
O.P.A.	Office of Price Administration.
R.E.M.A.	Refrigeration Equipment Manufacturers Association.
*R.M.A.**	Refrigerating Machinery Association.
S.A.E.	Society of Automotive Engineers.
S.B.I.	Steel Boiler Institute.
S.R.C.A.	Standard Refrigeration Compressor Association.
T.E.M.A.	Tubular Exchanger Manufacturers Association.
U.S.D.A.	United States Department of Agriculture.

*Superseded 1940 by *A.C. & R.M.A.*

CATALOG DATA SECTION



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Clarage Fan Co., 946
Hastings Air Conditioning Co., Inc., 947
Lau Blower Co., The, 1065
Morrison Products, Inc., 1066
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Torrington Mfg. Co., The, 1070-1071
United States Air Conditioning Corp., 953
Utility Appliance Corp., 1073
Viking Air Conditioning Corp., The, 1074

BLOWER MOTORS (*See Motors, Electric*)

BLOWERS, Centrifugal (*See Fans*)

BLOWERS, Fan (*See Fans, Supply and Exhaust*)

BLOWERS, Forced Draft
American Blower Corp., 940-941
American Coolair Corp., 1050-1051
American Foundry & Furnace Co., 962-963
Bayley Blower Co., 1052
Buffalo Forge Co., 1056
Campbell Heating Co., 964-965
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
La-Del Conveyor & Mfg. Co., 1062-1063
Marlo Coil Co., 1042
National Heater Co., 991
Herman Nelson Corp., The, 992-993
New York Blower Co., The, 1067
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Trane Co., The, 998-999
Utility Appliance Corp., 1073
L. J. Wing Mfg. Co., 1000-1002

BLOWERS, Heating and Ventilating
Air Controls, Inc., 1049
American Blower Corp., 940-941
American Coolair Corp., 1050-1051
American Foundry & Furnace Co., 962-963
Bayley Blower Co., 1052
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
Campbell Heating Co., 964-965
Carrier Corp., 944-945
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
Electromode Corp. (Div., American Foundry Equipment Co.), 977
Hartzell Propeller Fan Co., 1064
Hastings Air Conditioning Co., Inc., 947
Ilg Electric Ventilating Co., 982, 1060-1061
Jaden Mfg. Co., 983
La-Del Conveyor & Mfg. Co., 1062-1063
Lau Blower Co., The, 1065
Morrison Products, Inc., 1066
Herman Nelson Corp., The, 992-993
New York Blower Co., The, 1067
Schwitzer-Cummins Co., 1072
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Torrington Mfg. Co., The, 1070-1071
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
Utility Appliance Corp., 1073
L. J. Wing Mfg. Co., 1000-1002

BLOWERS, Pressure
American Blower Corp., 940-941
American Coolair Corp., 1050-1051
Bayley Blower Co., 1052
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
DeBothezat Fans Div., American Machine & Metals, Inc., 1059
Hartzell Propeller Fan Co., 1064
Ilg Electric Ventilating Co., 982, 1060-1061
La-Del Conveyor & Mfg. Co., 1062-1063
Jos. A. Martocello & Co., 1028
Nash Engineering Co., 1202-1203
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019

BLOWERS, Turbine
DeBothezat Fans Div., American Machine & Metals, Inc., 1059
Ilg Electric Ventilating Co., 982, 1060-1061
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
L. J. Wing Mfg. Co., 1000-1002

BLOWERS, Warm Air Furnace
Air Controls, Inc., 1049
Buffalo Forge Co., 1056
Campbell Heating Co., 964-965
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
Hastings Air Conditioning Co., Inc., 947
Ilg Electric Ventilating Co., 982, 1060-1061
La-Del Conveyor & Mfg. Co., 1062-1063
Lau Blower Co., The, 1065
Meyer Furnace Co., The, 969
Morrison Products, Inc., 1066
L. J. Mueller Furnace Co., 970-971
Schwitzer-Cummins Co., 1072
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
Utility Appliance Corp., 1073
Viking Air Conditioning Corp., The, 1074
L. J. Wing Mfg. Co., 1000-1002

BOILER-BURNER
Air Devices, Inc., 970-971
Airtemp Div., Chrysler Corp., 966-967
American Radiator & Standard Sanitary Corp., 1148-1149
Automatic Burner Corp., 1181
Bryant Heater Co., The, 961
Crane Co., 1154-1155
Gar Wood Industries, Inc., 974-975
General Electric Co. (Bloomfield, N. J.), 948-949
National Radiator Co., The, 1164-1165
Petroleum Heat & Power Corp., 1186-1187
Preferred Utilities Mfg. Corp., 1166
Terre Haute Boiler Works Co., 1170
Webster Engineering Co., The, 4180
York-Shipley, Inc., 1191

BOILER COMPOUNDS (*See Compounds, Boiler*)

BOILER COVERING (*See Covering, Pipe and Surfaces*)

BOILER FEED PUMPS (*See Pumps, Boiler Feed*)

BOILER FEEDERS (*See Feeders, Boiler*)

BOILER WATER FIELD KIT, for Testing and Treating
Vinco Co., Inc., The, 1192-1193

BOILER WATER TREATMENT
Vinco Co., Inc., The, 1192-1193

BOILERS, Cast-Iron
Airtemp Div., Chrysler Corp., 966-967
American Radiator & Standard Sanitary Corp., 1148-1149
Bryant Heater Co., The, 961
Burnham Boiler Corp., 1152
Crane Co., 1154-1155
National Radiator Co., The, 1164-1165
H. B. Smith Co., Inc., The, 1167
Spencer Heater Div., The Aviation Corp., 1168-1169
United States Radiator Corp., 1172-1173
Weil-McLain Co., 1174
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190

BOILERS, Down Draft
Farrar & Trefts, Inc., 1158
Kewanee Boiler Corp., 1160-1163
H. B. Smith Co., Inc., The, 1167
Terre Haute Boiler Works Co., 1170

BOILERS, Forced Recirculation—Oil Burning
Air Devices, Inc., 1080-1081
Airtemp Div., Chrysler Corp., 966-967
Cyclotherm Corp., 1153
International Boiler Works Co., The, 1159
H. B. Smith Co., Inc., The, 1167

BOILERS, Gas Burning
Airtemp Div., Chrysler Corp., 966-967
American Radiator & Standard Sanitary Corp., 1148-1149
Babcock & Wilcox Co., The, 1150
Brownell Co., The, 1175
Bryant Heater Co., The, 961
Burnham Boiler Corp., 1152
Crane Co., 1154-1155
Cyclotherm Corp., 1153
Farrar & Trefts, Inc., 1158

Fitzgibbons Boiler Co., Inc., 1156-1157
General Electric Co. (Bloomfield, N. J.), 948-949
International Boiler Works Co., 1159
Kewanee Boiler Corp., 1160-1163
L. J. Mueller Furnace Co., 970-971
National Radiator Co., The, 1164-1165
Pacific Steel Boiler Div., U. S. Radiator Corp., 1171
H. B. Smith Co., Inc., The, 1167
Spencer Heater Div., The Aviation Corp., 1168-1169
Terre Haute Boiler Works Co., 1170
Trane Co., The, 998-999
United States Radiator Corp., 1172-1173

BOILERS, Heating

Airtemp Div., Chrysler Corp., 966-967
American Radiator & Standard Sanitary Corp., 1148-1149
Babcock & Wilcox Co., The, 1150
Bigelow Co., The, 1151
Brownell Co., The, 1175
Bryant Heater Co., The, 961
Burnham Boiler Corp., 1152
Cyclotherm Corp., 1153
Farrar & Trefts, Inc., 1158
Fitzgibbons Boiler Co., Inc., 1156-1157
Gar Wood Industries, Inc., 974-975
International Boiler Works Co., The, 1159
Iron Fireman Mfg. Co., 1178-1179
Kewanee Boiler Corp., 1160-1163
L. J. Mueller Furnace Co., 970-971
National Radiator Co., The, 1164-1165
Pacific Steel Boiler Div., U. S. Radiator Corp., 1171
Petroleum Heat & Power Corp., 1186-1187
Preferred Utilities Mfg. Corp., 1166
H. B. Smith Co., Inc., The, 1167
Spencer Heater Div., The Aviation Corp., 1168-1169
Terre Haute Boiler Works Co., 1170
Trane Co., 998-999
United States Radiator Corp., 1172-1173
Weil-McLain Co., 1174
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190

BOILERS, Magazine Feed

Spencer Heater Div., The Aviation Corp., 1168-1169
Weil-McLain Co., 1174
York-Shipley, Inc., 1191

BOILERS, Oil Burning

Airtemp Div., Chrysler Corp., 966-967
American Radiator & Standard Sanitary Corp., 1148-1149
Automatic Burner Corp., 1181
Babcock & Wilcox Co., The, 1150
Bigelow Co., The, 1151
Brownell Co., The, 1175
Burnham Boiler Corp., 1152
Cyclotherm Corp., 1153
Farrar & Trefts, Inc., 1158
Fitzgibbons Boiler Co., Inc., 1156-1157
Gar Wood Industries, Inc., 974-975
International Boiler Works Co., The, 1159
Iron Fireman Mfg. Co., 1178-1179
S. T. Johnson Co., 1184-1185
Kewanee Boiler Corp., 1160-1163
National Radiator Co., The, 1164-1165
Pacific Steel Boiler Div., U. S. Radiator Corp., 1171
Petroleum Heat & Power Corp., 1186-1187
Preferred Utilities Mfg. Corp., 1166
H. B. Smith Co., Inc., The, 1167
Spencer Heater Div., The Aviation Corp., 1168-1169
Terre Haute Boiler Works Co., 1170
Trane Co., The, 998-999
United States Radiator Corp., 1172-1173
Weil-McLain Co., 1174
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190

BOILERS, Unit Steam Generator

Preferred Utilities Mfg. Corp., 1166

BOILERS, Water Tube

Babcock & Wilcox Co., The, 1150
Bigelow Co., The, 1151
Combustion Engineering Co., 1176
Farrar & Trefts, Inc., 1158
International Boiler Works Co., The, 1159
H. B. Smith Co., Inc., The, 1167
Spencer Heater Div., The Aviation Corp., 1168-1169
Trane Co., The, 998-999

BREECHINGS AND CHIMNEYS

Bethlehem Steel Co., Inc., 1098
Bigelow Co., The, 1151
Brownell Co., The, 1175
Farrar & Trefts, Inc., 1158
Terre Haute Boiler Works Co., 1170

BURNERS, Automatic (*See also Coal Burners, Stokers*)

Combustion Engineering Co., 1176
Detroit Stoker Co., 1177
Gar Wood Industries, Inc., 974-975
General Electric Co. (Bloomfield, N. J.), 948-949
Iron Fireman Mfg. Co., 1178-1179
S. T. Johnson Co., 1184-1185
Preferred Utilities Mfg. Corp., 1166
Ray Oil Burner Co., 1189

BURNERS, Combination Gas and Oil

Babcock & Wilcox Co., The, 1150
Combustion Engineering Co., 1176
Combustion Equipment Div., Todd Shipyards Corp., 1182
S. T. Johnson Co., 1184-1185
Ray Oil Burner Co., 1189
Webster Engineering Co., The, 1180

BURNERS, Gas (*See Gas Burners*)

BURNERS, Oil (*See Oil Burners*)

CALKING, Building

Baldwin-Hill Co., 1249
Eagle-Picher Co., The, 1254
Johns-Manville, 1258-1259
Kimberly-Clark Corp., 1260-1261

CASTINGS, Bronze and Dairy Nickel Silver Metal

Arthur Harris & Co., 1115
D. J. Murray Mfg. Co., 990

CEILING PLATES

Pyle-National Co., The, 1091
United States Register Co., 1095

CELLULAR GLASS (*See Glass, Cellular*)

CEMENT, Asbestos

Philip Carey Mfg. Co., The, 1252-1253
Johns-Manville, 1258-1259
Mundet Cork Corp., 1264
Ruberoid Co., The, 1268-1269
United States Register Co., 1095

CEMENT, Insulating

Baldwin-Hill Co., 1249
Philip Carey Mfg. Co., The, 1252-1253
Eagle-Picher Co., The, 1254
Johns-Manville, 1258-1259
Owens-Corning Fiberglas Corp., 1014-1015
Ruberoid Co., The, 1268-1269
United States Register Co., 1095

CEMENT, Mineral Wool

Baldwin-Hill Co., 1249
Philip Carey Mfg. Co., The, 1252-1253
Eagle-Picher Co., The, 1254
Johns-Manville, 1258-1259
Ruberoid Co., The, 1268-1269

CEMENT, Refractory (*See Refractories*)

CHAIN, Furnace Pulleys (*See also Pulleys, Chain*)

Hart & Cooley Mfg. Co., 1088-1089
Hendrick Mfg. Co., 1087
Revere Copper & Brass, Inc., 1108
United States Register Co., 1095

CHIMNEY TOPS

Air Devices, Inc., 1080-1081
G. C. Breidert Co., 1083

CIRCULATORS, Hot Water Heating

Bell & Gossett Co., 1196-1197
Crane Co., 1154-1155
Hoffman Specialty Co., 1213-1215
Iron Fireman Mfg. Co., 1178-1179
H. A. Thrush & Co., 1198-1199
Taco Heaters, Inc., 1200
Trane Co., The, 998-999

CLEANERS, Air (*See Air Cleanin Equipment*)

COAL BURNERS (*See Burners, Automatic*)

CODE, Fan Test

Propeller Fan Manufacturers Association, 1069

COILS, Aluminum

Kramer Trenton Co., 879
McQuay, Inc., 986-987
John J. Nesbitt, Inc., 994
Refrigeration Economics Co., Inc., 995
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Swan Engineering Co., 1116
Trane Co., The, 998-999

COILS, Ammonia

Acme Industries, Inc., 1030
Baker Ice Machine Co., 1038
G. & O. Manufacturing Co., The, 1035
Kramer Trenton Co., 985
Marlo Coil Co., 1042
McQuay, Inc., 986-987
Modine Mfg. Co., 988-989
Refrigeration Economics Co., Inc., 995
Swan Engineering Co., 1116
Trane Co., The, 998-999
Vilter Mfg. Co., The, 1048
Worthington Pump & Machinery Corp., 1046-1047
Yarnall-Waring Co., 1232
York Corp., 960

COILS, Blast

Aerofin Corp., 1031-1033
Airtherm Mfg. Co., 976
Fedders-Quigan Corp., 980
G & O Mfg. Co., The., 1035
Kennard Corp., 984
Kramer Trenton Co., 985
Marlo Coil Co., 1042
McQuay, Inc., 986-987
Modine Mfg. Co., 988-989
D. J. Murray Mfg. Co., 990
John J. Nesbitt, Inc., 994
New York Blower Co., The, 1067
Niagara Blower Co., 950
Refrigeration Economics Co., Inc., 995
Rome-Turney Radiator Co., 1036
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
Young Radiator Co., 997

COILS, Brass

E. B. Badger & Sons Co., 1010-1011
Arthur Harris & Co., 1115
McQuay, Inc., 986-987
Swan Engineering Co., 1116
Trane Co., The, 998-999

COILS, Cooling

Aerofin Corp., 1031-1032
Bell & Gossett Co., 1196-1197
Fedders-Quigan Corp., 980
Carrier Corp., 944-945
Frick Co. (Inc.), 1041
G & O Mfg. Co., The, 1035
General Electric Co. (Bloomfield, N. J.), 948-949
Insul-Wool Insulation Corp., 1255
Kennard Corp., 984
Kramer Trenton Co., 985
Marlo Coil Co., 1042
McQuay, Inc., 986-987
Modine Mfg. Co., 988-989
John J. Nesbitt, Inc., 994
Refrigeration Economics Co., Inc., 995
Rome-Turney Radiator Co., 1036
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Tenney Engineering, Inc., 1145
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
Vilter Mfg. Co., The, 1048
Worthington Pump & Machinery Corp., 1046-1047
York Corp., 960
Young Radiator Co., 997

COILS, Pipe, Copper

Acme Industries, Inc., 1030
E. B. Badger & Son Co., 1110-1111
Bell & Gossett Co., 1196-1197
Arthur Harris & Co., 1115
Marlo Coil Co., 1042
McQuay, Inc., 986-987
Mueller Brass Co., 1106-1107
Refrigeration Economics Co., Inc., 995
Swan Engineering Co., 1116

COILS, Pipe, Iron

Acme Industries, Inc., 1030
Bayley Blower Co., 1052
Swan Engineering Co., 1116
Worthington Pump & Machinery Corp., 1046-1047

COILS, Pipe and Tube, Non-Ferrous

Acme Industries, Inc., 1034
Arthur Harris & Co., 1115
Kramer Trenton Co., 985
Marlo Coil Co., 1042
John J. Nesbitt, Inc., 994
Mueller Brass Co., 1106-1107
Rome-Turney Radiator Co., 1036
Swan Engineering Co., 1116
York Corp., 960

COILS, Stainless Steel

Arthur Harris & Co., 1115
McQuay, Inc., 986-987
Swan Engineering Co., 1116

COILS, Tank

Bell and Gossett Co., 1196-1197
Taco Heaters, Inc., 1200

COMBUSTION CHAMBERS

Babcock & Wilcox Co., The, 1150
Combustion Engineering Co., 1176
Dravo Corp., 978-979

COMPOUNDS, Boiler

Vinco Co., Inc., The, 1192-1193

COMPOUNDS, Boiler and Radiator Sealing

Dole Valve Co., The, 1234
Johns-Manville, 1258-1259
Vinco Co., Inc., The, 1192-1193

COMPOUNDS, Soot Destroyer

Vinco Co., Inc., The, 1192-1193

COMPRESSOR MOTORS (*See Motors, Electric*)

COMPRESSOR TUBING, Flexible (*See Tubing, Flexible Metallic*)

COMPRESSORS, Air

Brunner Mfg. Co., 1039
Curtis Refrigerating Machine Div. of Curtis Mfg. Co., 1040
Nash Engineering Co., 1202, 1203
Worthington Pump and Machinery Corp., 1046-1047

COMPRESSORS, Refrigeration

Airtemp Div., Chrysler Corp., 966-967
Baker Ice Machine Co., 1038
Brunner Mfg. Co., 1039
Carrier Corp., 944-945
Curtis Refrigerating Machine Div. of Curtis Mfg. Co., 1040
Frick Co. (Inc.), 1041
General Electric Co. (Bloomfield, N. J.), 948-949
Mills Industries, Inc., 1043
Servel, Inc., 1044
Trane Co., The, 998-999
Universal Cooler Div., International Detroit Corp., 1045
Vilter Mfg. Co., The, 1048

B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Worthington Pump & Machinery Corp., 1046-1047
York Corp., 960

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Acme Industries, Inc., 1030
Aerofin Corp., 1031-1033
Baker Ice Machine Co., 1038
Bell & Gossett Co., 1196-1197
Brunner Mfg. Co., 1039
Carrier Corp., 944-945
Condenser Service & Engineering Co., Inc., 1034
Curtis Refrigerating Machine Div. of Curtis Mfg. Co., 1040
Farrar & Trefts, Inc., 1158
Fedders-Quigan Corp., 980
Frick Co. (Inc.), 1041
G & O Mfg. Co., The, 1025
Kramer Trenton Co., 985
Marlo Coil Co., 1042
Niagara Blower Co., 950
J. F. Pritchard & Co., 1025
Refrigeration Economics Co., Inc., 995
Rome-Turney Radiator Co., 1036
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
Universal Cooler Div., International Detrola Corp., 1045
Vilter Mfg. Co., The, 1048
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190
Worthington Pump & Machinery Corp., 1046-1047
Young Radiator Co., 997

CONDUITS, Underground Fittings

American District Steam Co., 1226, 1236
Johns-Manville, 1258-1259
H. W. Porter & Co., Inc., 1237
Ric-wiL Co., The, 1238

CONDUITS, Underground Pipe

American District Steam Co., 1226, 1236
E. B. Badger & Sons Co., 1110-1111
Johns-Manville, 1258-1259
H. W. Porter & Co., Inc., 1237
Ric-wiL Co., The, 1238

CONTROL, Air Volume Damper

Air Devices, Inc., 1080-1081
Anemostat Corp. of America, 1082
Barber-Colman Co., 1084, 1121
Johnson Service Co., 1130-1131
Minneapolis-Honeywell Regulator Co., 1136-1137
Powers Regulator Co., The, 1142
Register & Grille Mfg. Co., Inc., 1094
Tuttle & Bailey, Inc., 1092-1093
Young Regulator Co., 1096

CONTROL, Boiler Water Level

Crane Co., 1154-1155
Leeds & Northrup Co., 1133
McDonnell & Miller, 1194-1195
Mercoid Corp., The, 1134-1135
Penn Electric Switch Co., 1141
Sarco Co., Inc., 1218-1219

CONTROL EQUIPMENT, Combustion

Condenser Service & Engineering Co., Inc., 1034
Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
Fulton Sylphon Co., The, 1124-1125
Illinois Engineering Co., 1216-1217
Iron Fireman Mfg. Co., 1178-1179
Leeds & Northrup, 1133
Mercoid Corp., The, 1134-1135
Penn Electric Switch Co., 1141
Webster Engineering Co., The, 1180
White-Rodgers Electric Co., 1146
L. J. Wing Mfg. Co., 1000-1002

CONTROLLERS' AND CONTROL EQUIPMENT (*See also Humidity and Temperature Control*)

Alco Valve Co., 1118
American Moistening Co., 1021
Barber-Colman Co., 1084, 1121
Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
Friez Instrument Div., Bendix Aviation Corp., 1128
General Electric Co. (Schenectady, N. Y.), 1076-1077
Illinois Engineering Co., 1216-1217
Iron Fireman Mfg. Co., 1178-1179
Johnson Service Co., 1130-1131
Leeds & Northrup Co., 1133
Mercoid Corp., The, 1134-1135
Minneapolis-Honeywell Regulator Co., 1136-1137
Parks-Cramer Co., 951
Penn Electric Switch Co., 1141
Powers Regulator Co., The, 1142
Sarco Co., Inc., 1218-1219
Taylor Instrument Cos., 1144
Tenney Engineering, Inc., 1145
Webster Engineering Co., The, 1180
Westinghouse Electric Corp., 1078-1079
White-Rodgers Electric Co., 1146

CONVECTION HEATERS

American Radiator & Standard Sanitary Corp., 1148-1149
Bell & Gossett Co., 1196-1197
C. A. Dunham Co., 1206-1210
Electromode Corp., Div. American Foundry Equipment Co., 977
McQuay, Inc., 986-987
Modine Mfg. Co., 988-989
D. J. Murray Mfg. Co., 990
Rome-Turney Radiator Co., 1035-1036
Trane Co., The, 998-999
Tuttle & Bailey, Inc., 1092-1093
United States Radiator Corp., 1172-1173
Young Radiator Co., 997

COOLING EQUIPMENT, Air

Aerofin Corp., 1031-1033
Airtemp Div., Chrysler Corp., 966-967
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
Crarier Corp., 944-945
Curtis Refrigerating Machine Div. of Curtis Mfg. Co., 1040
DeBothezat Fans Div., American Machine & Metals, Inc., 1059
Farr Co., 1013
Fedders-Quigan Corp., 980
Frick Co. (Inc.), 1041
G & O Mfg. Co., The, 1035
General Electric Co. (Bloomfield, N. J.), 948-949
Kramer-Trenton Co., 985
La-Del Conveyor & Mfg. Co., 1062-1063
Marlo Coil Co., 1042
Modine Mfg. Co., 988-989
John J. Nesbitt, Inc., 994
Niagara Blower Co., 950
Refrigeration Economics, Inc., 995
Rome-Turney Radiator Co., 1036
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Tenney Engineering, Inc., 1145
Trane Co., The, 998-999
Utility Appliance Corp., 1073
Viking Air Conditioning Corp., The, 1074
Vilter Mfg. Co., The, 1048
L. J. Wing Mfg. Co., 1000-1002
Worthington Pump & Machinery Corp., 1046-1047

COOLING EQUIPMENT, Oil

Aerofin Corp., 1031-1033
Airtemp Div., Chrysler Corp., 966-967
Bell and Gossett Co., 1196-1197
Carrier Corp., 944-945
Condenser Service & Engineering Co., Inc., 1034
Fedders-Quigan Corp., 980
G & O Mfg. Co., The, 1035
General Electric Co. (Bloomfield, N. J.), 948-949
Marlo Coil Co., 1042
Modine Mfg. Co., 988-989
Refrigeration Economics Co., Inc., 995
Trane Co., The, 998-999
Universal Cooler Div., International Detrola Corp., 1045
Vilter Mfg. Co., The, 1048
Worthington Pump & Machinery Corp., 1046-1047

COOLING EQUIPMENT, Water (*See also Water Cooling*)

Acme Industries, Inc., 1030
Aerofin Corp., 1031-1033
Airtemp Div., Chrysler Corp., 966-967
April Showers Co., 1022
Bell & Gossett Co., 1196-1197
Carrier Corp., 944-945
Condenser Service & Engineering Co., Inc., 1034
Curtis Refrigerating Machine Div. of Curtis Mfg. Co., 1040

Fedders-Quigan Corp., 980
Frick Co. (Inc.), 1041
General Electric Co. (Bloomfield, N. J.), 948-949
Lilie-Hoffmann Cooling Towers, Inc., 1023
Marley Co., The, 1024
Marlo Coil Co., 1042
John J. Nesbitt, Inc., 994
Niagara Blower Co., 950
J. F. Pritchard & Co., 1025
Refrigeration Economics Co., Inc., 995
Rogers Diesel and Aircraft Corp. (Coey Floating Film Cooling Tower), 952, 1026
Tenney Engineering, Inc., 1145
Trane Co., The, 998-999
Vilter Mfg. Co., The, 1048
Water Cooling Equipment Corp., 1027
Worthington Pump & Machinery Corp., 1046-1047
Yarnall-Waring Co., 1232
York Corp., 960
Young Radiator Co., 997

COOLING TOWER FANS

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Hartzell Propeller Fan Co. (Div. of Castle Hills Corp.), 1064
Marlo Coil Co., 1042
J. F. Pritchard & Co., 1025
Propellair, Inc., 1068
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Water Cooling Equipment Corp., 1027

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Curtis Refrigerating Machine Div. of Curtis Mfg. Co., 1040
Lilie-Hoffman Cooling Towers, Inc., 1023
Marley Co., The, 1024
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D. J. Murray Mfg. Co., 990
J. F. Pritchard & Co., 1025
Rogers Diesel & Aircraft Corp., (Coey Floating Film Cooling Tower), 952, 1026
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Korfund Co., Inc., The, 1262
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Vinco Co., Inc., The, 1192-1193

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Philip Carey Mfg. Co., The, 1252-1253
Eagle-Picher Co., The, 1254
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H. W. Porter & Co., Inc., 1237
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Mercoid Corp., The, 1134-1135
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Penn Electric Switch Co., 1141
Trane Co., The, 998-999
United States Regulator Co., 1095
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American Radiator & Standard Sanitary Corp., 1148-1149
Automatic Products Co., 1120
Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
Minneapolis-Honeywell Regulator Co., 1136-1137
H. A. Thrush & Co., 1198-1199
Trane Co., 998-999
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Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
Fulton Sylphon Co., The, 1124-1125
Minneapolis-Honeywell Regulator Co., 1136-1137
Penn Electric Switch Co., 1126
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Anemostat Corp. of America, 1082
Barber-Colman Co., 1084, 1121
Belanger Fan & Blower Co., 1054-1055
W. B. Connor Engineering Corp., 1008-1009, 1085
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Webster Engineering Co., The, 1180
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United States Register Co., 1095
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Barber-Colman Co., 1084, 1121
Belanger Fan & Blower Co., 1054-1055
Johnson Service Co., 1130-1131
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John J. Nesbitt, Inc., 994
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J. F. Pritchard & Co., 1025
Rogers Diesel & Aircraft Corp. (Rogers-Research System), 952, 1026
H. J. Somers, Inc., 1016-1017
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
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Henry Valve Co., 1129
Mueller Brass Co., 1106-1107
J. F. Pritchard & Co., 1025
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Dow Chemical Co., The, 1114
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American District Steam Co., 1226, 1236
H. W. Porter & Co., Inc., 1237
Ric-wiL Co., The, 1238

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Aerofin Corp., 1031-1033
Buffalo Forge Co., 1056
Campbell Heating Co., 964-965
Carrier Corp., 944-945
G & O Mfg. Co., The, 1035
Modine Mfg. Co., 988-989
National Heater Co., 991
Niagara Blower Co., 950
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B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
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Worthington Pump & Machinery Corp., 1046-1047
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Air Devices, Inc., 1080-1081
Air-Maze Corp., The, 1004-1005
American Air Filter Co., Inc., 1006-1007
American Blower Corp., 940-941
Buffalo Forge Co., 1056
Chelsea Products, Inc., 1058
Dollinger Corp. (formerly Staynew Filter Corp.) 1010-1011
Farr Co., 1013
Ilg Electric Ventilating Co., 982, 1060-1061
Owens-Corning Fiberglas Corp., 1014-1015
J. F. Pritchard & Co., 1025
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019

DUST COLLECTORS, Cloth Type
American Air Filter Co., Inc., 1006-1007
Dollinger Corp. (formerly Staynew Filter Corp.), 1010-1011

EJECTORS, Sewage
Chicago Pump Co., 1204
Condenser Service & Engineering Co., Inc., 1034
Nash Engineering Co., 1202-1203

ELECTROSTATIC AIR CLEANERS
American Air Filter Co., Inc., 1006-1007
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019

ENGINES, Diesel
Worthington Pump & Machinery Corp., 1046-1047

ENGINES, Steam
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Vilter Mfg. Co., The, 1048

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EVAPORATORS
Acme Industries, Inc., 1030
Condenser Service & Engineering Co., Inc., 1034
Curtis Refrigerating Machine, Div. of Curtis Mfg. Co., 1040
Farrar & Trefts, Inc., 1158
Fedders-Quigan Corp., 980
General Electric Co. (Bloomfield, N. J.), 948-949
Kennard Corp., 984
Kramer-Trenton Co., 985
Marlo Coil Co., 1042
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Rome-Turney Radiator Co., 1036
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Trane Co., The, 998-999
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Worthington Pump & Machinery Corp., 1046-1047
York Corp., 960
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Air Devices, Inc., 1080-1081
American Blower Corp., 940-941
American Coolair Corp., 1050-1051
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
DeBothezat Fans Div., American Machine & Metals, Inc., 1059
Ilg Electric Ventilating Co., 982, 1060-1061
New York Blower Co., The, 1067
Schwitzer-Cummins Co., 1072
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Trane Co., The, 998-999
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E. B. Badger & Sons Co., 1110-1111
Chicago Metal Hose Corp., 1102
Crane Co., 1154-1155
Fulton Sylphon Co., The, 1124-1125
Arthur Harris & Co., 1115
Illinois Engineering Co., 1216-1217
Seamlex Co., Inc., 1103
Warren Webster & Co., 1222-1223
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Clarage Fan Co., 946
Hartzell Propeller Fan Co. (Div. of Castle Hills Corp.), 1064
J. F. Pritchard & Co., 1025
Schwitzer-Cummins Co., 1072
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Torrington Mfg. Co., The, 1070-1071
Utility Appliance Corp., 1073
Water Cooling Equipment Co., 1027
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Air Controls, Inc., 1049
American Coolair Corp., 1050-1051
Belanger Fan & Blower Co., 1054-1055

G. C. Breidert Co., 1083
Buffalo Forge Co., 1056
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
DeBothezat Fans Div., American Machine & Metals, Inc., 1059
Hartzell Propeller Fan Co. (Div. of Castle Hills Corp.), 1064
Ilg Electric Ventilating Co., 982, 1060-1061
La-Del Conveyor & Mfg. Co., 1062-1063
Lau Blower Co., The, 1065
Schwitzer-Cummins Co., 1072
H. J. Somers, Inc., 1016-1017
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Torrington Mfg. Co., The, 1070-1071
Viking Air Conditioning Corp., The, 1074
Water Cooling Equipment Co., 1027

FANS, Axial Flow
American Blower Corp., 940-941
Bahnson Co., The, 942-943
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
DeBothezat Fans Div., American Machine & Metals, Inc., 1059
Hartzell Propeller Fan Co. (Div. Castle Hills Corp.), 1064
Ilg Electric Ventilating Co., 982, 1060-1061
La-Del Conveyor & Mfg. Co., 1062-1063
Herman Nelson Corp., The, 992-993
J. F. Pritchard & Co., 1025
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Schwitzer-Cummins Co., 1072
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Viking Air Conditioning Corp., The, 1074
Water Cooling Equipment Co., 1027
L. J. Wing Mfg. Co., 1000-1002

FANS, Centrifugal
Air Controls, Inc., 1049
American Blower Corp., 940-941
American Coolair Corp., 1050-1051
Bayley Blower Co., 1052
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
Campbell Heating Co., 964-965
Carrier Corp., 944-945
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
Hastings Air Conditioning Co., Inc., 947
Ilg Electric Ventilating Co., 982, 1060-1061
Jaden Mfg. Co., 983
La-Del Conveyor & Mfg. Co., 1062-1063
Lau Blower Co., The, 1065
Morrison Products, Inc., 1066
New York Blower Co., The, 1067
Schwitzer-Cummins Co., 1072
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Torrington Mfg. Co., The, 1070-1071
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
Utility Appliance Corp., 1073
Viking Air Conditioning Corp., The, 1074
L. J. Wing Mfg. Co., 1000-1002

FANS, Electric
American Coolair Corp., 1050-1051
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
Ilg Electric Ventilating Co., 982, 1060-1061
La-Del Conveyor & Mfg. Co., 1062-1063
Herman Nelson Corp., The, 992-993
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Torrington Mfg. Co., 1070-1071

FANS, Furnace
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Buffalo Forge Co., 1056
Campbell Heating Co., 964-965
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
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Ilg Electric Ventilating Co., 982, 1060-1061
Jaden Mfg. Co., 983
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Utility Appliance Corp., 1073
Viking Air Conditioning Corp., The, 1074

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Belanger Fan & Blower Co., 1054-1055
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
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La-Del Conveyor & Mfg. Co., 1062-1063
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Propellair, Inc., 1068
Schwitzer-Cummins Co., 1072
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Torrington Mfg. Co., The, 1070-1071

FANS, Propeller
Air Controls, Inc., 1049
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American Coolair Corp., 1050-1051
Bahnson Co., The, 942-943
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
Champion Blower & Forge Co., 1057
Chelsea Products, Inc., 1058
Clarage Fan Co., 946
DeBothezat Fans Div., American Machine & Metals, Inc., 1059
Hartzell Propeller Fan Co. (Div. of Castle Hills Corp.), 1064
Ilg Electric Ventilating Co., 982, 1060-1061
Lau Blower Co., The, 1065
New York Blower Co., The, 1067
J. F. Pritchard & Co., 1025
Propellair, Inc., 1068
Schwitzer-Cummins Co., 1072
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Torrington Mfg. Co., The, 1070-1071
Trane Co., The, 998-999
Utility Appliance Corp., 1073
Viking Air Conditioning Corp., The, 1074
Water Cooling Equipment Co., 1027
L. J. Wing Mfg. Co., 1000-1002

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Propeller Fan Manufacturers Association, 1069

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American Coolair Corp., 1050-1051
Bayley Blower Co., 1052
Belanger Fan & Blower Co., 1054-1055
Buffalo Forge Co., 1056
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Hartzell Propeller Fan Co. (Div. of Castle Hills Corp.), 1064
Ilg Electric Ventilating Co., 982, 1060-1061
Jaden Mfg. Co., 983
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Penn Electric Switch Co., 1141
Sarco Co., Inc., 1218-1219
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FELT, Sound Deadening
Johns-Manville, 1258-1259
Lockport Cotton Batting Co., 1263
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Air-Maze Corp., 1004-1005
Air & Refrigeration Corp., 939
American Air Filter Co., Inc., 1006-1007
Badger Corp., 1012
Crane Co., 1154-1155
Dollinger Corp. (formerly Staynew Filter Corp.), 1010-1011
Farr Co., 1013
Owens-Corning Fiberglas Corp., 1014-1015
Research Products Corp., 1020
H. J. Somers, Inc., 1016-1017
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Babcock & Wilcox Co., The, 1150
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Taco Heaters, Inc., 1200
H. A. Thrush & Co., 1198-1199
Trane Co., The, 998-999

FITTINGS, Pipe, Flanged
Baker Ice Machine Co., Inc., 1038
Crane Co., 1154-1155
Grinnell Co., Inc., 981, 1112-1113, 1211
Henry Valve Co., 1129
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Baker Ice Machine Co., 1038
Crane Co., 1154-1155
Grinnell Co., Inc., 981, 1112-1113, 1211
Henry Valve Co., 1129
Mueller Brass Co., 1106-1107
Vilter Mfg. Co., The, 1048
Worthington Pump & Machinery Corp., 1046-1047
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FITTINGS, Pipe, Solder
Crane Co., 1154-1155
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FITTINGS, Pipe for Underground Conduit
American District Steam Co., 1226, 1236
H. W. Porter & Co., Inc., 1237
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FITTINGS, Welding
Crane Co., 1154-1155
Grinnell Co., Inc., 981, 1112-1113, 1211
Henry Valve Co., 1129
Tube Turns, Inc., 1117

FLANGES, Lead, Roof
Eagle-Picher Co., The, 1254

FLOATS, Ferrous and Nonferrous (*Seamless*)
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FLOATS, Metal (*See also Trap and Valve*)

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Grinnell Co., Inc., 981, 1112-1113, 1211
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FLUE GAS ANALYSIS
Leeds & Northrup Co., 1133
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Trane Co., The, 998-999
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Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
Payne Furnace Co., 972-973
United States Register Co., 1095

FURNACES, Electric
Leeds & Northrup Co., 1133

FURNACES, Oil Burning, Floor
Air Devices, Inc., 1080-1081
Automatic Burner Corp., 1181
H. C. Little Burner Co., 1188

FURNACES, Warm Air, Heavy Duty
American Foundry & Furnace Co., 962-963
Airtherm Mfg. Co., 976
Campbell Heating Co., 964-965
Dravo Corp., 978-979
Iron Fireman Mfg. Co., 1178-1179
Lee Engineering Co., 968
Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
National Heater Co., 991
Payne Furnace Co., 972-973

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American Foundry & Furnace Co., 962-963
American Radiator & Standard Sanitary Corp., 1148-1149
Bryant Heater Co., The, 961
Campbell Heating Co., 964-965
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Charles Demuth & Sons, 1086
Fitzgibbons Boiler Co., Inc., 1156-1157
Gar Wood Industries, Inc., 974-975
General Electric Co. (Bloomfield, N. J.), 948-949

Iron Fireman Mfg. Co., 1178-1179
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Ray Oil Burner Co., 1189
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Crane Co., 1154-1155
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GAGES, Ammonia
Crane Co., 1154-1155
Jas. P. Marsh Corp., 1028-1029
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GAGES, Compound
Crane Co., 1154-1155
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Meriam Instrument Co., The, 1138
Minneapolis-Honeywell Regulator Co., 1136-1137
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American Meter Co., Inc., 1119
Crane Co., 1154-1155
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GAGES, Tank
Meriam Instrument Co., The, 1138
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GAGES, Vacuum
Crane Co., 1154-1155
Meriam Instrument Co., The, 1138
Minneapolis-Honeywell Regulator Co., 1136-1137
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GAGES, Vapor
Crane Co., 1154-1155
Minneapolis-Honeywell Regulator Co., 1136-1137

GAGES, Water
Crane Co., 1154-1155
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Crane Co., 1154-1155
Ilg Electric Ventilating Co., 982, 1060-1061
Ray Oil Burner Co., 1189
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GASKETS, Asbestos
Crane Co., 1154-1155
Johns-Manville, 1258-1259

GASKETS, Cork
Armstrong Cork Co., 1248
Mundet Cork Corp., 1264

GLASS (*See Insulation, Double Glass*)

GLASS BLOCK ROOFLIGHTS (*See Skylights*)

GLASS BLOCKS
American 3 Way-Luxfer Prism Co., 1239
Owens-Illinois Glass Co., 1243
Pittsburgh Corning Corp., 1244-1245

GLASS, Cellular
Owens-Illinois Glass Co., 1243
Pittsburgh Corning Corp., 1244-1245

GOVERNORS, Pump
McDonnell & Miller, 1194-1195
Mueller Steam Specialty Co., Inc., 1227
Spence Engineering Co., Inc., 1143

GRATES FOR BOILERS AND FURNACES
Combustion Engineering Co., 1176
Detroit Stoker Co., 1177
Fitzgibbons Boiler Co., Inc., 1156-1157

GRILLES, REGISTERS AND ORNAMENTAL METAL WORK (*See also Louvers and Registers*)
Air Devices, Inc., 1080-1081
American Foundry & Furnace Co., 962-963
Anemostat Corp. of America, 1082
Barber-Colman Co., 1084, 1121
Hart & Cooley Mfg. Co., 1088-1089
Hendrick Mfg. Co., 1087
Independent Register Co., The, 1090
L. J. Mueller Furnace Co., 970-971
Pyle-National Co., The, 1091
Register & Grille Mfg. Co., Inc., 1094
Tuttle & Bailey, Inc., 1092-1093
United States Register Co., 1095
Vulcan Radiator Co., 1037
Young Regulator Co., 1096

HANGERS, Pipe
Grinnell Co., Inc., 981, 1112-1113, 1211
Ric-wiL Co., The, 1238

HANGERS, Radiator
Bell & Gossett Co., 1196-1197
United States Radiator Corp., 1172-1173
Vulcan Radiator Co., 1037

HEADS, Exhaust
G. C. Breidert Co., 1083
Crane Co., 1154-1155
Wright-Austin Co., 1231

HEADS, Sprinkler (*Fire Protection*)
Grinnell Co., Inc., 981, 1112-1113, 1211

HEAT SURFACE
Aerofin Corp., 1031-1033
G & O Mfg. Co., The, 1035
Kramer Trenton Co., 985
McQuay, Inc., 986-987
Modine Mfg. Co., 988-989
John J. Nesbitt, Inc., 994
New York Blower Co., The, 1067
Refrigeration Economics Co., 995
Rome-Turney Radiator Co., 1036
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Trane Co., The, 998-999
Young Radiator Co., 997

HEATERS, Air
Aerofin Corp., 1031-1033
Airtherm Mfg. Co., 976
Buffalo Forge Co., 1056
Campbell Heating Co., 964-965
Carrier Corp., 944-945
Combustion Engineering Co., 1176
Dravo Corp., 978-979
Electromode Corp. (Div. American Foundry Equipment Co.), 977
Fedders-Quigan Corp., 980
Grinnell Co., Inc., 981, 1112-1113, 1211
Ilg Electric Ventilating Co., 982, 1060-1061
Lee Engineering Co., 968
H. C. Little Burner Co., 1188
McQuay, Inc., 986-987
Marlo Coil Co., 1042
Meyer Furnace Co., The, 969
Modine Mfg. Co., 988-989
National Heater Co., 991

Herman Nelson Corp., The, 992-993
John J. Nesbitt, Inc., 994
Reznor Mfg. Co., 996
Rome-Turney Radiator Co., 1036
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Trane Co., The, 998-999
Utility Appliance Corp., 1073
Young Radiator Co., 997

HEATERS, Automatic Hot Water, Domestic

Air Devices Inc., 1080-1081
Airtemp Div., Chrysler Corp., 966-967
American Radiator & Standard Sanitary Corp., 1148-1149
Automatic Burner Corp., 1181
Bryant Heater Co., The, 961
Crane Co., 1154-1155
Gar Wood Industries, Inc., 974-975
General Electric Co. (Schenectady, N. Y.), 1076-1077
S. T. Johnson Co., 1184-1185
H. C. Little Burner Co., 1188
Petroleum Heat & Power Corp., 1186-1187
Ray Oil Burner Co., 1189
York-Shipley, Inc., 1191
Young Radiator Co., 997

HEATERS, Blast

Aerofin Corp., 1031-1033
Buffalo Forge Co., 1056
Carrier Corp., 944-945
Electromode Corp. (Div. American Foundry Equipment Co), 977
G & O Mfg. Co., The, 1035
Kennard Corp., 984
Kramer Trenton Co., 985
Lee Engineering Co., 968
Marlo Coil Co., 1042
McQuay, Inc., 986-987
Modine Mfg. Co., 988-989
John J. Nesbitt, Inc., 994
Refrigeration Economics Co., Inc., 995
Rome-Turney Radiator Co., 1036
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Trane Co. The, 998-999
Vulcan Radiator Co., 1037
Young Radiator Co., 997

HEATERS, Cabinet

C. A. Dunham Co., 1206-1210
Modine Mfg. Co., 988-989
Herman Nelson Corp., The, 992-993
John J. Nesbitt, Inc., 994
Payne Furnace Co., 972-973

HEATERS, Electric

Crane Co., 1154-1155
Electromode Corp. (Div. American Foundry Equipment Co), 977
General Electric Co. (Schenectady, N. Y.), 1076-1077
Ilg Electric Ventilating Co., 982, 1060-1061

HEATERS, Feed Water

Bell & Gossett Co., 1196-1197
Brownell Co., The, 1175
Condenser Service & Engineering Co., Inc., 1034
National Radiator Co., The, 1164-1165
Worthington Pump & Machinery Corp., 1046-1047

HEATERS, Fuel Oil

American District Steam Co., 1226, 1236
Automatic Burner Corp., 1181
Bell & Gossett Co., 1196-1197
Campbell Heating Co., 964-965
Condenser Service & Engineering Co., Inc., 1034
Dravo Corp., 978-979
National Heater Co., 991
National Radiator Co., The, 1164-1165
Combustion Equipment Div., (Todd Shipyards Corp.), 1182
Petroleum Heat & Power Corp., 1186-1187
York-Shipley, Inc., 1191

HEATERS, Gas

Bryant Heater Co., The 961
Campbell Heating Co., 964-965
Crane Co., 1154-1155
Dravo Corp., 978-979
Ilg Electric Ventilating Co., 982, 1060-1061
Lee Engineering Co., 968
Meyer Furnace Co., The, 969
National Heater Co., 991
Payne Furnace Co., 972-973
Reznor Mfg. Co., 996
Utility Appliance Corp., 1073

HEATERS, Hot Water Service

Air Devices Inc., 1080-1081
American District Steam Co., 1226, 1236
American Radiator & Standard Sanitary Corp., 1148-1149
Bell and Gossett Co., 1196-1197
Brownell Co., The, 1175
Burnham Boiler Corp., 1152
Crane Co., 1154-1155
Fitzgibbons Boiler Co., Inc., 1156-1157
Kewanee Boiler Corp., 1160-1163
H. C. Little Burner Co., 1188
L. J. Mueller Furnace Co., 970-971
H. B. Smith Co., Inc., The, 1167
Swan Engineering Co., 1116
Taco Heaters, Inc., 1200
Terre Haute Boiler Works Co., 1170
Trane Co., The, 998-999
United States Radiator Corp., 1172-1173
Weil-McLain Co., 1174
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190

HEATERS, Indirect

American Radiator & Standard Sanitary Corp., 1148-1149
Bell & Gossett Co., 1196-1197
Crane Co., 1154-1155
John J. Nesbitt, Inc., 994
Taco Heaters, Inc., 1200
H. A. Thrush & Co., 1198-1199

HEATERS, Storage

American District Steam Co., 1226, 1236
Bell and Gossett Co., 1196-1197
Brownell Co., The, 1175
Kewanee Boiler Corp., 1160-1163
Terre Haute Boiler Works Co., 1170

HEATERS, Tank

American Radiator & Standard Sanitary Corp., 1148-1149
Bell and Gossett Co., 1196-1197
Bryant Heater Co., The, 961
Burnham Boiler Corp., 1152
Fitzgibbons Boiler Co., Inc., 1156-1157
H. B. Smith Co., Inc., The, 1167
Taco Heaters, Inc., 1200
Terre Haute Boiler Works Co., 1170
Weil-McLain Co., 1174

HEATERS, Unit

Airtherm Mfg. Co., 976
American Blower Corp., 940-941
American Foundry & Furnace Co., 962-963
Bayley Blower Co., 1052
Bryant Heater Co., The, 961
Buffalo Forge Co., 1056
Burnham Boiler Corp., 1152
Carrier Corp., 944-945
Clarage Fan Co., 946
Dravo Corp., 978-979
C. A. Dunham Co., 1206-1210
Electromode Corp. (Div. American Foundry Equipment Co.), 977
Fedders-Quigan Corp., 980
Grinnell Co., Inc., 981, 1112-1113, 1211
Hartzell Propeller Fan Co. (Div. of Castle Hills Corp.), 1064
Hastings Air Conditioning Co., Inc., 947
Ilg Electric Ventilating Co., 982, 1060-1061
Jaden Mfg. Co., 983
Kennard Corp., 984
Kramer Trenton Co., 985
Lee Engineering Co., 968
McQuay, Inc., 986-987
Modine Mfg. Co., 988-989
D. J. Murray Mfg. Co., 990
National Heater Co., 991
National Radiator Co., The, 1164-1165
Herman Nelson Corp., The, 992-993
John J. Nesbitt, Inc., 994
New York Blower Co., The, 1067
Niagara Blower Co., 950
Refrigeration Economics Co., Inc., 995
Reznor Mfg. Co., 996
B. F. Sturtevant Co., Div., Westinghouse Electric 954-959, 1018-1019
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
Warren Webster & Co., 1222-1223
L. J. Wing Mfg. Co., 1000-1002
Young Radiator Co., 997

HEATERS, Unit, Gas Fired

Airtherm Mfg. Co., 976
American Foundry & Furnace Co., 962-963
American Radiator & Standard Sanitary Corp., 1148-1149
Bryant Heater Co., The, 961
Buffalo Forge Co., 1056
Dravo Corp., 978-979
Ilg Electric Ventilating Co., 982, 1060-1061
Lee Engineering Co., 968
McQuay, Inc., 986-987
L. J. Mueller Furnace Co., 970-971
National Radiator Co., The, 1164-1165
Reznor Mfg. Co., 996
Utility Appliance Corp., 1073

HEATING SYSTEMS, Air, Heavy Duty

Aerofin Corp., 1031-1033
Airtherm Mfg. Co., 976
American Blower Corp., 940-941
American Foundry & Furnace Co., 962-963
Campbell Heating Co., 964-965
Carrier Corp., 944-945
Dravo Corp., 978-979
Iron Fireman Mfg. Co., 1178-1179
Lee Engineering Co., 968
Meyer Furnace Co., The, 969
National Heater Co., 991
Payne Furnace Co., 972-973
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Trane Co., 998-999

HEATING SYSTEMS, Air, Residence

Aerofin Corp., 1031-1033
Airtemp Div., Chrysler Corp., 966-967
American Foundry & Furnace Co., 962-963
American Radiator & Standard Sanitary Corp., 1148-1149
Bahnson Co., The, 942-943
Bryant Heater Co., The, 961
Buffalo Forge Co., 1056
Burnham Boiler Corp., 1152
Campbell Heating Co., 964-965
Carrier Corp., 944-945
Charles Demuth & Sons, 1086
Electromode Corp. (Div. American Foundry Equipment Co.), 977
Fedders-Quigan Corp., 980
Gar Wood Industries, Inc., 974-975
General Electric Co., (Bloomfield, N. J.), 948-949
Iron Fireman Mfg. Co., 1178-1179
H. C. Little Burner Co., 1188
Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
Payne Furnace Co., 972-973
Ray Oil Burner Co., 1189
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190

HEATING SYSTEMS, Automatic

Airtemp Div., Chrysler Corp., 966-967
American Foundry & Furnace Co., 962-963
American Radiator & Standard Sanitary Corp., 1148-1149
Anderson Products, Inc., 1233
Barnes & Jones, Inc., 1205
Bell & Gossett Co., 1196-1197
Bryant Heater Co., The, 961
Burnham Boiler Corp., 1152
Campbell Heating Co., 964-965
Crane Co., 1154-1155
Dravo Corp., 978-979
Gar Wood Industries, Inc., 974-975
General Electric Co. (Bloomfield, N. J.), 948-949
Illinois Engineering Co., 1216-1217
Iron Fireman Mfg. Co., 1178-1179
Lee Engineering Co., 968
H. C. Little Burner Co., 1188
Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
National Heater Co., 991
National Radiator Co., The, 1164-1165
Herman Nelson Corp., The, 992-993
Payne Furnace Co., 972-973
Ray Oil Burner Co., 1189
Sarco Co., Inc., 1218-1219
Spence Engineering Co., Inc., 1143
Spencer Heater Div., The Aviation Corp., 1168-1169
Taco Heaters, Inc., 1200
Trane Co., The, 998-999
United States Radiator Corp., 1172-1173
Warren Webster Co., 1222-1223
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190
L. J. Wing Mfg. Co., 1000-1002

HEATING SYSTEMS, Coal-fired

Airtemp Div., Chrysler Corp., 966-967
American Foundry & Furnace Co., 962-963
Burnham Boiler Corp., 1152
Campbell Heating Co., 964-965
Dravo Corp., 978-979
Iron Fireman Mfg. Co., 1178-1179
Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
National Heater Co., 991
National Radiator Co., The, 1164-1165

HEATING SYSTEMS, Electric

Electromode Corp. (Div. American Foundry Equipment Co.), 977

HEATING SYSTEMS, Furnace

Airtemp Div., Chrysler Corp., 966-967
Airtherm Mfg. Co., 976
Campbell Heating Co., 964-965
Crane Co., 1154-1155
Dravo Corp., 978-979
Lee Engineering Co., 968
Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
National Heater Co., 991

HEATING SYSTEMS, Gas Fired

Airtemp Div., Chrysler Corp., 966-967
American Foundry & Furnace Co., 962-963
Bryant Heater Co., The, 961
Burnham Boiler Corp., 1152
Campbell Heating Co., 964-965
Crane Co., 1154-1155
Dravo Corp., 978-979
Gar Wood Industries, Inc., 974-975
General Electric Co. (Bloomfield, N. J.), 948-949
Lee Engineering Co., 968
Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
National Heater Co., 991
National Radiator Co., The, 1164-1165
Payne Furnace Co., 972-973

HEATING SYSTEMS, Hot Water

Airtemp Div., Chrysler Corp., 966-967
Bell & Gossett Co., 1196-1197
Burnham Boiler Corp., 1152
Gar Wood Industries, Inc., 974-975
Hoffman Specialty Co., Inc., 1213-1215
Iron Fireman Mfg. Co., 1178-1179
L. J. Mueller Furnace Co., 970-971
National Radiator Co., The, 1164-1165
Ric-wiL Co., The, 1238
Sarco Co., Inc., 1218-1219
Taco Heaters, Inc., 1200
Trane Co., The, 998-999
H. A. Thrush & Co., 1198-1199
Warren Webster & Co., 1222-1223
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190

HEATING SYSTEMS, Oil Fired

Air Devices, Inc., 1080-1081
Airtemp Div., Chrysler Corp., 966-967
American Foundry & Furnace Co., 962-963
Burnham Boiler Corp., 1152
Campbell Heating Co., 964-965
Charles Demuth & Sons, 1086
Dravo Corp., 978-979
Gar Wood Industries, Inc., 974-975
General Electric Co. (Bloomfield, N. J.), 948-949
Iron Fireman Mfg. Co., 1178-1179
S. T. Johnson Co., 1184-1185
Kewanee Boiler Corp., 1160-1163
Lee Engineering Co., 968
H. C. Little Burner Co., 1188
Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
National Heater Co., 991
National Radiator Co., The, 1164-1165
Petroleum Heat & Power Corp., 1186-1187
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190
York-Shipley, Inc., 1191

HEATING SYSTEMS, Steam

Aerofin Corp., 1031-1033
Airtemp Div., Chrysler Corp., 966-967
Anderson Products, Inc., 1233
Barnes & Jones, Inc., 1205
Bryant Heater Co., The, 961
Burnham Boiler Corp., 1152
C. A. Dunham Co., 1206-1210
Gar Wood Industries, Inc., 974-975
William S. Haines & Co., 1212
Hoffman Specialty Co., Inc., 1213-1215
Illinois Engineering Co., 1216-1217
Iron Fireman Mfg. Co., 1178-1179
L. J. Mueller Furnace Co., 970-971
National Radiator Co., The, 1164-1165
Ric-wiL Co., The, 1238
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1222-1223

HEATING SYSTEMS, Vacuum

Aerofin Corp., 1031-1033
Airtemp Div., Chrysler Corp., 966-967
American Radiator & Standard Sanitary Corp., 1148-1149
Anderson Products, Inc., 1233
Barnes & Jones, Inc., 1205
Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
William S. Haines & Co., 1212
Hoffman Specialty Co., Inc., 1213-1215
Illinois Engineering Co., 1216-1217
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1222-1223

HEATING SYSTEMS, Vapor

Aerofin Corp., 1031-1033
Airtemp Div., Chrysler Corp., 966-967
Barnes & Jones, Inc., 1205
Bryant Heater Co., The, 961
C. A. Dunham Co., 1206-1210
Gar Wood Industries, Inc., 974-975
William S. Haines & Co., 1212
Hoffman Specialty Co., Inc., 1213-1215
Illinois Engineering Co., 1216-1217
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223
Williams Oil-O-Matic Div., Eureka Williams Corp., 1190

HOSE, Flexible Metal

Atlantic Metal Hose Co., Inc., 1101
Chicago Metal Hose Corp., 1102
Seamlex Co., Inc., 1103

HOSE, Liquid, Gas, Vapor

Atlantic Metal Hose Co., Inc., 1101
Chicago Metal Hose Corp., 1102

HOT WATER HEATING SYSTEMS (*See Heating Systems, Hot Water*)

HUMIDIFIERS

Air-Maze Corp., The, 1004-1005
Air & Refrigeration Corp., 939
American Blower Corp., 940-941
American Moistening Co., 1021
American Radiator & Standard Sanitary Corp., 1148-1149
Armstrong Machine Works, 1224-1225
Bahnson Co., The, 942-943
Baker Ice Machine Co., Inc., 1038
Barber-Colman Co., 1084, 1121
Buffalo Forge Co., 1056
Burnham Boiler Corp., 1152
Carrier Corp., 944-945
Clarage Fan Co., 946
Farr Co., 1013
Grinnell Co., Inc., 981, 1112-1113, 1211
Johnson Service Co., 1130-1131
Marlo Coil Co., 1042
McDonnell & Miller, 1194-1195
McQuay, Inc., 986-987
Meyer Furnace Co., The, 969
L. J. Mueller Furnace Co., 970-971
D. J. Murray Mfg. Co., 990
Niagara Blower Co., 950
Parks-Cramer Co., 951
J. F. Pritchard & Co., 1025
H. J. Somers Inc., 1016-1017
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Tenney Engineering, Inc., 1145
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
L. J. Wing Mfg. Co., 1000-1002
York Corp., 960

HUMIDIFIERS, Central Plant

Air & Refrigeration Corp., 939
American Blower Corp., 940-941
Bahnson Co., The, 942-943
Barber-Colman Co., 1084, 1121
Buffalo Forge Co., 1056
Carrier Corp., 944-945
Farr Co., 1013
Johnson Service Co., 1130-1131
McDonnell & Miller, 1194-1195
Niagara Blower Co., The, 950
Parks-Cramer Co., 951
J. F. Pritchard & Co., 1025
H. J. Somers, Inc., 1016-1017
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Viller Mfg. Co., 1048
York Corp., 960

HUMIDIFIERS, Unit

American Moistening Co., 1021
Armstrong Machine Works, 1224-1225
Badger Corp., 1012
Bahnson Co., The, 942-943
Buffalo Forge Co., 1056
Carrier Corp., 944-945
Chelsea Products, Inc., 1058
Marley Co., The, 1024
D. J. Murray Mfg. Co., 990
Niagara Blower Co., 950
Parks-Cramer Co., 951
J. F. Pritchard & Co., 1025
H. J. Somers, Inc., 1016-1017
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Trane Co., The, 998-999
United States Air Conditioning Corp., 953
Viking Air Conditioning Corp., 1074
York Corp., 960

HUMIDITY CONTROL

American Moistening Co., 1021
Bahnson Co., The, 942-943
Barber-Colman Co., 1084, 1121
Friez Instrument Div., Bendix Aviation Corp., 1128
Johnson Service Co., 1130-1131
Leeds & Northrup Co., 1133
Minneapolis-Honeywell Regulator Co., 1136-1137
Parks-Cramer Co., 951
Penn Electric Switch Co., 1141
Powers Regulator Co., 1142
H. J. Somers, Inc., 1016-1017
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Taylor Instrument Cos., 1144
Tenney Engineering, Inc., 1145
White-Rodgers Electric Co., 1146

HUMIDITY RECORDERS and INDICATORS

American Moistening Co., 1021
Friez Instrument Div., Bendix Aviation Corp., 1128
Illinois Testing Laboratories, Inc., 1132
Johnson Service Co., 1130-1131
Leeds & Northrup Co., 1133
Minneapolis-Honeywell Regulator Co., 1136-1137
Moeller Instrument Co., 1139
Powers Regulator Co., 1142
Taylor Instrument Cos., 1144

HYGROMETERS (*See also Humidity Recorders and Indicators*)

American Moistening Co., 1021
Illinois Testing Laboratories, Inc., 1132
Johnson Service Co., 1130-1131
Moeller Instrument Co., 1139
Palmer Co., The, 1140
Parks-Cramer Co., 951
Taylor Instrument Cos., 1144

INDUCED DRAFT COOLING TOWERS (*See also Cooling Towers, Forced Draft, Mechanical Draft*)

Baker Ice Machine Co., Inc., 1038
Buffalo Forge Co., 1056
Lilie-Hoffman Cooling Towers, Inc., 1263
Marley Co., The, 1024
D. J. Murray Mfg Co., 990
J. F. Pritchard & Co., 1025
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
Water Cooling Equipment Co., 1027
Young Radiator Co., 997

INSTRUMENTS, Indicating, Controlling and Recording

American Meter Co., Inc., 1119
Friez Instrument Div., Bendix Aviation Corp., 1128
General Electric Co. (Schenectady, N. Y.), 1076-1077
Illinois Testing Laboratories, Inc., 1132
Leeds & Northrup Co., 1133
Minneapolis-Honeywell Regulator Co., 1136-1137
Moeller Instrument Co., 1139
Palmer Co., The, 1140
Powers Regulator Co., 1142
Taylor Instrument Cos., 1144

INSULATION, Building

Alfol Insulation Co., Inc., 1246
American Flange & Mfg. Co., Inc., 1247
April Showers Co., 1022
Armstrong Cork Co., 1248
Baldwin-Hill Co., 1249
Philip Carey Mfg. Co., The, 1252-1253
Celotex Corp., The, 1250
Eagle-Picher Co., The, 1254
Insulite, 1256-1257
Insul-Wool Insulation Corp., 1255
Johns-Manville, 1258-1259
Kimberly-Clark Corp., 1260-1261
Lockport Cotton Batting Co., 1263
Mundet Cork Corp., 1264
Owens-Corning Fiberglas Corp., 1014-1015
Pacific Lumber Co., The, 1265
United States Gypsum Co., 1270-1271
Wood Conversion Co., 1272

INSULATION, Cellular Glass

Armstrong Cork Co., 1248
Owens-Illinois Glass Co., Insulux Products Div., 1243
Pittsburgh Corning Corp., 1244-1245

INSULATION, Cork

Armstrong Cork Co., 1248
Cork Insulation Co., Inc., 1251
Korfund Co., Inc., The, 1262
Mundet Cork Corp., 1264
H. W. Porter & Co., 1237

INSULATION, Cotton

Lockport Cotton Batting Co., 1263
Reynolds Metals Co., 1266-1267

INSULATION, Double Glass

American 3 Way-Luxfer Prism Co., 1239
Libby-Owens-Ford Glass Co., 1242
Owens-Illinois Glass Co., Insulux Products Div., 1243
Pittsburgh Corning Corp., 1244-1245

INSULATION, Ducts, Ventilating, Air Conditioning

Alfol Insulation Co., 1246
Armstrong Cork Co., 1248
Baldwin-Hill Co., 1249
Philip Carey Mfg. Co., The, 1252-1253
Celotex Corp., The, 1250
Cork Insulation Co., Inc., 1251
Eagle-Picher Co., The, 1254
Insulite, 1256-1257
Johns-Manville, 1258-1259
Mundet Cork Corp., 1264
Owens-Corning Fiberglas Corp., 1014-1015
Pittsburgh Corning Corp., 1244-1245
Products Research Co., 1230
Ruberoid Co., The, 1268-1269
United States Register Co., 1095

INSULATION, Felt

Baldwin-Hill Co., 1249
Johns-Manville, 1258-1259
Kimberly-Clark Corp., 1260-1261
Lockport Cotton Batting Co., 1263
Products Research Co., 1230
Ruberoid Co., The, 1268-1269
Wood Conversion Co., 1272

INSULATION, Magnesia

Philip Carey Mfg. Co., The, 1252-1253
Johns-Manville, 1258-1259
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Babcock & Wilcox Co., The, 1150
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Gar Wood Industries, Inc., 974-975
Iron Fireman Mfg. Co., 1178-1179
S. T. Johnson Co., 1184-1185
L. J. Mueller Furnace Co., 970-971
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Enterprise Engine & Foundry Co., 1183
S. T. Johnson Co., 1184-1185
Preferred Utilities Mfg. Corp., 1166
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Chicago Pump Co., 1204
Condenser Service & Engineering Co., Inc., 1034
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Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
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Hart & Cooley Mfg. Co., 1088-1089
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Dollinger Corp. (formerly Staynew Filter Corp.), 1010-1011
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Worthington Pump & Machinery Corp., 1046-1047
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Warren Webster & Co., 1220-1223
Worthington Pump & Machinery Corp., 1046-1047
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American Rolling Mill Co., The, 1097
Bethlehem Steel Co., 1098
Carnegie-Illinois Steel Corp., 1100
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SHEETS, High Tensile
American Rolling Mill Co., The, 1097
Bethlehem Steel Co., 1098
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SHEETS, Steel
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Bethlehem Steel Co., 1098
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Babcock & Wilcox Co., The, 1150
Brownell Co., The, 1175
Combustion Engineering Co., 1176
Crane Co., 1154-1155
Detroit Stoker Co., 1177
Iron Fireman Mfg. Co., 1178-1179
Meyer Furnace Co., The, 969

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C. A. Dunham Co., 1206-1210
Grinnell Co., Inc., 981, 1112-1113, 1211
Hoffman Specialty Co., Inc., 1213-1215
Illinois Engineering Co., 1216-1217
Mueller Steam Specialty Co., Inc., 1227
Sarco Co., 1218-1219
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Automatic Products Co., 1120
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Henry Valve Co., 1129
Mueller Brass Co., 1106-1107
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Illinois Engineering Co., 1216-1217
Sarco Co., Inc., 1218-1219
Spence Engineering Co., Inc., 1143
Trane Co., The, 998-999
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Crane Co., 1154-1155
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Grinnell Co., Inc., 981, 1112-1113, 1211
Illinois Engineering Co., 1216-1217
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Mueller Steam Specialty Co., Inc., 1227
Sarco Co., 1218-1219
Spence Engineering Co., Inc., 1143
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Brownell Co., The, 1175
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Kewanee Boiler Corp., 1160-1163
Taco Heaters, Inc., 1200
H. A. Thrush & Co., 1198-1199

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Farrar & Trefts, Inc., 1158
Kewanee Boiler Corp., 1160-1163
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American Radiator & Standard Sanitary Corp., 1148-1149
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Barber-Colman Co., 1084, 1121
Crane Co., 1154-1155
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Illinois Engineering Co., 1216-1217
Iron Fireman Mfg. Co., 1178-1179
Johnson Service Co., 1130-1131
Leeds & Northrup Co., 1133
Mercoid Corp., 1134-1135
Minneapolis-Honeywell Regulator Co., 1136-1137
Penn Electric Switch Co., 1141
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Sarco Co., Inc., 1218-1219
Spence Engineering Co., Inc., 1143
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Powers Regulator Co., 1142
Sarco Co., Inc., 1218-1219
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Illinois Testing Laboratories, Inc., 1132
Johnson Service Co., 1130-1131
Leeds & Northrup Co., 1133
Minneapolis-Honeywell Regulator Co., 1136-1137
Moeller Instrument Co., 1139
Palmer Co., The, 1140
Powers Regulator Co., 1142
Taylor Instrument Co., 1144

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Friez Instrument Div., Bendix Aviation Corp., 1128
Leeds & Northrup Co., 1133
Jas. P. Marsh Corp., 1228-1229
Minneapolis-Honeywell Regulator Co., 1136-1137
Moeller Instrument Co., 1139
Palmer Co., The, 1140
Powers Regulator Co., 1142
Taylor Instrument Cos., 1144

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Barber-Colman Co., 1084, 1121
Crane Co., 1154-1155
Friez Instrument Div., Bendix Aviation Corp., 1128
Fulton Sylphon Co., 1124-1125
General Controls, 1126-1127
Iron Fireman Mfg. Co., 1178-1179
Johnson Service Co., 1130-1131
Mercoid Corp., 1134-1135
Minneapolis-Honeywell Regulator Co., 1136-1137
Penn Electric Switch Co., 1141
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Sarco Co., Inc., 1218-1219
H. A. Thrush & Co., 1198-1199
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Mercoid Corp., 1134-1135
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
H. A. Thrush & Co., 1198-1199
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TRAPS, Bucket
Armstrong Machine Works, 1224-1225
Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
Mueller Steam Specialty Co., Inc., 1227
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223
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American District Steam Co., 1226, 1236
Armstrong Machine Works, 1224-1225
Barnes & Jones, Inc., 1205
Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
Grinnell Co., Inc., 981, 1112-1113, 1211
William S. Haines & Co., 1212
Illinois Engineering Co., 1216-1217
Mueller Steam Specialty Co., Inc., 1227
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223
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American District Steam Co., 1226, 1236
Barnes & Jones, Inc., 1205

C. A. Dunham Co., 1206-1210
Grinnell Co., Inc., 981, 1112-1113, 1211
William S. Haines & Co., 1212
Hoffman Specialty Co., Inc., 1213-1215
Illinois Engineering Co., 1216-1217
Jas. P. Marsh Corp., 1228-1229
Powers Regulator Co., 1142
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223
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TRAPS, Radiator

Barnes & Jones, Inc., 1205
C. A. Dunham Co., 1206-1210
Grinnell Co., Inc., 981, 1112-1113, 1211
William S. Haines & Co., 1212
Hoffman Specialty Co., Inc., 1213-1215
Illinois Engineering Co., 1216-1217
Jas. P. Marsh Corp., 1228-1229
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223

TRAPS, Return

Barnes & Jones, Inc., 1205
Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
William S. Haines & Co., 1212
Illinois Engineering Co., 1216-1217
Jas. P. Marsh Corp., 1228-1229
Mueller Steam Specialty Co., Inc., 1227
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223
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TRAPS, Scale

Henry Valve Co., 1129
Illinois Engineering Co., 1216-1217
Sarco Co., Inc., 1218-1219

TRAPS, Steam

Armstrong Machine Works, 1224-1225
Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
William S. Haines & Co., 1212
Illinois Engineering Co., 1216-1217
Jas. P. Marsh Corp., 1228-1229
Mueller Steam Specialty Co., Inc., 1227
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223
Wright-Austin Co., 1231
Yarnall-Waring Co., 1232

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American District Steam Co., 1226, 1236
Barnes & Jones, Inc., 1205
Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
Grinnell Co., Inc., 981, 1112-1113, 1211
William S. Haines & Co., 1212
Illinois Engineering Co., 1216-1217
Jas. P. Marsh Corp., 1228-1229
Powers Regulator Co., The, 1142
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223

TRAPS, Vacuum

Armstrong Machine Works, 1224-1225
William S. Haines & Co., 1212
Illinois Engineering Co., 1216-1217

TUBE CLEANERS

Martocello, Jos. A. & Co., 1028

TUBES, Boiler

Babcock & Wilcox Tube Co., The, 1150

TUBES, Copper

American Brass Co., 1104-1105
Mueller Brass Co., 1106-1107
Revere Copper & Brass, Inc., 1108
Wolverine Tube Div., Calumet and Hecla Consolidated Copper Co., 1109

TUBES, Pitot (*See Air Measuring and Recording Instruments*)

TUBING, Aluminum

Revere Copper & Brass, Inc., 1108

TUBING, Copper

American Brass Co., 1104-1105
Mueller Brass Co., 1106-1107
Revere Copper & Brass, Inc., 1108
Wolverine Tube Div., Calumet and Hecla Consolidated Copper Co., 1109

TUBING, Fabricated

Arthur Harris & Co., 1115
Wolverine Tube Div., Calumet and Hecla Consolidated Copper Co., 1109

TUBING, Finned

Wolverine Tube Div., Calumet and Hecla Consolidated Copper Co., 1109

TUBING, Flexible Metallic

Atlantic Metal Hose Inc., 1101
Chicago Metal Hose Corp., 1102
Seamlex Co., Inc., 1103

TUBING, Plastic

Dow Chemical Co., The, 1114

TUBING, Steel

Babcock & Wilcox Co., The, 1150
Jones & Laughlin Steel Corp., 1099
Revere Copper & Brass, Inc., 1108

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General Electric Co. (Schenectady, N. Y.), 1076-1077
B. F. Sturtevant Co., Div., Westinghouse Electric, 954-959, 1018-1019
L. J. Wing Mfg. Co., 1000-1002
Worthington Pump & Machinery Corp., 1046-1047

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American Radiator & Standard Sanitary Corp., 1148-1149
Anderson Products, Inc., 1233
Crane Co., 1154-1155
Dole Valve Co., The, 1234
Hoffman Specialty Co., Inc., 1213-1215
Jenkins Bros., 1235
Sarco Co., Inc., 1218-1219

VALVES, Angle, Globe and Cross

Grinnell Co., Inc., 981, 1112-1113, 1211
Jenkins Bros., 1235
Mueller Brass Co., 1106-1107
Worthington Pump & Machinery Corp., 1046-1047
York Corp., 960

VALVES, Automatic

Anderson Products, Inc., 1233
Automatic Products Co., 1120
Barber-Colman Co., 1084, 1121
Crane Co., 1154-1155
General Controls, 1126-1127
Johnson Service Co., 1130-1131
McDonnell & Miller, 1194-1195
Minneapolis-Honeywell Regulator Co., 1136-1137
Webster Engineering Co., 1180

VALVES, Back Pressure

Alco Valve Co., 1118
Automatic Products Co., 1120
Illinois Engineering Co., 1216-1217
Mueller Steam Specialty Co., Inc., 1227
Spence Engineering Co., Inc., 1143

VALVES, Balanced

Illinois Engineering Co., 1216-1217

VALVES, Blow-off

Jenkins Bros., 1235
Yarnall-Waring Co., 1232

VALVES, By-Pass

Crane Co., 1154-1155
Jenkins Bros., 1235

VALVES, Check

Fedders-Quigan Corp., 980
Grinnell Co., Inc., 981, 1112-1113, 1211
Henry Valve Co., 1129
Illinois Engineering Co., 1216-1217
Jenkins Bros., 1235
York Corp., 960

VALVES, Diaphragm

Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
General Controls, 1126-1127
Grinnell Co., Inc., 981, 1112-1113, 1211
Henry Valve Co., 1129
Johnson Service Co., 1130-1131
Minneapolis-Honeywell Regulator Co., 1136-1137
Mueller Brass Co., 1106-1107
Mueller Steam Specialty Co., Inc., 1227
Parks-Cramer Co., 951
Powers Regulator Co., The, 1142
Taylor Instrument Cos., 1144
White-Rodgers Electric Co., 1146
York Corp., 960

VALVES, Expansion

Automatic Products Co., 1120
Crane Co., 1154-1155
Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
General Controls, 1126-1127
Henry Valve Co., 1129
Tenney Engineering, Inc., 1145
York Corp., 960

VALVES, Float

Alco Valve Co., 1118
Anderson Products, Inc., 1233
Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
Frick Co. (Inc.), 1041
Illinois Engineering Co., 1216-1217
McDonnell & Miller, 1194-1195
Mueller Steam Specialty Co., Inc., 1227
Viking Air Conditioning Corp., 1074
Vilter Mfg. Co., The, 1048
York Corp., 960

VALVES, Flow Control

Automatic Products Co., 1120
Barber-Colman Co., 1084, 1121
Bell & Gossett Co., 1196-1197
Crane Co., 1154-1155
Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
General Controls, 1126-1127
Illinois Engineering Co., 1216-1217
Johnson Service Co., 1130-1131
McDonnell & Miller, 1194-1195
Minneapolis-Honeywell Regulator Co., 1136-1137
Mueller Steam Specialty Co., Inc., 1227
Taco Heaters, Inc., 1200
H. A. Thrush & Co., 1198-1199

VALVES, Gate

American Brass Co., The, 1104-1105
Crane Co., 1154-1155
Detroit Lubricator Co. Div. of American Radiator & Standard Sanitary Corp., 1122-1123
Grinnell Co., Inc., 981, 1112-1113, 1211
Jenkins Bros., 1235
Mueller Brass Co., 1106-1107
United States Radiator Corp., 1172-1173

VALVES, Hydraulic

Crane Co., 1154-1155
General Controls, 1126-1127
Yarnall-Waring Co., 1232

VALVES, Magnetic

Alco Valve Co., 1118
Barber-Colman Co., 1084, 1121
General Controls, 1126-1127
Minneapolis-Honeywell Regulator Co., 1136-1137
White-Rodgers Electric Co., 1146

VALVES, Mixing, Thermostatic

Fulton Sylphon Co., 1124-1125
Powers Regulator Co., The, 1142
Sarco Co., Inc., 1218-1219

VALVES, Motor Operated

Barber-Colman Co., 1084, 1121
Bell & Gossett Co., 1196-1197
Crane Co., 1154-1155
General Controls, 1126-1127
Illinois Engineering Co., 1216-1217
Jenkins Bros., 1235
Johnson Service Co., 1130-1131
Powers Regulator Co., The, 1142

VALVES, Non-Return

Illinois Engineering Co., 1216-1217
Jenkins Bros., 1235

VALVES, Packless

C. A. Dunham Co., 1206-1210
Fulton Sylphon Co., 1124-1125
Henry Valve Co., 1129
Illinois Engineering Co., 1216-1217
Jas. P. Marsh Corp., 1228-1229
Sarco Co., Inc., 1218-1219

VALVES, Pressure Reducing (*See Regulators, Pressure*)

VALVES, Pump

Crane Co., 1154-1155
Jenkins Bros., 1235
Trane Co., The, 998-999

VALVES, Purge

Henry Valve Co., 1129
Mueller Brass Co., 1106-1107

VALVES, Radiator

American Radiator & Standard Sanitary Corp., 1148-1149
Barnes & Jones, Inc., 1205
Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
Fulton Sylphon Co., 1124-1125
Grinnell Co., Inc., 981, 1112-1113, 1211
Hoffman Specialty Co., Inc., 1213-1215
Illinois Engineering Co., 1216-1217
Jenkins Bros., 1235
Jas. P. Marsh Corp., 1228-1229
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999
Warren Webster & Co., 1220-1223

VALVES, Radiator Orifice

Barnes & Jones, Inc., 1205
C. A. Dunham Co., 1206-1210
Grinnell Co., Inc., 981, 1112-1113, 1211
Illinois Engineering Co., 1216-1217
Sarco Co., Inc., 1218-1219
Trane Co., The, 998-999

VALVES, Radiator, Pneumatic Diaphragm

Bell & Gossett Co., 1196-1197
Johnson Service Co., 1130-1131
Minneapolis-Honeywell Regulator Co., 1136-1137
Powers Regulator Co., The, 1142

VALVES, Reducing

American District Steam Co., 1226, 1236
Bell & Gossett Co., 1196-1197
Crane Co., 1154-1155
C. A. Dunham Co., 1206-1210
General Controls, 1126-1127
Illinois Engineering Co., 1216-1217
Spence Engineering Co., Inc., 1143
Taco Heaters, Inc., 1200
Taylor Instrument Cos., 1144
H. A. Thrush & Co., 1198-1199

VALVES, Refrigerant Line

Alco Valve Co., 1118
Automatic Products Co., 1120
Baker Ice Machine Co., Inc., 1038
Detroit Lubricator Co., Div. of American Radiator & Standard Sanitary Corp., 1122-1123
Frick Co. (Inc.), 1041
General Controls, 1126-1127
Henry Valve Co., 1129
Mueller Brass Co., 1106-1107
Tenney Engineering, Inc., 1145
Vilter Mfg. Co., The, 1048
Worthington Pump & Machinery Corp., 1046-1047
York Corp., 960

VALVES, Relief

Bell & Gossett Co., 1196-1197
Crane Co., 1154-1155
Henry Valve Co., 1129
Jas. P. Marsh Corp., 1228-1229
McDonnell & Miller, 1194-1195
Taco Heaters, Inc., 1200
H. A. Thrush & Co., 1198-1199
York Corp., 960

VALVES, Safety

Henry Valve Co., 1129
McDonnell & Miller, 1194-1195
D. J. Murray Mfg. Co., 990

VALVES, Solenoid

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MANUFACTURERS' CATALOG DATA

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On pages 939-1280 will be found the Catalog Data of 233 manufacturers whose products are described and illustrated.

For the convenience of the user of THE GUIDE 1946 there are eight main divisions:

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On pages 913-936, 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.

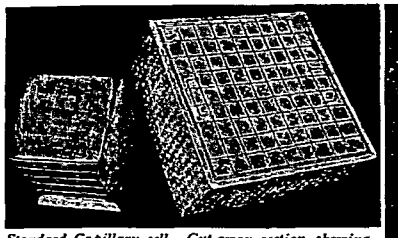
By reference to these indices, the manufacturers names and the page numbers, any item of equipment or materials, and the producers address, may be located quickly.

Air & Refrigeration Corporation

475 Fifth Avenue, New York 17, N. Y.

Atlanta, Ga. Detroit, Mich.

CAPILLARY AIR CONDITIONERS



Standard Capillary cell. Cut-away section showing oriented glass filaments. Size: 20 in x 20 in. x 8 in.

Air Conditioning Engineers and Industrial Engineers responsible for air conditioning should be familiar with the uses of this advanced equipment.

The standard Capillary cell is the basic element in all Capillary conditioners. The patented arrangement of glass filaments, essentially parallel to the flow of air and water through the cell, accounts for the highly efficient heat transfer between air and water as well as low resistance. At the same time, the cells act as an efficient air cleaner and the arrested dirt is continuously flushed from the cell.

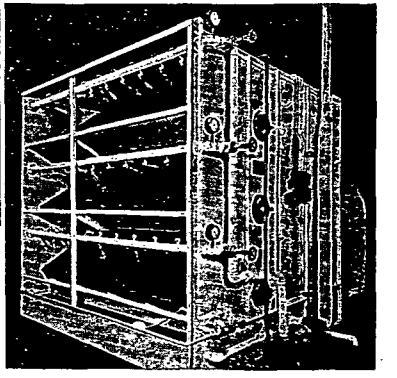
As a simple air washer, humidifier or evaporative cooler. Class I Capillary conditioners call for the recirculation of only 3 gpm per 1000 cfm distributed over the cells at 6 lbs nozzle pressure. The saturation efficiency is 97 per cent. Less efficient spray washers require 15 gpm or more at 20 lbs nozzle pressure.

For cooling and dehumidifying a single stage of Capillary cells equals or exceeds in capacity a 2-bank spray type dehumidifier.

Other Capillary types offer true counterflow performance with leaving cooling water temperature exceeding that of leaving air.

Where a closed system for the cooling medium is required (direct expansion, brine or cold water), Capillaries are offered with suitable coils after the Capillary cells. No filters are required. Coils are kept clean and evaporative cooling is available whenever entering wet-bulb conditions permit.

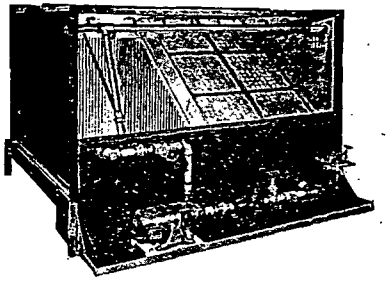
Simple, continuous water treatment is available to protect coils and other metal parts against corrosion.



A standard Size 6-6 Class I Capillary Central Station Installation.

Varied uses for Capillary conditioners are suggested by the intimate contact of fluid and gas passing through the cells. Among these are: condenser water cooling; fume absorption, aeration and concentration; air sterilization with agents added to water.

Capillary conditioners of all classes are made in central station units ranging from 2200 cfm to 132,000 cfm or larger. Assembled units including fans, heaters, coils, pump, insulated casing, etc., suitable for suspension or floor mounting range from 1000 to 16,000 cfm.



A standard Size 3-4 Capillary unit air conditioner complete in insulated casing. Capacity 16,000 cfm.

Submit design and capacity for specific recommendations or write for catalog and engineering data.

American Blower Corporation

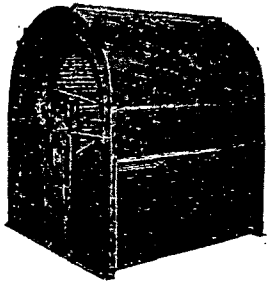
P. O. Box 58, Roosevelt Park Annex, **Detroit 32, Michigan**

CANADIAN SIROCCO COMPANY, LTD.
in Canada, 310 Ellis Street, Windsor, Ontario

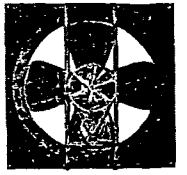
Branch Offices in Principal Cities

Division of American Radiator and Standard Sanitary Corporation

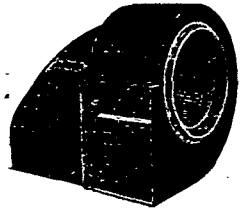
AIR CONDITIONING — HUMIDIFYING — DEHUMIDIFYING — COOLING VENTILATING — HEATING — VAPOR-ABSORPTION — DRYING — AIR WASHING AND PURIFICATION — EXHAUSTING EQUIPMENT AND MECHANICAL DRAFT APPARATUS



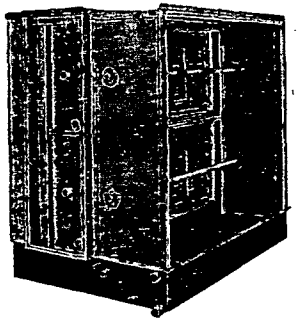
Double Inlet "ABC" Multiblade Fan—above, is a heavy duty ventilating fan. Its wheel has narrow, forward pitched blades. Low tip speeds assure quiet operation. Request Bulletin A-701. Write for Bulletin A-403 for backwardly inclined, non-overloading H. S. Fan.



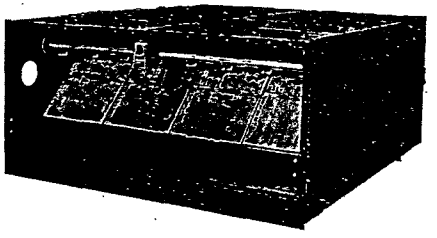
Commercial V-Belt Drive Ventura Fans—for ventilating applications without duct systems, where extremely quiet operation is desired. Inlet-outlet streamlined for high efficiency. Also direct connected. Request Bulletin B-2529.



"ABC" Utility Sets—right, complete packaged units, direct connected or V-Belt short coupled drive, for duct applications. Famous "ABC" Multiblade Wheel operates at low tip speeds. Quiet, compact. Bulletin B-2529.



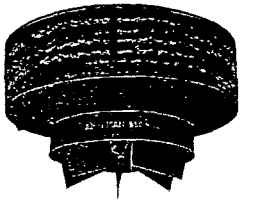
American Blower Air Washer—above, cleans, purifies and freshens the air, removes dust, odors and bacteria, cools if desired and provides an effective method of controlling humidity. Bulletin 3623.



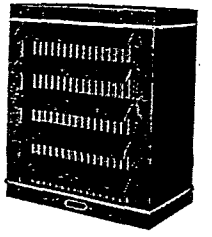
American Blower Capillary Air Washers—above, for high efficiency in cleaning, humidification, cooling and dehumidification of air. A highly efficient surface contact mechanism, the capillary cell, is used. Air is forced at low resistance through long, irregular passages of small size formed by a large amount of thoroughly wetted glass surface. Unit includes a substantial metal casing and tank of air washer design, capillary cells, improved low head sprays, metal or glass fibre low resistance moisture eliminators, non-ferrous, extended surface cooling or heating coils. Write for Bulletin 3723.

TYPES OF AMERICAN BLOWER CORPORATION AIR HANDLING AND CONDITIONING EQUIPMENT

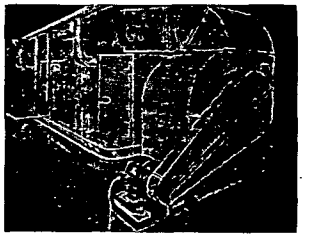
All types of air handling and air conditioning equipment for industrial applications, process work, drying, cooling; also equipment for stores, offices, shops, public buildings, power plants, etc., and attic ventilation for homes.



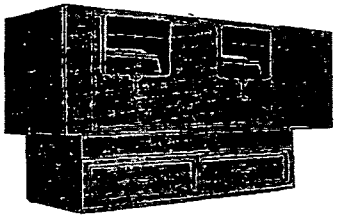
"ABC" Vertical Heaters—for ceiling applications, give an even, wide floor area distribution of heat. For either steam or hot water heating systems. Variable speed, 2-speed and constant speed models. Write for Bulletin A-9418.



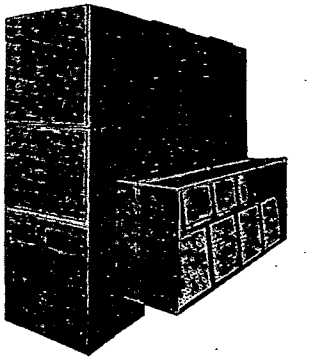
Venturafin Unit Heaters—for many general purpose heating jobs. Wall or ceiling mounting. Streamline construction, rugged heating elements. Steam or hot water. Write for Bulletin A-8218.



Air Conditioning Central Systems—provide an effective way of cooling, heating, humidifying, dehumidifying and purifying air in all classes of business and public buildings where a dust system is desirable. Write for Special Data.



"HV" General Purpose Units—with air filters and Aileron control. Ideal wherever attractive, quiet and economical heating and ventilating units are required. Wall, floor or ceiling mounting. Offer great flexibility of design and arrangement to meet specific needs. Write for Bulletin 5927.



American Blower Series "H" Air Conditioners with Sprayed Coils—are usually applied for industrial uses where air washing and evaporative cooling are required. Sprayed coils give cleaner air, cut coil maintenance and refrigeration costs, reduce necessary air volumes, permit use of smaller ducts and grilles. Horizontal or floor types (as shown). Aileron control provides simple method of regulating flow of air from the fans. Write for Bulletin 6027.

The Bahnson Company

AIR CONDITIONING ENGINEERS

Winston-Salem, N. C.

Offices:

NEW YORK, N. Y., 93 Worth St.

ATLANTA, GA.....886 Drewry St.
WESTFIELD, N. J.....703 Embree Crescent
HAMILTON, ONT., CAN.....W. J. Westaway Co., Ltd.
LOS ANGELES, CALIF.....553 S. Figueroa St.

HUMIDIFYING—COOLING—HEATING—AIR CLEANING—VENTILATING

Bahnson Industrial Air Conditioning and Humidifying Equipment has been installed throughout the United States and in over 30 foreign countries for more than three decades.

In designing various types of equipment to meet the requirements of Industrial Air Conditioning and Humidifying, the Bahnson Company has considered as paramount: Flexibility of capacity and control, Economy of Operation, and Simplicity of Design.

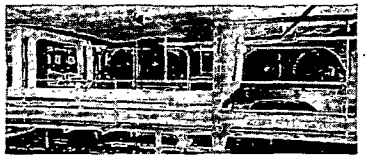
CENTRISPRAY

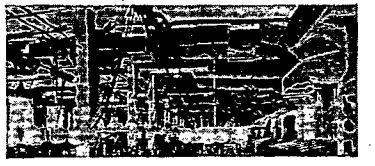
Certain types of industrial air conditioning problems lend themselves more readily to central station systems or unitary systems that deliver saturated air.

The Bahnson Company has designed and installed a large number of saturated air systems of both unitary and central station types.

Conventional air washers manufactured by the Bahnson Company show a high efficiency of operation in the low pumping cost and low static pressures.

The Bahnson Centrispray is a development of an air washer to deliver saturated air, based on the principle of centrifugal atomization. This washer affords an extremely low operating cost, approaching that of capillary washers. Maintenance of the Centrispray is extremely low, since there are no nozzles to stop up and the centrifugal discs are self-cleaning.





Bahnson central station installations as employed with refrigeration, and as employed for evaporative cooling only.

AIR-VITALIZER

The Centrispray can be engineered for small compact units or large central station units. It may be designed for use with electrostatic air cleaning or refrigeration, as well as for evaporative cooling.

The Bahnson Air-Vitalizer air conditioning system employs a dry duct unit distribution system combined with the Bahnson Centrifugal Humidifier or the Bahnson Atomizer, to obtain evaporation.

The Vitalizer Units are usually arranged in the room so that the general movement of air is carried down one side of the room and back the other side of the room. This feature, known as *horizontal circulation*, evens out the condition in the room despite the concentrated heat loads in motor alleys and sections of machinery with high horsepower requirements.

The flexibility of the Vitalizer system permits the system to be designed for any air handling or evaporative capacity.

ECONOMIZER

ATOMIZER

The Bahnson Economizer, an atomizer embodying a new principle of variable capacity, will produce a finer spray at a given evaporation, with minimum air pressure and air consumption.

It is an outstanding development in the atomizer field, featuring Flexibility, Economy, and Simplicity of installation, operation and design.

The Bahnson Company

Winston-Salem, N. C.

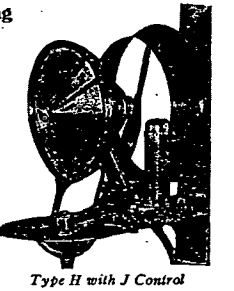
CENTRIFUGAL HUMIDIFIER

With this unit, Bahnson has pioneered in developing the principle of centrifugal atomization for over 30 years. Continuous refinement has produced three improved types of units with varying capacities to meet different requirements. (Types D, H and L).

The Bahnson Centrifugal Humidifier requires only feed water at city main pressures (15 to 30 lbs.), a small amount of electric current at the usual voltage specifications, and a drain line. Equipped with an individual control, it is the only type of self-contained industrial humidifying unit that will produce a fine mist, distribute this mist throughout the space to be conditioned, maintain air circulation within that space, and automatically control the humidity within that space.

These units may be equipped with the Type J individual control, or may be operated with the highly sensitive Master B group control that actuates an electric motor valve on the feed water line to a group of humidifiers.

The Atomizing Disc is mounted on one end of the fractional horsepower motor. A row of stationary atomizing teeth surround the periphery of this disc. Water, fed onto the rapidly whirling disc, is hurled against the stationary teeth and broken into a fine, cloudlike mist. A fan mounted on the back end of the motor shaft distributes this fine mist and produces a directional air flow. This maintains air circulation and uniformity of condition throughout the space to be conditioned.



Type H with J Control

Write for Bulletin 327S, or special bulletins on special applications.

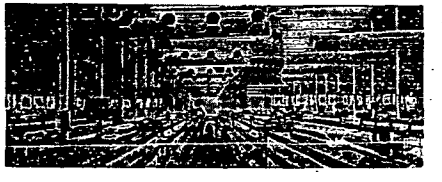
HUMIDUCT

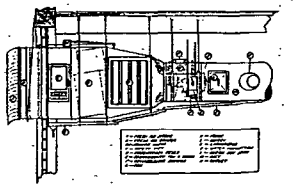
AIR CONDITIONING SYSTEM

The Bahnson Company has led in the development of two new principles of industrial air conditioning for more than a decade with its "Humiduct" system of air conditioning.

The Bahnson Humiduct was the first practical unit for utilizing the advantages of delivering saturated air plus entrained moisture from an air conditioning system, rather than saturated air alone to meet certain air conditioning requirements. Where it is necessary to maintain constant relative humidities in areas with concentrated heat loads, this principle of delivering entrained moisture so that part of the evaporation may take place in the room, shows marked advantages in more accurate control of the humidity. The humidity can be maintained by the use of this principle with a much lower volume of air than with saturated air types of systems.

The Bahnson Humiduct is designed as a unit system of air conditioning. It has shown marked advantages in maintaining uniform conditions throughout the rooms of industrial plants that have relatively high heat loads or varying concentrations of machinery and heat. These advantages together with the flexibility of operating the unit system, have been widely recognized. The Bahnson Company introduced the principle of *horizontal circulation* with a unit system. The units are installed to produce a general air movement down one side of a room and back the other. This produces unusual evenness in the room conditions together with a sensible cooling effect.





Carrier Corporation • Syracuse 1, N. Y.

MARINE DIVISION:
405 Lexington Ave.
New York 17, N. Y.

Carrier

INTERNATIONAL DIVISION:
122 East 42nd St.
New York 17, N. Y.

Offices and Dealers in principal cities—refer to your telephone directory.

Carrier AIR CONDITIONING

Room Air Conditioners—compact units in attractive cabinets. Built to provide summer comfort air conditioning for individual rooms, private offices, and other small enclosures.

Self-Contained Air Conditioners—completely enclosed in neat cabinets, these units provide summer comfort for retail shops, general offices, beauty salons, and other commercial spaces of medium size.

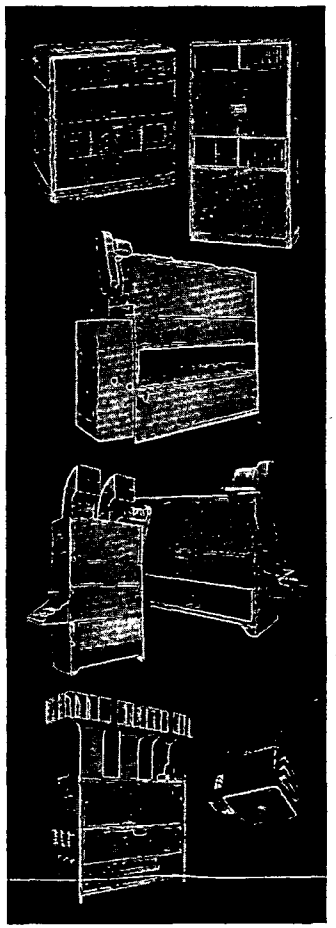
Assembled Air Conditioners—fully enclosed, compact units designed for installation outside the space to be air conditioned, and using ducts to distribute the air. Ideal for year round conditioning of laboratories, offices, stores, and similar interiors.

Unitary and Central Station Air Conditioners—for groups of rooms such as offices and laboratories, and for large spaces such as stores, factories, theaters, industrial plants, and other interiors requiring year round air conditioning. Units available in floor or suspension models. Supplemented by refrigeration where cooling and dehumidifying is required.

"Weathermaster" Systems—for multi-room buildings such as apartments, hospitals, hotels, and office buildings. System consists of room units, each with individual control of temperature, and a central station system. In one system the air is distributed through conduits requiring but little space—practical for new or old buildings. An exclusive development of the Carrier Corporation.

Blast Freezers and Cold Diffusers—for food freezing and storage, meat packing operations, and other industries requiring low temperatures. Units are available in suspension or floor models and can be used within the space to be refrigerated or remotely located and connected by ducts.

Dehydration Units—for industrial applications such as drying, chemical processing, and all production requiring low moisture content air. Dehydrated air is distributed through ducts, and refrigeration may be added to achieve low temperatures.



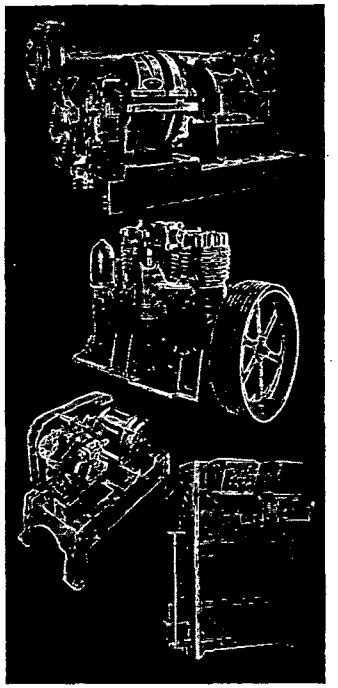
Carrier REFRIGERATION

Centrifugal Refrigerating Machine—for large comfort and industrial air conditioning applications and for cooling processes down to below −100 F. These efficient machines operate with any standard drive, are practically vibrationless, and require little supervision. Uses safe refrigerant. Available in capacities from 100 to 1200 tons cooling.

Reciprocating Refrigerating Machines—to provide refrigeration for industrial and comfort air conditioning of moderate size, and for cooling processes down to below zero temperatures. Complete machines include a belt drive compressor, a water-cooled condenser, receiver, and necessary connections available for evaporative cendenser. Capacities from 5 to 60 hp.

Evaporative Condensers—for condensing refrigerants and cooling liquids by a process of blowing air over wetted coils. The evaporation of the water from the surface of the coils removes a maximum amount of heat with a minimum of water consumption. Can be used in place of a cooling tower. Afford savings wherever water costs are high. May be placed outdoors without protection. Capacities range from 6 to 75 tons condensing.

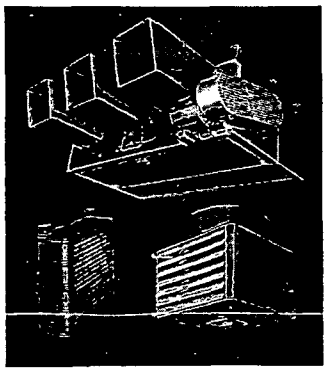
Commercial Refrigeration Compressors—for storage refrigerators, display cases, milk coolers, ice makers, farm and home food freezers in a wide range of capacities and temperatures. Units use "Freon" or Methyl Chloride refrigerants and are complete with compressor, drive, air or water cooled condenser, and controls mounted on a one-piece base.



Carrier INDUSTRIAL HEATING

Unit Heaters—for commercial and industrial heating uses. Available in two suspended types: one horizontal discharge model for small space heating and one vertical 4-way discharge model for larger areas. Both models have adjustable louvres for directional control. Units consist of propellor type fan, coils for steam or hot water, and drive, all neatly encased. Available with manual or thermostatic controls. Capacities range from 32,000 to 579,000 Btu per hour at 2 lb steam pressure.

Heat Diffusing Units—for commercial and industrial buildings. Suspended or floor models with centrifugal type fan, coils for steam or hot water, and selective air distribution, all factory assembled for easy installation. Capacities are from 105,000 to 1,035,000 Btu per hour at 2 lb steam pressure.



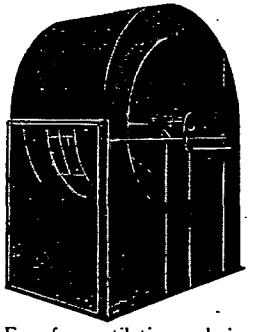
Clarage Fan Company

Kalamazoo, Michigan

Application Engineering Offices In Principal American Cities

(Consult Telephone Directory)

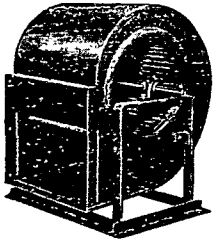
Clarage Air-Handling and Conditioning Equipment



Fans • Blowers • Air Washers Air Conditioning Systems and Units • Unit Heaters & Coolers

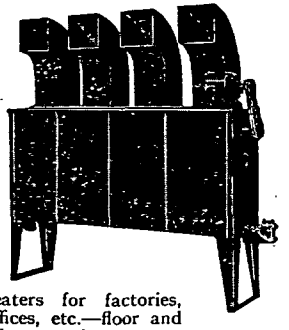
For over a quarter-century Clarage has been a leading manufacturer of equipment for ventilating, heating, cooling, drying, air cleaning, humidifying, dehumidifying, complete air conditioning, exhausting, pneumatic conveying and mechanical draft. This equipment is designed to meet all types of industrial, commercial, public building and marine requirements. Whatever your air-handling or conditioning problem, Clarage is an excellent source of supply.

Fans for ventilating and air conditioning.
28 sizes; 200 to 100,000 cfm.



Fans for warm air furnaces, oil burners, stokers, etc.
200 to 5000 cfm.

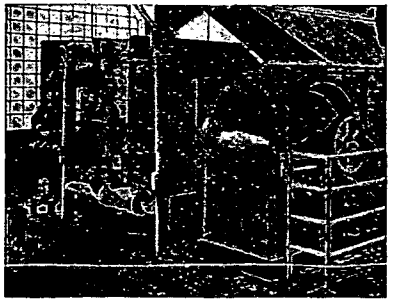
We build many other types of fans and allied equipment. Write for a Clarage catalog covering our complete line.



Unit Heaters for factories, stores, offices, etc.—floor and suspended type units.
24 standard sizes.

Fans for exhaust systems, pressure blowing, etc.
500 to 50,000 cfm.; built for a wide range of pressures.

Air Conditioning central systems and units to solve any temperature and humidity control problem.



Hastings Air Conditioning Co., Inc.

Hastings, Nebr.

Manufacturers of

AIR-CONTROL AIR CONDITIONING

Air Conditioners.
Unit Heaters.
Utility and Package Blowers.

Dealers and Representatives in Principal Cities

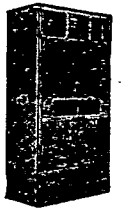
A Complete Line of Highly Successful COLD WATER Air Conditioners. Capacities listed depend on entering air and water temperatures.

All equipment available for combination heating and cooling.

FLOWMETERS (to visually control water flow) standard on all models.

FLOOR MODELS

Floormasters—Unusual design and special features permit maximum installation possibilities with minimum floor space and installation costs.

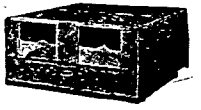


Air Delivery—2240 cfm. Cooling Capacity—3 to 6 tons. Dimensions—Height 93 in., Width 48 in. Depth 25 in. Motor—½ hp. Filters—3 16 in. x 25 in.

Royal—For offices, homes, hospitals, etc.

Air Delivery—590 cfm. Cooling Capacity—1 to 2 tons. Motor—1/6 hp. Filter—1 16 in. x 25 in. Dimensions—Height 40 in., Width 28 in., Depth 20½ in.

(C.P.) CENTRAL PLANTS



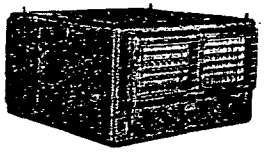
Sectional construction for ease of handling. Motors inside mounted to provide very neat appearing compact units.

SPECIFICATIONS

Size	CFM	Motor Hp	Filters	Capacity Tons
CP 30	3,000	1	5	4- 9
CP 40	4,000	1	8	6-12
CP 60	6,000	2	10	9-18
CP 80	8,000	3	12	12-24
CP120	12,000	5	20	18-36

GENERAL UTILITY MODELS

Master—Singly or in multiple are suitable for any business or space size. Large jobs handled without duct work by proper location of units.



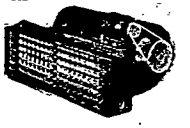
Air Delivery—2,240 cfm. Cooling Capacity—3 to 6 tons. Dimensions—Height 29 in., Width 49 in., Depth 50 in. Motor—½ hp. Filters—4 16 in. x 23 in.

Majestic—Similar to Master except size. Air Delivery—1120 cfm. Cooling Capacity—1½ to 3 tons. Motor—¼ hp. Filters—2 16 in. x 25 in. Dimensions—Height 26 in., Width 28 in., Depth 40 in.

Zephyr—Same capacity, motor and filter as the Royal. For use where suspended or concealed units are desired. Dimensions—Height 26 in., Width 24 in., Depth 28 in.

UNIT HEATERS

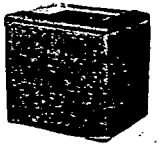
Centrifugal Type for extreme quietness and efficiency.



Steam pressure—to 150 lbs per sq in.

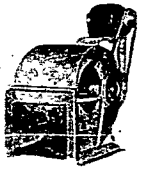
Finish—Brown wrinkle enamel and stainless steel louvers.

PACKAGE AND OPEN TYPE BLOWERS



May be knocked down for narrow doorways. Finished in attractive green wrinkle.

Utility type blowers are available with or without motors and in any discharge desired.



All sizes from 9 in. to twin 21 in.
Air deliveries from 1000 cfm to 16,000 cfm.

Write for Catalogues, Literature, or Information

GENERAL ELECTRIC

AIR CONDITIONING DEPARTMENT
BLOOMFIELD NEW JERSEY

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1405 Locust Street

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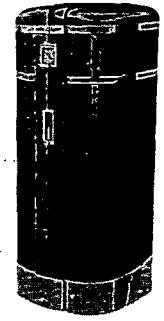
P. O. Box 909

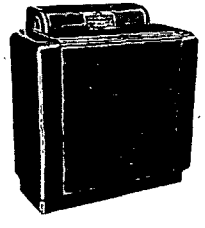
AUTOMATIC HEATING EQUIPMENT

G-E OIL-FIRED BOILER available in 7 types with a range of from 75,000 to 450,000 Btu per hour output. G-E OIL-FIRED CONVERSION BURNER attachment type, 141,000 to 423,000 Btu per hour input. G-E OIL-FIRED WARM AIR CONDITIONER factory wired and assembled for installation, 120,000 Btu per hour output. G-E GAS-FIRED BOILERS available in 8 types, 76,000 to 345,600 Btu per hour output. G-E GAS-FIRED WARM AIR CONDITIONER packaged unit available in 5 models ranging in air heating capacities from 48,000 to 168,000 Btu per hour output.



G-E Oil-Fired Boiler

PACKAGED AIR CONDITIONING EQUIPMENT



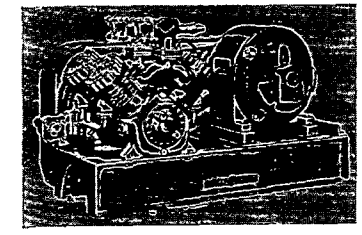
G-E Floor Mounted Room Air Conditioner

G-E FLOOR MOUNTED ROOM AIR CONDITIONER provides cooling, dehumidification, circulation, ventilating and filtering, 40⅝ in. high, net cooling capacity of 8,000 Btu per hour at *A.S.R.E.* standard rating conditions. G-E SELF-CONTAINED AIR CONDITIONERS for domestic or commercial applications, with or without ducts—optional equipment makes it possible to provide heating and humidification in winter, for complete year 'round air conditioning.

COMMERCIAL REFRIGERATION and WATER COOLERS

G-E WATER COOLERS bottle type, holding 3 or 4 gal. water bottles, cools 2.4 gal. per hour *A.S.R.E.* standard capacity rating. G-E FOOT PEDAL OPERATED WATER COOLER, ranging in cooling capacity from 5.0 to 19.8 gal. per hour *A.S.R.E.* standard capacity ratings. G-E FOOD STORAGE CABINETS in a range of sizes for restaurants, hospitals, hotels, grocery stores, and meat markets.

CENTRAL PLANT AIR CONDITIONING AND REFRIGERATION EQUIPMENT



Refrigeration Condensing Units 1/6 HP to 125 HP—A complete line of reciprocating type condensing units for use with "Freon" refrigerants. Air cooled models available from 1/6 hp to 3 hp. Motor-compressor units for use with evaporative condensers or cooling towers available from 3 hp to 125 hp.

Evaporative Condensers—For use with refrigeration motor-compressor units in those applications where cooling water is scarce or contaminated, or where water rates are high—a line of eight standard sizes ranging in condensing capacities from 3 to 100 tons of refrigeration with "Freon-12."

Evaporative Coolers—Provide water saving and economic advantages in those applications where large quantities of cooling water are used, the temperature of which need not be lower than five degrees above maximum wet-bulb temperature. Available in eight standard models ranging in capacity from 250,000 to 3,000,000 Btu per hour for water.

Unit Coolers—A line of wall and ceiling-mounted "Conditioned Air" cooling units designed to maintain high humidity for correct preservation of perishable foods. General Electric also offers a line of product coolers for such applications as refrigerated warehouses, fur storage plants, large locker plants and other large refrigerated storage spaces.

Central Plant Air Conditioners—A line of horizontal units ranging in size from 2½ to 30 tons of refrigeration in cooling service with comparable capacities for heating service. General Electric will also furnish vertical air conditioners ranging in size from approximately 2 to 50 tons of refrigeration in cooling service, with comparable capacities for heating.

Heating and Cooling Coil Surface—High heat transfer efficiency finned copper surface for use with steam, hot water, chilled water, or refrigerant. Particularly valuable for all types of built-up air conditioners or blast coils. Designed for multiple mounting in housings or ducts, these coils can be bolted together to form a continuous unbroken surface.

Complete Air Conditioning Systems—Multiple unit systems for the air conditioning of hotels, office buildings, apartment houses, and hospitals. Attractive, compact room units make possible personal control of conditions in individual rooms. Refrigeration in central plant furnished by reciprocating type condensing units or centrifugal refrigeration units as requirements dictate.

Niagara Blower Company

General Sales Office: 6 East 45th Street, New York 17, N. Y.

CHICAGO-5: 37 W. Van Buren St. BUFFALO-7: 673 Ontario St. SEATTLE-4: Fourth and Cherry Bldg.

District Engineers in Principal Cities

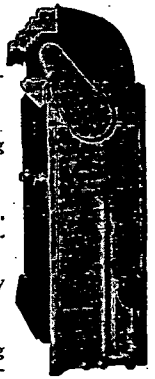
Over 30 Years' Experience in Industrial Air Conditioning, Liquid Cooling and Air Drying

NIAGARA AERO HEAT EXCHANGER

For cooling industrial liquids, water, oils, solutions, chemicals, compressed air and gases, with Niagara "Balanced Wet-Bulb" temperature control to improve efficiency and obtain precise results. Patented (U. S. Nos. 2,296,946 and R. I. 22,553). Ask for Bulletin 96.

NIAGARA AIR CONDITIONING SYSTEMS

For human comfort and for all industrial applications requiring controlled conditions of temperature, relative humidity, air purity and air movement.



Niagara "No Frost" Cooler

NIAGARA AIR CONDITIONER, TYPE A

High precision apparatus using saturation to obtain control of R. H. to 1 per cent for laboratory work and control of hygroscopic materials. Ask for Bulletin 58.

NIAGARA AIR CONDITIONER, TYPE C

A year around air conditioning unit providing heating and humidifying or dehumidifying. Ask for Bulletin 80.

NIAGARA FAN COOLER AND DISK FAN COOLER

For comfort cooling, process cooling, low temperature storage for dairies, fruits, meats, food products, fur storage vaults, etc. Bulletins 72 and 78.

NIAGARA SPRAY COOLER

For all cooling applications requiring high humidity or high capacity in small space. Ask for Bulletins 72 and 78.

NIAGARA "NO FROST" SYSTEM

Using Niagara "No Frost" Liquid in spray coolers, prevents frosting of cooling coils, automatically keeps spray solution at proper concentration, gives freedom from brine troubles, corrosion. Constant, efficient operation. Temperature to −100 F. Ask for Bulletins 83 and 95.

NIAGARA EXTENDED SURFACE COILS

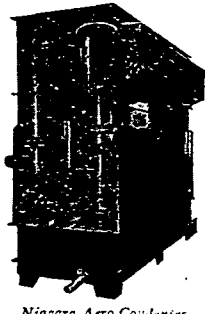
Encased for use with heating, cooling or air conditioning systems. Full range of sizes. Ask for Bulletin 92.

NIAGARA DUO-PASS AERO CONDENSER (Illustrated)

Saves power and water cost utilizing atmospheric air to remove heat of condensation. Patented Duo-Pass prevents scaling, saves power. "OILOUT" positively removes oil and dirt from refrigerant lines, assuring always full capacity. Ask for Bulletins 91 and 93.

NIAGARA "DUAL" COOLERS

Simultaneously cools a room and furnishes chilled water as a refrigerant. Saves equipment cost, operating expense. Patented. Ask for Bulletin 70.



Niagara Aero Condenser with "Oilout"

NIAGARA INDUSTRIAL LIQUID COOLER

Furnishes refrigerated water or aqueous solution in any quantity up to 220 gpm. Positive control of temperature regardless of load variation. Delivers "sweet" water at 33° F without danger of freezing damage.

NIAGARA FAN HEATERS AND DISK FAN HEATERS

For heating and ventilating large areas. Units of the highest quality in engineering, material and workmanship. Ask for Bulletin 97.

NIAGARA AIR SUPPLY HEATER

Balances exhausted air in factories when exhaust systems are operating, saves steam and power, gives more effective heating. Patent pending. Ask for Bulletin 74.

NIAGARA MOTOR BLOWERS

One, two and three-fan units. High and low static pressure models. Ask for Bulletin 89.

Parks-Cramer Company

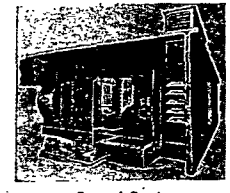
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

Complete Air Conditioning Systems including Heating, Cooling, Humidifying or De-humidifying, Air Changing, Refrigeration, Air Filtering, Air Washing

AUTOMATIC REGULATION

Merrill Process System of Hot Oil Circulation for Heating Industrial Materials



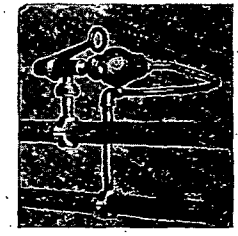
Central Station

Central Station Air Conditioning

Centrally located AIR WASHER. Proper moisture. Positive, pre-determined air removal or re-circulation. Heating coils and refrigeration optional. Helps such industries as Celluloid; Cement; Ceramics; Cereals; Cigars, Cigarettes and Tobacco; Clothing; Confectionery; Glassine; Leather; Paper and Envelopes; Printing and Lithographing; Shoes; Starch and Dextrine; Storage of Perishables; Textiles; Wood Products. Similar installations effective in Hospitals, Art Galleries, Auditoriums, Restaurants.

Air Washer or Central Station Units.

Nozzles for Central Station Air Washers.



Turbomatic Humidifier

Turbomatic Humidifier

Efficient humidifier of the atomizer type. For direct humidification, as humidity boosters for Central Station systems of all makes. Self-cleaning, both air and water ports. Streamlined to prevent lint and dirt accumulation.

High Duty Humidifier

(not illustrated)

Water under pressure generates spray. Excess water returns to filter tank and re-circulates. Evaporation per unit high; two sizes of heads each with three sizes of nozzles give flexible capacity for varying conditions. Circulation increased by individual motor-driven fan. Spray thoroughly diffused and distributed over wide area.

Psychrostat

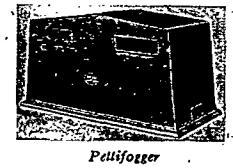
Parks Automatic Airchanger

(not illustrated)

For use with High Duty or Turbomatic Humidifiers. Insures fixed humidity and maximum evaporative cooling.

Automatic Regulation

The Psychrostat for accuracy, durability, sensitivity. Employs the principle of the Sling Psychrometer, used in all U. S. Weather Bureau Stations. Hygrostat (not illustrated) where requirements are not so exacting. An Air Conditioning System is no better than its Regulation.



Pettifogger

The Pettifogger

A compact humidifier for offices, stores, storerooms, laboratories, or other isolated departments. Self-contained in lacquered copper casing. Permanently though flexibly connected to water and electrical supplies. Automatic control. Adjustable capacity. Reduces dust. Neutralizes drying effect of heating.

Rogers Diesel and Aircraft Corporation

manufacturers of

ROGERS-RESEARCH SYSTEMS OF AIR CONDITIONING AND AIR STERILIZATION

1120 Leggett Avenue **New York 59, N. Y.**

AIR CONDITIONING THROUGH DEHUMIDIFICATION

The Rogers-Research System of Air Conditioning employs a chemical—triethylene glycol—to remove unwanted moisture from the air. Precise humidity and temperature conditions are maintained automatically and economically by this separate control of temperature and humidity. Conditioned air can be obtained at a low dew point with little change between the temperature of the incoming and the outgoing air.

The Rogers-Research System features simplified equipment, low maintenance, and operating costs, single unit installation for continuous duty operation with minimum space requirements.

Industrial and Comfort Air Conditioning—The following table indicates indoor conditions which can be obtained through a Rogers-Research Air Conditioning installation:

When Available Cooling Water Temperature for Absorber Coil is . . .
40° 45° 50° 55° 60° 65° 70° 75° 80° 85° 90° F

Then Temperature of Air Leaving Absorber can be as low as . . .
45° 50° 55° 60° 65° 70° 75° 80° 85° 90° 95° F

And the Dew Point of Air Leaving Absorber can be as low as . . .
12° 16° 20° 24° 28° 33° 37° 41° 46° 50° 54° F

These conditions can be varied to the requirements of the purchaser.

AIR STERILIZATION

Triethylene glycol used in the Rogers-Research Systems sterilizes air, renders inactive harmful, air-borne bacteria. All air passing through a Rogers-Research Air Conditioning unit comes in contact with this air sterilizing agent. In addition there is a complete line of Rogers-Research vaporizers for evaporating triethylene glycol into the air. These vaporizers can be used in conjunction with an air conditioning system, or independent of any air conditioning system. Vaporizers are made in sizes ranging from small table models for home use, to large, cabinet-sized units for hospitals, factories or large commercial premises.

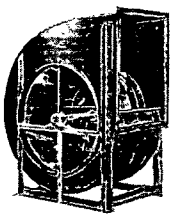
Rogers-Research Air Conditioning and Air Sterilization equipment is available in a wide range of capacities for many applications. For complete information, write to the Rogers Diesel and Aircraft Corporation, Department A.

United States Air Conditioning Corporation

Heating, Cooling, Ventilating and Air Conditioning Equipment

For Industrial, Commercial and Residential Applications

General Offices and Factory: Northwestern Terminal, **Minneapolis, Minn.**



Blowers, Exhausters and Fans

USAirCo Blowers
Heavy and light duty blowers, single or double inlet, in sizes and capacities for any heating, cooling, ventilating and air conditioning application.

USAirCo Air Washers
Single, double or triple stage 2,500 to 100,000 cfm for cleansing, cooling by cold water or refrigerant, humidifying or dehumidifying.

USAirCo Heating Units
Suspended types with Deflecto diffusing grilles. Floor or wall type blower heaters. Sizes and types for every heating need.

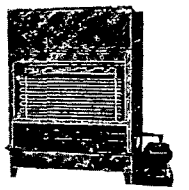
USAirCo Cooling or Heating Cores
Five standard series for central station heating or cooling applications.

USAirCo Cooling Units
Suspended type for cold water or direct expansion applications.

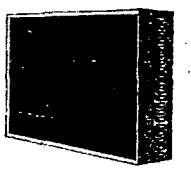
USAirCo Blower Filters
Complete assemblies for warm-air furnace applications.

Kooler-aire Package Units
Complete self-contained units for refrigerative, cold water and evaporative cooling. Also room coolers and humidifiers.

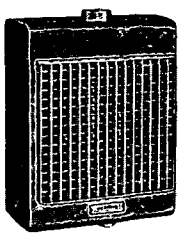
USAirCo Deflecto Grilles
Patented diffusing grilles for controlled directional distribution of air.



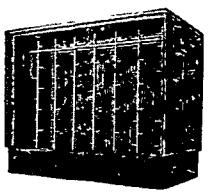
Evaporative Condensers



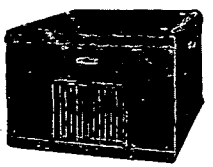
Cooling and Heating Coils



Unit Heaters

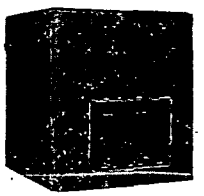


Air Washers

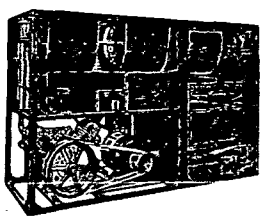


Coil Cooling Units

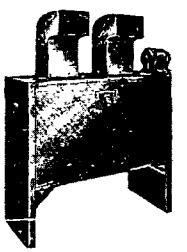
Write for Latest USAirCo Catalog



Blower-Filter Units



Refrigerated Air Conditioning Units



Floor Type Heaters

B. F. STURTEVANT COMPANY

Division of Westinghouse Electric

Air Conditioning, Heating, Ventilating, Dust Control and Fume Removal Equipment, Vacuum Cleaners, Dryers, Compressors, Motors, Turbines, Mechanical Draft Equipment

Main Office and Works

Hyde Park, Boston, Mass.

Sales Engineering Offices

Akron, Ohio				Richmond, Va.
Albany, N. Y.				St. Louis, Mo.
Atlanta, Ga.				Salt Lake City, Utah
Baltimore, Md.	Dallas, Texas	Houston, Texas	Minneapolis, Minn.	San Francisco, Calif.
Boston, Mass.	Denver, Colo.	Indianapolis, Ind.	Newark, N. J.	Seattle, Wash.
Buffalo, N. Y.	Des Moines, Iowa	Jacksonville, Fla.	New Orleans, La.	Spokane, Wash.
Camden, N. J.	Detroit, Mich.	Kansas City, Mo.	New York, N. Y.	Springfield, Mass.
Chicago, Ill.	El Paso, Texas	Little Rock, Ark.	North Hero, Vt.	Syracuse, N. Y.
Cincinnati, Ohio	Galveston, Texas	Los Angeles, Calif.	Phoenix, Ariz.	Tampa, Fla.
Cleveland, Ohio	Greensboro, N. C.	Memphis, Tenn.	Pittsburgh, Pa.	Toledo, Ohio
Columbus, Ohio	Hartford, Conn.	Milwaukee, Wis.	Portland, Ore.	Washington, D. C.

PLANTS: Located at HYDE PARK, BOSTON, MASS.; LaSALLE, ILL.; CAMDEN, N. J.; BERKELEY, CALIF.; and GALT, ONT.

B. F. STURTEVANT COMPANY OF CANADA, LTD., GALT, ONT.; Sales Offices in Toronto and Montreal, and representatives in principal Canadian Cities.

HOW STURTEVANT ENGINEERING SERVICE CAN HELP YOU

As a leader in the manufacture of Air Handling Equipment, backed by more than 80 years' experience, B. F. STURTEVANT COMPANY is exceptionally qualified to help you attain the most efficient and economical solution of any air handling problem. Skilled Sturtevant Engineering experts, located in many leading cities are prepared to render the following 5-point service: (1) Analyze your problem. (2) Recommend the solution. (3) Specify the equipment. (4) Supervise the installation. (5) Check the operation.

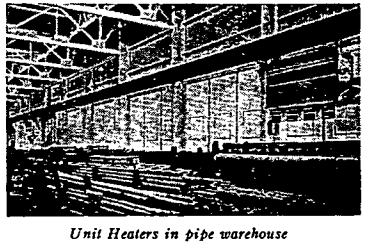
Whether requirements call for a single unit of apparatus or a complete engineered system, Sturtevant engineers can recommend the solution best suited to fulfill your individual needs. Do not hesitate to call the Sturtevant representative nearest you for assistance. Needless to say, no obligation is incurred.

As a preliminary aid, we list on the following two pages the major types of equipment manufactured by us, together with their general applications and the reference numbers of catalogs available.

INDUSTRIAL HEATING AND VENTILATING

Mr. Sturtevant invented and was granted U. S. Letters Patent No. 92,490 covering the first so-called hot blast industrial heating system on July 13th, 1869. Since that time Sturtevant has been responsible for many basic improvements in factory heating, moving forward with the trend of times to the modern and efficient hot blast central station heating system, the combined heating, ventilating and humidifying systems, and the unitary system which is clearly indicated for many installations, and which is now so popular. Sturtevant manufactures the component parts for all of these systems, including fans and blowers, heating surface, air washers, humidifiers, unit heaters and motors. This fact enables our engineers to make an unbiased selection and recommendation of the type of system and apparatus offering the most efficient and economical solution to any factory heating and ventilating problem.

Heating and Ventilating System in Machine Shop



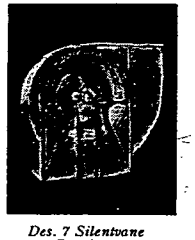
Unit Heaters in pipe warehouse

TABULAR VIEW OF STURTEVANT EQUIPMENT AND APPLICATIONS

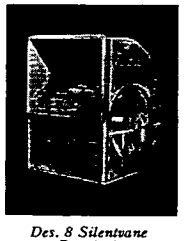
This table shows at a glance types of equipment manufactured by B. F. Sturtevant Company and their general application, together with catalog numbers on specific equipment. The numbers shown under "General Application," in the left column refer to the Index of Products. If no catalog number is given, write to B. F. STURTEVANT COMPANY, main office, or to the nearest branch office, stating specific requirements, and complete information will be sent immediately.

General Catalog No. 500 gives brief descriptions, Capacities, etc. of all Sturtevant Products

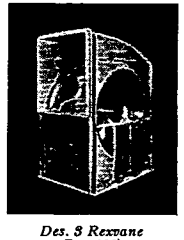
GENERAL APPLICATIONS		STURTEVANT EQUIPMENT INDEX	TRADE NAME OR DESIGN NO.	CAT. NO.
AIR CONDITIONING	1	Air Blenders	"Air Blenders"	
	2	Air Conditioning Apparatus	Sturtevant	425-2
1, 2, 3, 4, 5, 6, 14, 20, 25, 26, 28, 34, 38, 40, 42, 46, 52	3	Air Conditioning Systems, Industrial	Sturtevant	
	4	Air Conditioning, Railway	"Railvane"	401-1
	5	Air Washers	"Filterwasher"	453
	6	Blowers, Ventilating	"Rexvane Vent Sets"	400-9
	7	Blowers, Small Portable	"Big Midget"	F1266
DUST AND FUME REMOVAL	8	Blowers, Pressure	"Steel Pressure"	297-1
	9	Collecting and Conveying Systems	Sturtevant	291-2A
	10	Compressors, Centrifugal		
6, 7, 8, 9, 10, 11, 12, 13, 16, 20, 22, 26, 27, 40, 47, 48, 49, 50, 51		—vols. 35-75 c.f.m., 12-20 oz. press.	Design 7	408-2
	11	—vols. to 5,500 c.f.m., 1/2-3 lbs. press.	Design 9	386-1
	12	—vols. 35-650 c.f.m., press. to 5 lbs.	Design 1	431
	13	—vols. to 60,000 c.f.m., press. to 5 lbs.	Design 14	458
	14	Cooling Coils (Extended Surface) Water and Direct Expansion	Sturtevant	
HEATING	15	Drying Systems	Sturtevant	461
	16	Dust Collectors	Sturtevant	
6, 20, 25, 26, 28, 35, 36, 37, 38, 44, 45, 52	17	Exhausters (Material Handling)	"Planovane," Des. 3	410-3
	18	Exhausters, Slasher (Textile)	Sturtevant	432-1
	19	Fans—AV Propeller	"Atticvane"	400-9
	20	—Axial Flow, Pressure	"Axiflo"	444-1
	21	—Engine Cooling	Engine Cooler	418-5
INDUSTRIAL DRYING	22	—Fume and Dust Removal, Materials Handling	"Planovane," Des. 3	410-3
	23	—High Temperature		
8, 15, 20, 23, 25, 26, 27, 37, 38, 40	24	—Propeller, Motor Driven	Design 7	400-9
	25	—Low Speed, General Purpose	"Multivane," Des. 6	271-4
	26	—Medium Speed, General Purpose	"Rexvane," Des. 3	414-4
	27	—High Speed, Industrial	"Silentvane," Des. 7	449
	28	—High Speed, Heating and Ventilating	"Silentvane," Des. 8	457



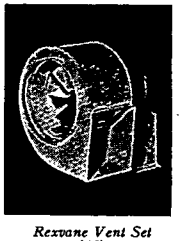
Des. 7 Silentvane Fan (27)



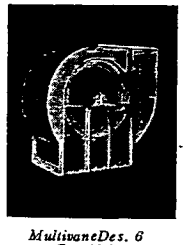
Des. 8 Silentvane Fan (28)



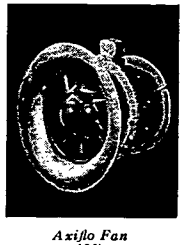
Des. 3 Rexvane Fan (26)



Rexvane Vent Set (46)



MultivaneDes. 6 Fan (25)



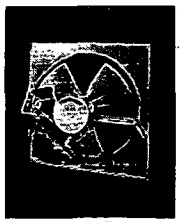
Axiflo Fan (20)

Planovane Des. 3 (22)

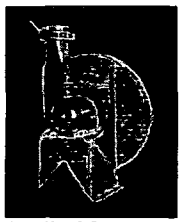
Propeller Fan Des. 7 (24)

General Catalog No. 500 gives brief description, Capacities, etc. of all Sturtevant Products

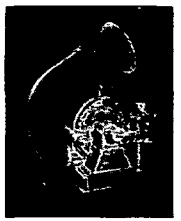
GENERAL APPLICATIONS		STURTEVANT EQUIPMENT INDEX	TRADE NAME OR DESIGN NO.	CAT. NO.
MECHANICAL DRAFT	29	—Mechanical Draft, Duplex type (Combined Forced and Induced)	Duplex	436
8, 11, 12, 13, 20, 27, 29, 30, 31, 32, 33, 40, 43, 52	30	—Mechanical Draft, Low Speed, Induced Draft, Abrasion-Resistant	S.P.I.D., Des. 2	447
	31	—Mechanical Draft, Medium Speed, Large Volume, Forced and Induced	M.V.M.D., Des. 6	409
PNEUMATIC CONVEYING	32	—Mechanical Draft, High Speed, Forced Draft	T.V.F.D., Des. 9	448
	33	—Mechanical Draft, High Speed, Induced Draft	T.V.I.D., Des. 2	445
8, 9, 10, 11, 12, 13, 17, 22, 39, 40	34	—Theatre Ventilating	Theatre Fans	424-1
	35	Heaters, Unit, Directional Flow Type for Wall and ceiling mounting	"Speed Heater"	396-9
PRIME MOVERS	36	Heaters, Unit, Downblast Type, for ceiling mounting	"Downblast"	454
40, 43	37	Heaters, Unit, Large Capacity, for floor, wall, ceiling mounting	"Multivane"	452
	38	Heating Coils (Extended surface)	Sturtevant	462
	39	Melt-Recovery Units (Welding)	Sturtevant	438-1
	40	Motors, Electric	Sturtevant	433-1
VACUUM CLEANING	41	Roof Ventilators	"Roofvane"	422
	42	Surface Dehumidifiers		
11, 12, 16, 40, 47, 48, 49, 50	43	Turbines, Steam, Helical Flow Type	Sturtevant	426
	44	Unit Ventilators, Schoolroom Type	Sturtevant	377-1
	45	Unit Ventilators, Auditorium Type	Sturtevant	S377-1
VENTILATING	46	Ventilating Sets, Direct Connected Motor	"Rexvane" Vent Sets	406
	47	Vacuum Cleaners, Portable	"Vortex"	413-2
6, 19, 20, 21, 24, 25, 26, 28, 40, 41, 44, 45, 46	48	Vacuum Cleaners, Portable Furnace Type	"Vortex"	373-6
	49	Vacuum Cleaning Systems, Commercial Buildings	Sturtevant	397-2
	50	Vacuum Cleaning Systems, Industrial	Sturtevant	368-3
	51	Vacuum Cleaner Attachments	Sturtevant	387-1
	52	Vane Control (Fan)	Sturtevant	446



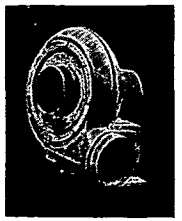
Allicvane Fan (19)



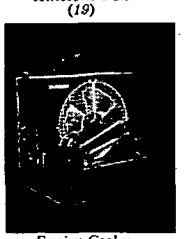
Centrifugal Compressor Des. 9 (11)



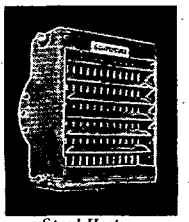
Centrifugal Compressor Des. 14 (15)



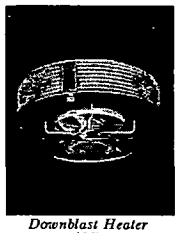
Pressure Blower (8)



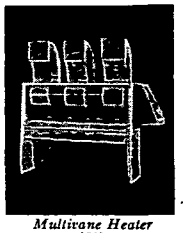
Engine Cooler (21)



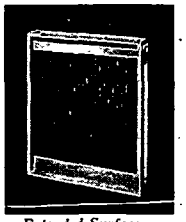
Speed Heater (35)



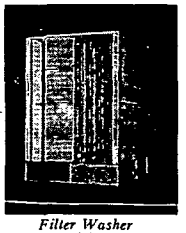
Downblast Heater (36)



Multivane Heater (37)

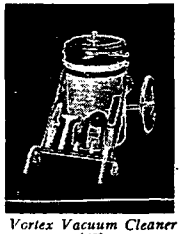


Extended Surface (38)



Filter Washer (5)

Electric Motor (40)

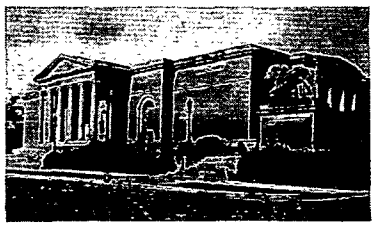


Vortex Vacuum Cleaner (47)

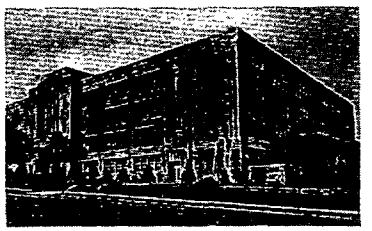
B. F. STURTEVANT COMPANY

PUBLIC BUILDING HEATING AND VENTILATING

Selection of a System: Many standards have been set up for the temperature required for comfort, and the fresh air required for proper ventilation and sanitation. No definite standard, however, can be set which will apply to and solve all heating and ventilating problems. The standard of ventilation for a given project is controlled largely by its location; more ventilation in congested areas; less ventilation in suburban districts. Other considerations are:—the purpose for which the building is used; the number of occupants in relation to floor space; and the amount of money available in the budget. *Apparatus*: The foundation of Sturtevant's business is heating and ventilating apparatus and Sturtevant shops produce practically all the mechanical equipment necessary for a modern and efficient heating and ventilating system including supply and exhaust fans, blast extended surface heaters, unit ventilators, air blenders, air washers and humidifiers, roof ventilators and electric motors.



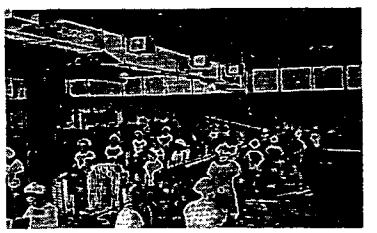
Museum of Art, Baltimore, Md.



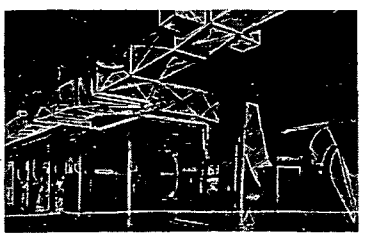
Werquahic H. S., Newark, N. J.

INDUSTRIAL AIR CONDITIONING

Air can be conditioned inside manufacturing areas, packaging rooms and storage vaults to provide the exact atmospheric conditions desired, so rendering producer and manufacturer independent of outside weather. This is accomplished either by means of a complete system embodying mechanical refrigeration or by evaporative cooling and humidifying. Air conditioning, employing mechanical cooling and dehumidifying, permits weather-sensitive products to be manufactured, packaged, and stored under ideal indoor conditions which can be properly regulated for the type of materials handled, and uniformly maintained regardless of outside weather. Such mechanical refrigeration gives absolute control of both temperature and relative humidity simultaneously. Evaporative Cooling is a less expensive method, and is particularly applicable in certain sections of the country where a wide spread exists between the inside dry bulb temperature and the outside wet bulb temperature.



Efficient production in candy plant



Quality controlled in dye plant

B. F. STURTEVANT COMPANY

Division of

Westinghouse Electric

Hyde Park, **Boston, Mass.**

OFFICES AND CONTRACTORS IN ALL PRINCIPAL CITIES

The compressor is, of course, the heart of the air conditioning system. In large measure, the users satisfaction rests upon its design, its construction, its size, its efficiency, its dependability. The following facts about Westinghouse Hermetically-Sealed Freon-12 Compressors are, accordingly, presented herewith for your information:

The basic design was established over 20 years ago. Thousands of successful installations have proved its dependability.

Shaft seals have been eliminated through the hermetically-sealed design. As a result, leaks through shaft seals, most common cause of refrigerant loss and system failure, are unknown with Westinghouse compressors.

Belts, pulleys and couplings do not exist in this compressor. These are items which require frequent adjustment and replacement. Their elimination greatly reduces maintenance time and expense.

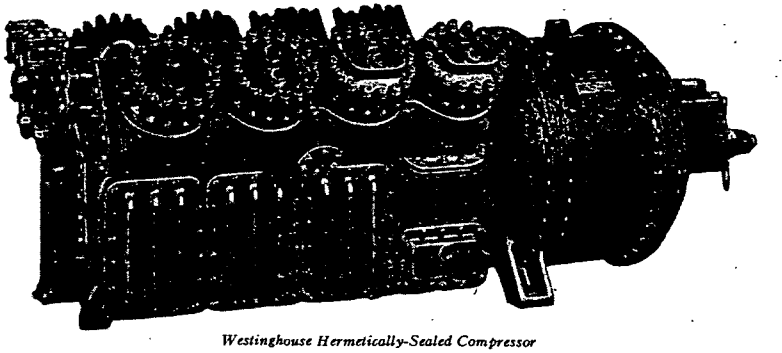
Direct drive conserves power. Power losses common to belt-driven machines are eliminated.

Sealed-in mechanism is protected from rust, dirt and corrosion. Pressure lubricated, it gives years of trouble-free service. Yet the entire mechanism is readily accessible for service when needed.

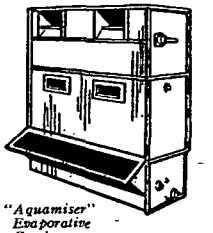
Compact size and light weight are important when planning equipment installation. These are inherent advantages of the hermetically-sealed design.

Flexible Operation is made possible by the application of a variable capacity accessory which loads and unloads one or more cylinders of the compressor as cooling requirements fluctuate.

Installation in unventilated spaces is practical with the Westinghouse Compressor because the driving motor is refrigerant-cooled. Thus otherwise waste space can be effectively used for equipment.



Westinghouse Hermetically-Sealed Compressor



"Aquamiser" Evaporative Condensers

"Aquamiser" Evaporative Condensers. Where water is expensive or its use restricted, the Aquamiser accomplishes effective condensing of the refrigerant at a saving in water consumption of as much as 95 per cent. Available in a wide range of sizes.

Air Handling Units. Two types are supplied—a horizontal type for ceiling suspension, and a vertical type for floor installation. Each embodies quiet blower-type fans, and provides facilities for the installation of required cooling or heating coils, filters and humidifier sprays.

Heat-Transfer Surfaces. Direct-expansion coils with patented refrigerant distributor; chilled water coils; steam and hot water coils are available in a great many sizes for all types of systems.

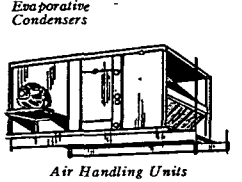
The Westinghouse Unitaire is a self-contained air conditioning plant incorporating the Hermetically-Sealed Compressor. Within-the-space and central-plant types in sizes from 2 to 25 hp.

Water-Cooled Condensers. Where sufficient water is available, the water-cooled condenser is economical and efficient. Sizes to match the Westinghouse Compressor for varying requirements.

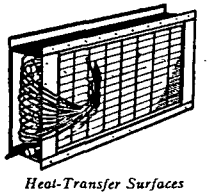
The Westinghouse Precipitron.* Where complete air cleanliness is desired, the PRECIPITRON offers the most effective possible method of obtaining it. Sizes fit every requirement.

*Trade-mark registered in U. S. A.

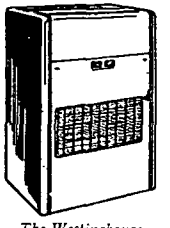
Water-Chilling Units. For the chilling of water for distribution to a number of cooling coils, and for use as process liquid coolers, these efficient water-chilling units range in size from 5 ton to 100 ton capacity.



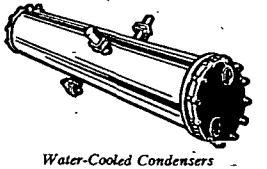
Air Handling Units



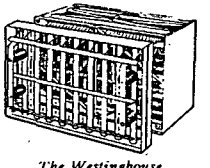
Heat-Transfer Surfaces



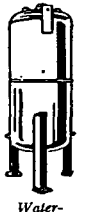
The Westinghouse Unitaire



Water-Cooled Condensers



*The Westinghouse Precipitron**



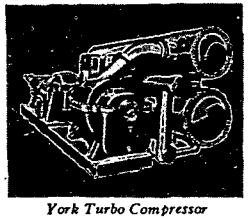
Water-Chilling Units

York Corporation

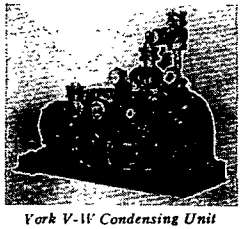
York, Pennsylvania

Factory Branches and Distributor Engineering and Sales Offices throughout the World.

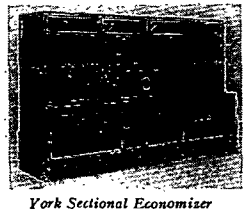
Air Conditioning and Refrigeration for maintaining proper atmospheric conditions for industrial processes or comfort requirements. Installations of unit and central systems in a complete range of capacities and types for every design requirement.



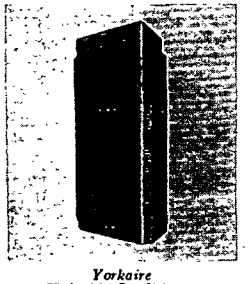
York Turbo Compressor



York V-W Condensing Unit



York Sectional Economizer



Yorkaire Unit Air Conditioner

Condensing and Water Cooling Systems—Centrifugal brine and water cooling systems available over wide range of capacities up to 1500 tons refrigeration, steam turbine or motor drive.

Self-contained dynamically balanced, non-vibrating V/W type reciprocating compressors available in capacities up to 350 tons refrigeration in a single unit, with water cooled or economizer type condensers.

Efficient automatic capacity reduction available for economical operation at reduced load.

The York Economizer—A combined forced-draft cooling tower and refrigerant condenser, is available for installations where prohibitive water costs or inadequate drainage facilities preclude the use of a water cooled condenser. Standard factory constructed and built-up units may be used singly or in multiple for applications of any specified capacity. Economizers for use with Freon as the refrigerant are furnished, as standard, with a liquid sub-cooling coil. Economizers also designed for cooling of quench oil and other liquid coolants.

Air Conditioning Units: A complete line of finned coil, dry coil, wetted surface and spray type sectional air conditioners for horizontal or vertical applications, designed to facilitate installation and the distribution of air. Standard units can be equipped with by-pass feature and arranged for cooling and dehumidifying, heating and humidifying, for year-round processing.

Yorkaire Unit Air Conditioner—A compact, self-contained model occupying but 21 x 42 inches of floor space and requiring only water, drain and electrical connections to operate. Special features provide utmost flexibility to meet varying conditions. Temperature dial control provides both automatic and manual temperature control. Air volume and motion may also be adjusted by a special control and the directional grille provides *directed air flow*—up, down or from side to side. May be used with ducts if desired.

Yorkaire Conditioners are ruggedly built, quiet in operation, equipped with standard fan and compressor motors for AC or DC.

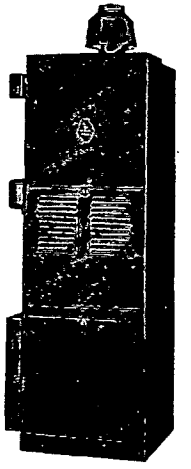
Dehumidifiers—For central station systems where a large volume of air is to be handled and where control of humidity is an essential requirement, the York dehumidifier is especially applicable. Construction features insure a minimum space demand and maximum performance conditions. Standard washers are available in a full range of capacities for industrial installation.

Let the pup be furnaceman

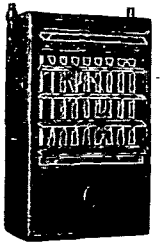
THE BRYANT HEATER COMPANY

17825 St. Clair Avenue - - - Cleveland, Ohio

Engineering, Sales and Installation information on Bryant Equipment available through Bryant Distributors, Dealers and Gas Companies in principal cities.



Vertical Winter Air Conditioner



Suspended Type Gas-Fired Unit Heater

Bryant Gas designed boilers include tubular cast iron sections, ribbed lower tubes, large steam liberating areas, all heating surfaces readily accessible for cleaning. Insulated metal jacketed covers and Bryant gas controls. Complete range of *AGA* inputs from 45,000 to 3,996,000 Btu/hr for steam and hot water heating systems, volume water heating and industrial process.

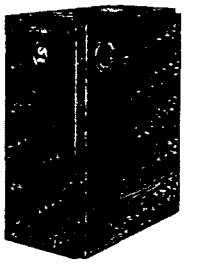
Bryant Vertical Winter Air Conditioners complete with blowers, humidifier and filters are compactly designed for small housing, office and industrial use. Bryant tubular cast iron section design and quality controls are standard equipment. Capacities range from 55,000 to 115,000 Btu/hr *AGA* inputs.

Complete line of forced Warm Air Gas-Fired equipment from 60,000 to 750,000 Btu/hr *AGA* inputs. Efficient cast iron heating sections of vertical tubular construction and large capacity blowers are featured. Humidifiers, filters and Bryant Automatic controls are standard equipment.

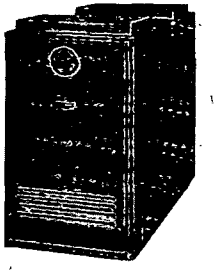
Bryant suspended type Gas-Fired Unit Heaters available in five sizes ranging from 65,000 to 255,000 Btu/hr *AGA* inputs. Efficient heat exchange of staggered vertical tube construction. Available in both cast iron combustion chamber, alloy steel tube and all steel types. Quick, clean, efficient heat for all types of industrial and commercial space. Flexible, automatic control and large volume air circulation produce ideal space heating results.

Bryant Dehumidifiers with rotary silica gel drum, is completely automatic in operation and finds application for exact humidity control in industrial processing, comfort air conditioning and the drying and storing of hygroscopic materials. They are available in three sizes, 800, 1300 and 2900 cfm. Standard units are reactivated by gas. Indirect units arranged for use with high pressure steam coils or electric strip heaters are available.

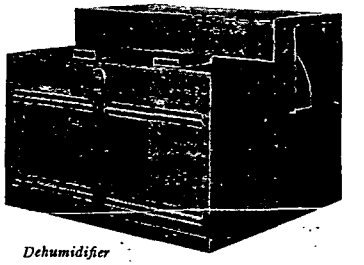
See your local Bryant Distributor or write for complete details and specifications.



Gas-Fired Boiler



Forced Warm Air Gas-Fired Equipment



Dehumidifier

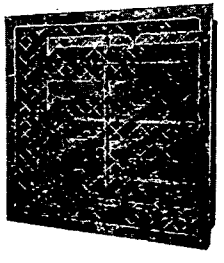
American Foundry and Furnace Co.

Sales Offices in Principal Cities

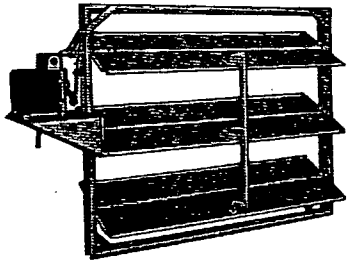
P. O. Box 904, **Bloomington, Ill.**

AIR CONTROLLING SPECIALTIES

KD Grille and Fire Damper



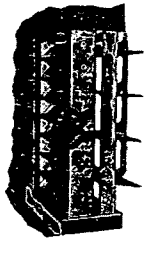
At the right is shown a Type KD Grille and Fire Damper consisting of a wire grille, with an angle iron frame and eight inch steel sleeve, and a ball bearing louver damper, with pull chain and two fusible links. The steel sleeve is made to fit the unit into a wall opening. The chain operator works through two pulleys and is held in place by a claw on the damper frame. The fusible links melt at 160 deg., thus allowing the damper to close by its own weight in case of fire.



F-12 Louver Damper

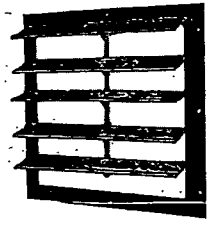
At the left is shown a Type F-12 Louver Damper. These dampers are made on order to fit any size opening. Blades are 16 gauge steel, frames are 2 in. x ½ in. x ⅛ in. channel. Ball bearings are standard, with steel trunnion bearings optional. Motor, linkage and bracket shown are not standard but may be ordered as an extra.

S-454-F Radio Range Stormproof Louver Combination



Consists of a galvanized iron frame with 26-gauge galvanized iron stationary horizontal stormproof louver blades, riveted securely to outside frame and all built into outside wall (fits 8-in. thick wall). Apron extends over sill. Back of stormproof louver is a No. 16 mesh screen.

Back of screen is a multiple-blade, ball bearing louver damper (similar to F-12 but with off-center axle) to control volume of air admitted. Louver damper blades of 16-gauge steel, galvanized. Frame of 2 x ½ x ⅛-in. galvanized channel iron. Dampers can be automatically or manually controlled. Blades all work in unison.

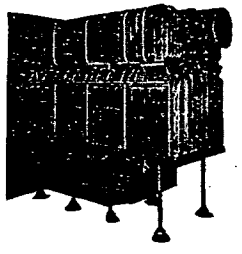


FL-Aluminum Automatic Fan Outlet Louver

These louvers are for use where the air from a fan discharges into atmosphere and the velocity of the discharged air is less than 2000 fpm. Aluminum blades, working in unison, open from fan pressure and close automatically when fan is not in operation. Frames of angle iron drilled for fastening to wall, duct or penthouse. Standard finish of frame—black or gray enamel or prime coat. Regularly made in 14 sizes from 12 x 12 in. to 50 x 50 in. *Special sizes on request.*

HEAVY DUTY HORIZONTAL HEATERS

CENTRAL PLANT SYSTEMS

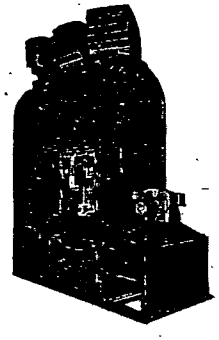


American SUPERFIN Heaters are made entirely of cast iron and designed for use with blower, air filters, humidifier, etc., in Central Plant Winter Air Conditioning Systems.

Fireboxes and top arches are finned heavily to provide maximum durability and furnish largest possible heating surfaces. Design is such that all air is forced over hottest surfaces of the heaters at all times.

Finned, pear-shaped radiators, streamlined, add enormously to the radiating surface of the heaters, thus adding to their efficiency.

Capacities range from 600,000 to 1,300,000 Btu per single unit.



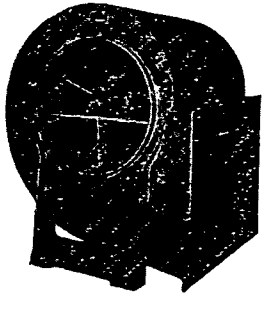
UNIT HEATER SYSTEMS

American Gas or Oil Fired Unit Heaters have been designed for use in commercial and industrial applications. These units have the multiple blowers underneath the heating section. Air is drawn into the heater at the floor line, then passes over the heater, and may either re-enter the room from the top of the heater or at the floor line on the opposite side of the heater.

Heater section is made entirely of cast iron and is designed to use either oil or gas as a fuel.

Single units have from 750,000 to 3,500,000 Btu at the outlets.

SUPERIOR BLOWERS



At the right is shown a standard Superior Blower. Superior blowers are all that their name implies. Rugged design, quiet operation of mechanical parts, freedom from vibration, have been stressed throughout the construction of the entire line. Made in wheel sizes ranging from 10 to 65 in., inclusive, having wheel diameters of corresponding values at 5-in. intervals, with 12 and 18-in. sizes in addition.

Each size blower is available in single or double width, single or double inlet, any discharge arrangement. Deliveries range from 800 to 105,000 c.f.m., with uniform capacity ranges between.

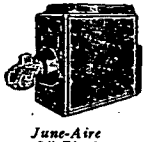
DOMESTIC HEATING EQUIPMENT

June-Aire Gas Fired

Made in sizes 72,000 to 416,000 Btu at the bonnet.

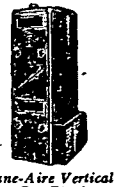
June-Aire Oil Fired

Made in sizes 130,000 to 335,000 Btu at the bonnet.



June-Aire Oil Fired

Capacity of 115,000 Btu at bonnet.



June-Aire Vertical Gas Fired

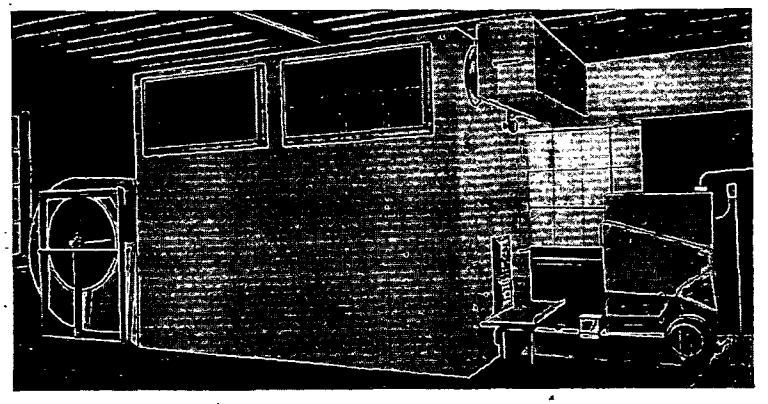
Capacity of 60,000 Btu at bonnet.

Campbell Heating Company

31st and Dean, **Des Moines, Iowa**

SUMMER and WINTER AIR CONDITIONING

Industrial, Commercial—Institutions, Residences



Campbell Heater in factory of National Mfg. & Stamping Co., Des Moines, Iowa

The Campbell Heater shown above is installed in a two-story factory building 80 x 120 ft. with 12 ft. ceiling height on each floor. There are no ducts—this heater sends heat to all parts of the building without ducts. On a zero day thermometers near four corners of the first floor varied only from 59 to 62 deg.—the north west corner was 60 deg. When warm air is desired to remote rooms it can be provided by a properly designed duct system.

This heating performance was accomplished in spite of the fact that the heater is in the south east corner of the building. Heated air traveling to the north end of the room must overcome the obstruction to air flow caused by the timbers above—the air must flow across the timbers, not with them.

In this installation the heater rests upon the floor of the room to be heated. Where floor space is limited and valuable the blower can be mounted on top of the heater or put under it by raising the heater, provided the ceiling height is sufficient.

Five Advantages of Campbell Heaters

1. **Easy to Install**—In most cases the Campbell Heater can be put into operation within 72 hours after delivery.
2. **Especially Adaptable to Ductless Convection Heating**—Elimination of ducts reduces cost in heating a single large room.
3. **Compact in Design**—Over-all dimensions can be adapted to meet any reasonable limitations of floor space or ceiling height.
4. **Efficient with any Type of Fuel**—Especially adaptable to stoker firing, but operates equally well with fuel oil or gas.
5. **Low in Cost**—Because of standardized design and efficient manufacturing methods, Campbell Heaters are low in cost.

Campbell Heaters are guaranteed to deliver full rated capacity.

CAMPBELL "WINTER-CHASER" AIR CONDITIONING SYSTEM

The Campbell "Winter-Chaser" System provides all the essentials of winter air conditioning: Simultaneous control of temperature, humidity, air circulation and air cleanliness, besides providing fresh air for ventilation, quick heating, flexibility; and a summer cooling effect. Campbell equipment is built of the best materials obtainable, and has been developed through over sixty years of experience. The system is designed by competent experienced engineers and installed by experienced mechanics. It is guaranteed as to results and for 10 years as to durability.

Advantages of Campbell "Winter-Chaser" Design

Compact design, requiring minimum space—the heater can be placed readily in any ordinary boiler or furnace room.

Efficient radiating surface—permits an immense amount of radiating surface in a small space. Ratio of prime radiating surface to grate area is more than 30 to 1. This produces very low flue gas temperature with consequent high efficiency.

Vertical self-cleaning radiating surfaces constitute most of the heating surface—hence are not insulated with soot and ashes.

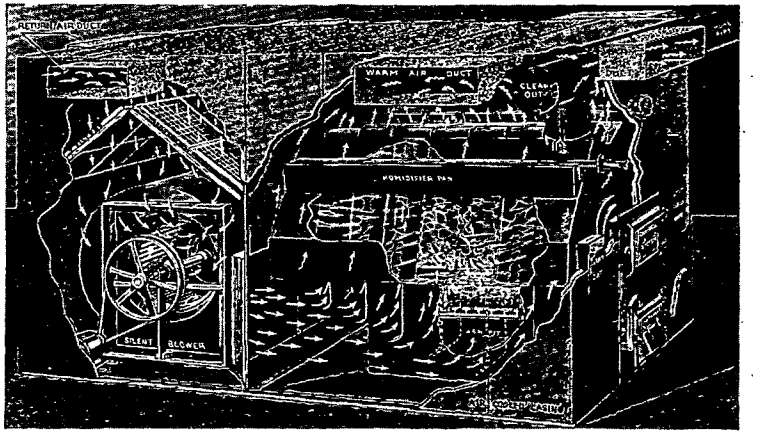
Vertical air-warming tubes completely surround the fire, subjecting greater prime heating surface to direct contact with the fire—less liable to be overheated.

Extra large combustion space above grate permits smoke and air to mix and burn before striking heating surface—less unburned gases escape up the chimney.

Long return smoke travel—the tubes form a dividing wall each side of the fire, separated in the back to let smoke pass through behind the tubes and to the front of heater. This assures long contact with heating surfaces.

Two convenient cleanouts permit easy cleaning of the smoke passages without putting out the fire.

Basement heating on the same level as the heater, is solved by this Campbell System. Entire capacity of heater can be forced into the basement and thus heat a large room in a few minutes.

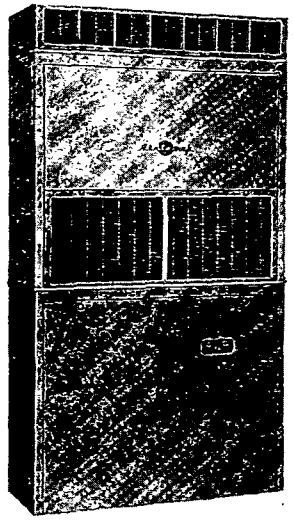


Furnace Number	Sq. Ft. Grate	Heating Surface	For Building Heat Loss Btu	Maximum Capacity Btu	Blower CFM	Size Motor	Dimensions Casing	Addnl. Space for Blower	Approx. Shipping Weight
8075	7½	280	483,000	725,000	8850	¾	75 x 80— 88" high	54"	4500
8100	10	320	596,000	893,000	10900	1	75 x 93— 96" "	60"	5500
8125	12½	360	720,000	1,080,000	13200	1½	75 x 105— 96" "	66"	6500
8150	15	440	850,000	1,275,000	15600	2	75 x 118—102" "	66"	7500
8175	17½	480	960,000	1,440,000	17500	2	75 x 130—102" "	72"	9000
8200	20	600	1,150,000	1,725,000	21100	2	94 x 137—120" "	76"	10000
8250	25	750	1,440,000	2,160,000	26400	3	94 x 157—120" "	76"	12000

Unit includes furnace, casing, blower, motor, V-flat drive.

CHRYSLER AIRTEMP

AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON, OHIO



"PACKAGED" AIR CONDITIONERS

AVAILABLE IN 3 and 5 H. P.

COMPLETE—Cools, dehumidifies, filters and circulates the air. Free air discharge or duct distribution. Heating coil for year 'round service, optional.

COMPACT—Everything enclosed in a rust-resistant "Bonderized" cabinet of modern design with chrome hardware and trim. Occupies only 4.7 sq ft of floor space.

EASILY, QUICKLY INSTALLED—Tested and completely assembled at the factory. Needs only three connections, electricity, water and drain.

SEALED RADIAL COMPRESSOR—All moving parts are balanced for quiet operation without vibration. Compressor assembly suspended from a single rubber mounting.

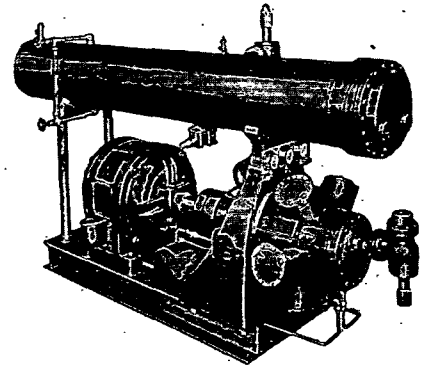
ECONOMICAL OPERATION — Full pressure lubrication and "Superfinishing" of vital moving parts assures long life with efficient operation.

FLEXIBLE—Airtemp "Packaged" Air Conditioners can be installed singly or in multiple to meet almost any requirements. Can easily be moved at any time.

CHRYSLER AIRTEMP RADIAL CONDENSING UNITS

AVAILABLE IN 10 TO 75 HORSEPOWER CAPACITIES

This heavy-duty Radial Condensing Unit for use with Freon is especially adapted for refrigeration, for industrial processes or air conditioning. Airtemp radial compressors are directly connected and have force-feed lubrication. The automatic capacity-reduction device gives high operating efficiency. Light in weight and economical to operate, these compressors are shipped ready to run. They are especially easy to install since vibration is practically eliminated and no special foundations are necessary.



AUTOMATIC UNLOADER

The automatic cylinder unloading device permits starting the compressor under no load and keeps the compressor automatically adjusted to varying loads with no stopping and starting during operation.

- AUTOMATIC CAPACITY REGULATION
- UNLOADED STARTING
- DIRECT CONNECTED
- SIMPLIFIED INSTALLATION
- COMPACT DESIGN
- PRACTICALLY NO VIBRATION
- NO SPECIAL FOUNDATIONS NEEDED
- INTERCHANGEABLE PARTS

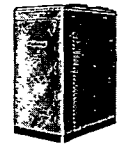
CHRYSLER AIRTEMP

VAPORIZING OIL-BURNING AUTOMATIC FURNACE

Models for forced-air—52,000 Btu output—gravity—45,000 Btu output. Sure-Draft fan assures highest overall efficiency. Bonderized and insulated jacket. Approved for closet installation, Underwriters' Laboratories, Inc.

OIL-FIRED AUTOMATIC FURNACE

Heats, humidifies, filters and circulates the air. Five models, from 70,000 to 160,000 Btu output. "Bonderized" and insulated jacket. Metal combustion chamber, seam-welded firebox of copper-bearing steel; large, slow-speed, rubber-mounted fan. Airtemp conversion oil burners on all models.



GAS-FIRED AUTOMATIC FURNACE

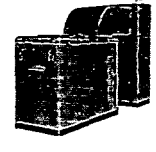
Heats, humidifies, filters and circulates the air. Steel models from 70,000 to 160,000 Btu output. Cast-iron models for 50,000 and 75,000 Btu output. "Bonderized" and insulated jacket. The Airtemp "Silent Flame" Gas Burner starts, stops and operates quietly, has many exclusive features—no popping or flash-backs. Approved, *A.G.A.* Laboratories.

PERCOLATOR BOILERS—OIL OR GAS-FIRED

Two oil-fired models 400 and 600 EDR (steam). Two gas-fired models 400 and 600 EDR (steam). Fire chamber surrounded with water on all sides and bottom. High efficiency and faster heating result from "percolator" principle. Steam or hot water. "Bonderized" and insulated jacket.

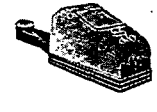
COAL FIRED FURNACES, FORCED-AIR GRAVITY

Forced-air models, steel. Sizes from 105,000 to 130,000 Btu at Bonnet. Oversized blower motor mounted in rubber, has automatic overload and low-voltage protection. Gravity models, steel—sizes from 70,000 to 90,000 Btu at register.



COMBINATION HEATING AND COOLING FOR THE HOME

Combination of a 3 hp Chrysler Airtemp "Packaged" Air Conditioner and any of the larger Chrysler Airtemp automatic furnaces. The same blower, filters and ducts of the automatic heating system are employed for cooling in the summer.



STOKERS

Domestic, available in both hopper and bin-feed models for anthracite or bituminous coal for burning 15 to 60 lb per hour. Chrysler Airtemp Stokers have the famous hydraulic transmission.

MODEL B-10 AND C-10—OIL BURNERS

A conventional pressure-atomizing oil burner, 1.35 to 4.5 gallons No. 3 furnace oil per hour.

All Airtemp Oil Burners are approved by Underwriters' Laboratories, Inc., and bear the seal of the Official Inspection Agency of the Oil Burner Industry as evidencing compliance with commercial Standard CS75-39, as issued by the National Bureau of Standards of the U. S. Department of Commerce.

Lee Engineering Company

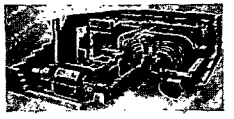
Union National Bank Bldg., **Youngstown, Ohio**

LEE DIRECT WARM AIR HEATING

The Lee System of warm air heating generally costs less to install than steam or hot water; utilizes fuel with a high degree of efficiency; distributes the heat exactly where needed; responds promptly without lag; requires little or no maintenance; and needs no licensed attendant. Heaters for use with the Lee System are made in the four types illustrated and described briefly below.

BRICK-SET TUBULAR HEATER

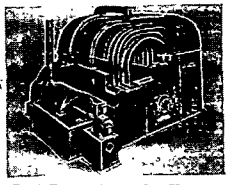
For use with central heating system in connection with duct distribution. Single heater capacities from 2,800,000 Btu per hour to 8,000,000 Btu per hour. Two heaters, installed as a battery serving as one unit, provide capacities over 10,000,000 Btu per hour.



Brick Set Tubular Heater

STEEL ENCASED TUBULAR HEATER

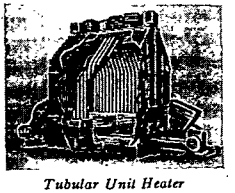
For use with central heating systems in connection with duct distribution over a capacity range of from 2,000,000 Btu per hour to 6,000,000 Btu per hour. Heater may be installed in heated area without enclosure, requires no foundation, and may be moved from one location to another by taking unit apart and reassembling.



Steel Encased Tubular Heater

TUBULAR UNIT HEATER

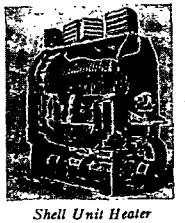
For use either as a central system in connection with duct distribution or with adjustable outlet nozzles as a unit heater. Capacity range from 2,000,000 Btu to 6,000,000 Btu per hour. In sizes up to 4,000,000 Btu heater is shipped as a completely assembled unit with all but mechanical equipment, refractory lining and controls in place. Heaters require no foundation and are equipped with crane hooks so that they may be moved from one location to another.



Tubular Unit Heater

SHELL UNIT HEATER

For use with or without distributing duct system. Heaters have capacity range of from 400,000 Btu to 2,000,000 Btu per hour. Available for either stoker or hand firing. All units are shipped completely assembled, wired and ready for operation. Stoker-fired units are shipped assembled with all parts in place except mechanical equipment, refractory lining and controls. Available in both hand and stoker fired models.



Shell Unit Heater

For further information write for Catalog HV-44.

THE MEYER FURNACE COMPANY

PEORIA, ILLINOIS

Manufacturers of Heating and Air Conditioning Equipment for Coal, Gas and Oil Burning

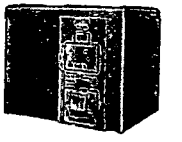
Branches and Distributors

ALBUQUERQUE, N. M.
CHICAGO, ILL.
COLUMBUS, O.
DES MOINES, IA.
FLORENCE, S. C.
GREEN BAY, WIS.
KANSAS CITY, MO.
NEW YORK, N. Y.
OMAHA, NEBR.
PHILADELPHIA, PA.
PITTSBURGH, PA.

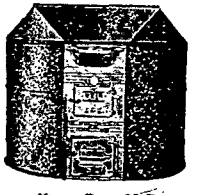
WEIR and MEYER Steel Warm-Air Furnaces, of welded-and-riveted gas-tight construction, have a 60 year reputation for efficiency, dependability and durability. Service in hundreds of military establishments both here and abroad has again attested to their satisfactory performance. They are available for small and large requirements and for all fuels in a wide variety of firing applications.

U Series Gravity

The U SERIES WEIR Hand-Fired Coal Furnace embodies a new construction principle (patent applied for) which, when combined with its other time-tested Weir features, provides an outstanding heater. The gravity furnace, as shown on the left, ranges in register capacity from 50,000 to 170,000 Btu per hour. The rectangular-cased forced-air furnace, shown on the right, ranges from 50,000 to 250,000 Btu per hour output at the register. This series may be stoker-fired, though other Weir stoker-designed furnaces are available.

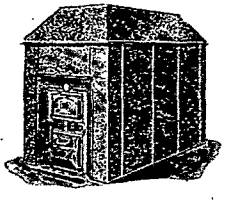


U Series F-A

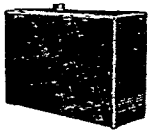


Heavy Duty 88-R

WEIR Heavy-Duty Furnaces, consisting of the R Series on the left and the 500 Series on the right, are ideal for industrial and commercial service and for schools, churches and other large spaces. Capacities from 300,000 to 1,500,000 Btu per hour. Designed only for forced-air circulation. Suitable for firing with coal by hand or stoker, with gas or with oil.



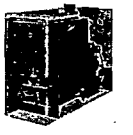
Heavy Duty 644B



Oil Fired A-100-M

The WEIR Oil-Fired Air Conditioner does a complete job of winter air conditioning. Designed for oil fuel and forced circulation. Completely self-contained in low compact casing enclosing burner and blower as well as heater and all controls, yet with everything easily accessible.

The MEYER Gas-Fired Air Conditioner automatically provides completely controlled winter air conditioning. Efficient performance, compact design, modern appearance. Heavy gauge welded steel heating element; die-formed casing. *A.G.A.* approved.



Gas Fired F-10

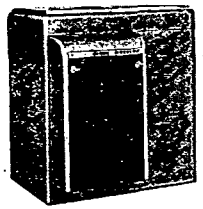
For any warm air heating or drying problem employing the use of any fuel, call our nearest office or representative or write to the home office.

L. J. Mueller Furnace Co.

ESTABLISHED 1857

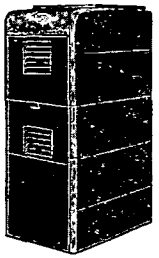
2009 W. Oklahoma Ave., **Milwaukee 7, Wis.**

Mueller Climatrol Gas-Fired Equipment



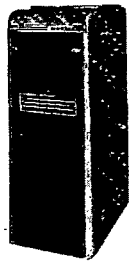
Series "EPS"

This winter air conditioner is available in three sizes, with *A.G.A.* input ratings from 90,000 to 180,000 Btu. Each size is available with various size blowers in accordance with individual air delivery requirements.



Series "SHP"

Winter air conditioner. May be installed in basement or utility room. Available in three sizes, with *A.G.A.* input ratings from 60,000 to 100,000 Btu per hour. Provides wide range of air deliveries to meet specific needs.

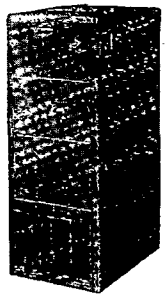


Series "CVP"

An all-cast-iron winter air conditioner for basement or utility room installation. Furnished in one size, with *A.G.A.* input rating of 100,000 Btu per hour. Adjustable motor sheave permits variation in air deliveries.

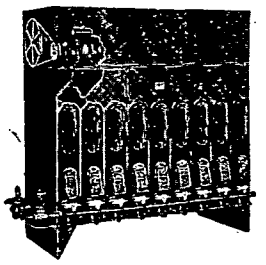
Series "G"

A gas-fired gravity unit for home applications. Made in two casing styles, i.e., square, as shown, and round. Furnished in two sizes, with *A.G.A.* input ratings of 90,000 and 135,000 Btu per hour.



Gas Boiler

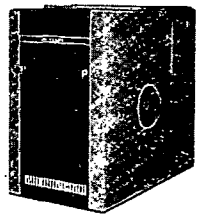
Type "10" (shown) and "11" have *A.G.A.* ratings from 180 to 1,260 sq. ft. steam, and 290 to 2,015 sq. ft. hot water. Type "20" boilers from 630 to 6,300 sq. ft. steam, 1,010 to 10,080 sq. ft. hot water.



Unit Heater

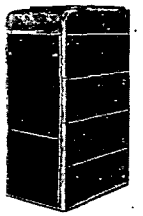
For large space-heating requirements, and industrial applications. Units are available in 45 sizes, *A.G.A.* approved, from a 4-section unit with input of 180,000 Btu per hour, to the 48-section size, with input of 2,160,000 Btu per hour.

Mueller Climatrol Oil-Fired and Coal-Fired Units



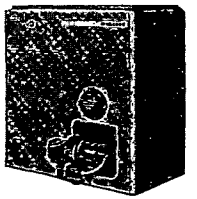
Series "50"

Oil-fired winter air conditioner. Available with Mueller pressure atomizing, or vaporizing burner. Three sizes, 100,000 to 225,000 Btu.



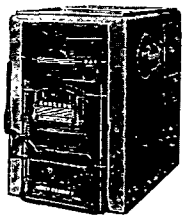
Series "OVP"

Oil-fired winter air conditioner. Equipped with Mueller Vaporizing Oil Burner. One size, with maximum capacity of 80,000 Btu at bonnet.



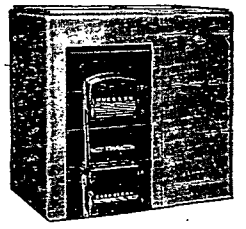
Series "OHP"

Horizontal type oil-fired winter air conditioner. Equipped with Mueller vaporizing burner. One size, 80,000 maximum at bonnet.



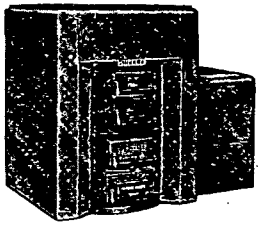
Series "FB"

Cast-iron, coal-fired winter air conditioner. Six sizes, with capacities at register from 69,000 to 199,000 Btu.



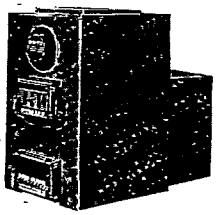
Series "P-400"

Steel, coal-fired winter air conditioner. 4 sizes, 20 in. to 27 in. drums, with capacities at register ranging from 79,000 to 141,500 Btu per hour.



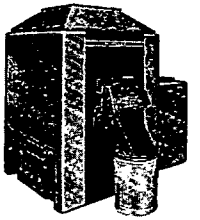
Series "AP"

Cast-iron, coal-fired winter air conditioner. Six sizes, 20 in. to 33 in. firepots, with forced air capacities ranging from 93,000 to 269,000 Btu.



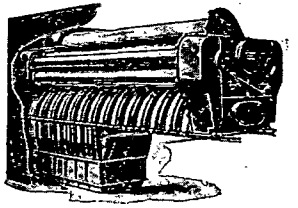
Series "WR"

Cast-iron, coal-fired furnace. Two styles, i.e., forced air and gravity. One size only. Forced air capacity, 72,000 Btu. Gravity, 42,000 Btu.



Stoker Furnace

Winter air conditioner. Any stoker may be used. Cast-iron heating unit. Available in two sizes, with capacities of 110,000 and 175,000 Btu.



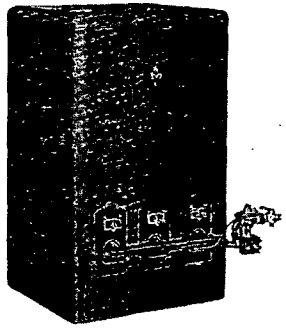
Horizontal Tubular

Adaptable for schools, churches and other large buildings. Three sizes, with capacity range from 1,188,000 to 1,390,000 Btu per hour. For gravity or forced air applications.

PAYNE FURNACE COMPANY
(One of the Dresser Industries)
BEVERLY HILLS, CALIFORNIA

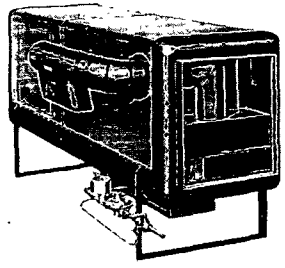
PAYNEHEAT
OVER 30 YEARS OF LEADERSHIP

GAS FUEL (HAS) EVERYTHING



MODEL "AS" UNIT-TYPE WARM AIR FURNACE

Gravity-type circulating unit for space-saving basement installation. Can be used as single unit or in multiples for zoned heating. Six standard sizes; *A.G.A.* input rating, 30,000 to 180,000 Btu's. Grey, metallized enamel finish.

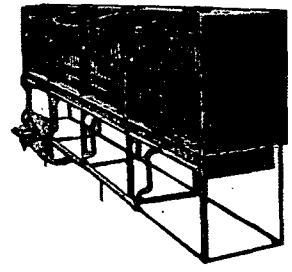


"SPACESAVER" UNIT

Horizontal, forced air heating system for attic, subfloor or ceiling installation in basementless homes, shops or industrial plants. All-welded heating element of heavy guage, die-formed steel. Two sizes: *A.G.A.* input rating, 75,000 to 100,000 Btu's.

PAYNE FURNACE COMPANY has always specialized in *gas-fired* furnaces. Eight salient points characterize the complete PAYNE "line":

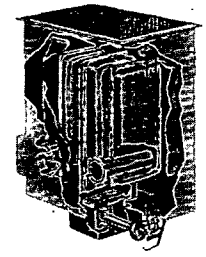
1. 69 *A.G.A.*-approved styles and sizes.
2. Each unit fully vented.
3. Can use natural, manufactured or LP gas.
4. Adaptable to "Zone-Conditioning" (control by zones or rooms).
5. First-class materials; sound, advanced engineering; rugged construction; long life.
6. Compact, light-weight, attractive, modern designs.
7. Ample filter area in forced air models assures thorough air-cleansing.
8. No gas leakage.



PAYNE "CB" COMMERCIAL BLOWER UNIT

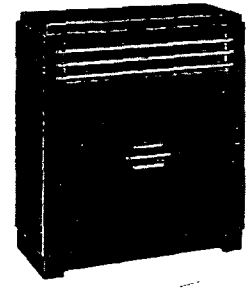
Compactly built commercial blower furnace for theatres, buildings and industrial plants, providing large volume of air at high velocity, with minimum fuel consumption. One-piece combustion chamber and radiator are of heavy gauge, die-formed steel, all-welded construction, preventing gas escape; ideal for high-pressure system. Entire radiation surface is exposed to air-currents traveling at high velocity with rapid heat transfer from element to air stream. "CB" Furnaces may be installed individually or in batteries to provide any capacity up to 1,680,000 Btu input. Shipped completely assembled except for supports and controls. *A.G.A.*-approved with limit control, current-breaking type, operating on current failure valve and automatic pilot.

PAYNEHEAT
OVER 30 YEARS OF LEADERSHIP



PAYNE FLOOR FURNACE

Heavy, cold-rolled, rounded edge steel grille, with wide margin for close floor-fit. (*See other details at right, above.*)

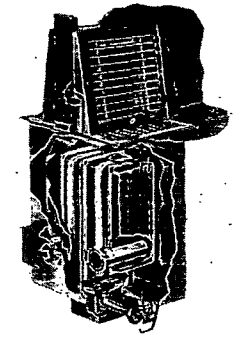


CONSOLE HEATER

Attractively finished to harmonize with any furnishings or decorative scheme.

FLOOR FURNACE (left) made in eight *A.G.A.*-approved sizes, shipped completely assembled. Deep-drawing steel heating element, heavy gauge for both combustion chambers and radiators. Single burner is first-quality casting, precision-machined with venturi tube and adjustable primary air shutter. Inner casing is suspended to allow for contraction and expansion. *A.G.A.* Btu input ratings: 15,000 to 75,000.

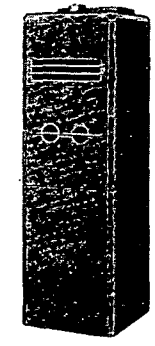
PAYNE DUPLEX REGISTER FURNACE (right) includes all Floor Furnace features, plus scientifically-designed register head for heating two rooms or suites from both sides of wall. *A.G.A.* input ratings: 25,000 to 50,000 Btu's.



DUPLEX FURNACE

PAYNE CONSOLE HEATER (left) Fully-vented, *A.G.A.*-approved heater for use with natural, manufactured or LP gas. Heavy-gauge heating element with special inner baffles for longer fire-travel and uniform heat distribution. Insulated air passages control cabinet temperature; undershield protects floor. Six sizes: Btu input rating, 15,000 to 60,000.

SENTRY FORCED AIR UNIT (right)—Compact forced air unit for basementless installations. Fits in any nook or corner. All-welded heating element of deep-drawing steel. All moving parts mounted on rubber blocks. Four sizes with Btu input rating, 60,000 to 160,000. Designed for up-discharge only. (Inquire about the ZONEAIR Forced Air Unit for basement installations.)



PAYNE "SENTRY" UNIT

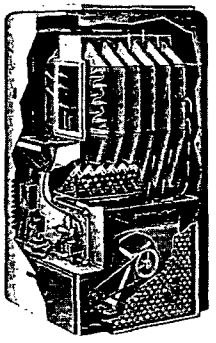
PAYNE "AA" VENT—Inner tube of aluminum is surrounded by outer galvanized casing with controlled air space between. Patented. Prevents condensation of water vapor in products of combustion as well as over-heating of outside casing. Quickly installed. Sizes: 3 in. to 6 in.

PAYNE ZONE-CONDITIONING—Time-tested PAYNE "Unit" heating, using two or more units, controlled by zones or individual rooms. Many advantages over "central" heating. Write for special booklet.

IMPORTANT—Complete catalog sheets and specifications on these and other PAYNE Furnaces available on request. Please specify type, heating value and specific gravity of gas.

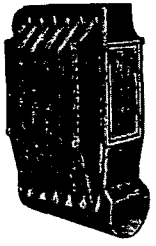
Gar Wood Industries, Inc.

HEATING DIVISION • 7924 Riopelle Street • **Detroit 11, Michigan**

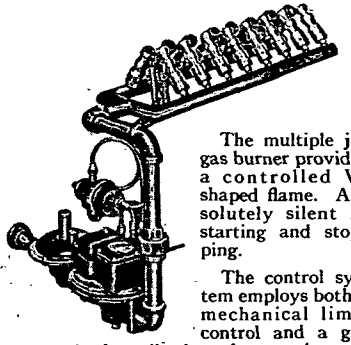


GAS-FIRED TEMPERED-AIRE UNIT

The heart of this efficient heating unit is the cast iron heat exchanger. Combustion takes place in the large firebox which has sufficient volume to prevent flame impingement. From the firebox, the products of combustion pass upward, diverge, and enter the triangular-shaped vertical channelways which contain cast-in baffles to direct the gases laterally against the side walls in their upward course.



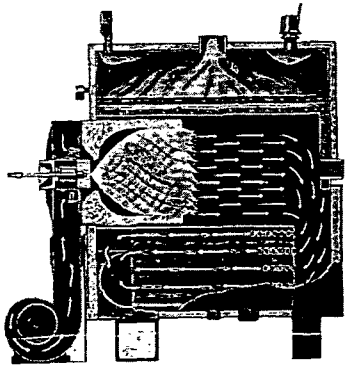
Durable cast iron was chosen for the construction of the heat exchanger after consultations with Utility Engineers. The corrugated firebox and overall design provide an intimate contact for gases and air resulting in a high rate of heat transfer.



The multiple jet gas burner provides a controlled V-shaped flame. Absolutely silent in starting and stopping.

The control system employs both a mechanical limit control and a gas actuated safety pilot for safe manual operation of the gas valve in the event of a current outage.

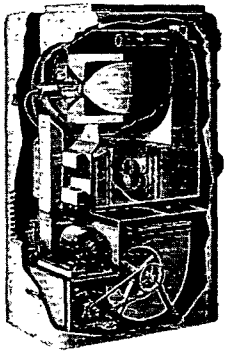
SERIES "M" OIL-FIRED BOILER BURNER UNITS



The Gar Wood Series "M" (steam or water) boiler burner units are built to fill the requirements of radiation systems of either the conventional or the new radiant type, or for use with indirect air conditioning systems. They are of welded steel construction with horizontal fire tubes, have downdraft gas travel and are internally fired with the Gar Wood type "O" burner which is an integral part of the boiler unit. Models range from 200,000 to 1,000,000 Btu/hr nozzle output.

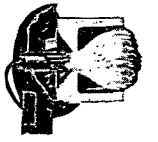
Gar Wood Industries, Inc.

HEATING DIVISION • 7924 Riopelle Street • Detroit 11, Michigan

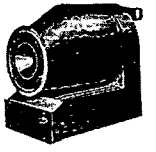


OIL-FIRED TEMPERED-AIRE UNIT

A high efficiency, direct fired, complete heating unit designed for home installation. The primary transfer of heat occurs in the large firebox which has its outlet at the bottom through which the hot gases pass into the economizer. Here the gases divide into long, thin slices within tubes. Each tube is swept at high velocity by cold return air which absorbs a maximum of heat from the economizer.



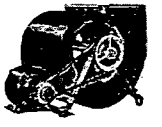
Integral burner and combustion chamber intermingles and rotates oil and air for perfect combustion.



Integrally built firebox-economizer unit. The teardrop shape removes the peak from the high temperature zone eliminating hot spots.

All parts of the power unit are die formed and welded into a single unit. Rubber mounted.

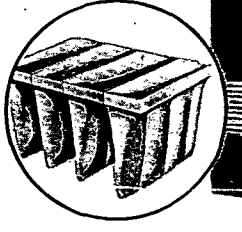
Quick-acting flash humidifier provides adequate humidity under short firing cycle conditions.

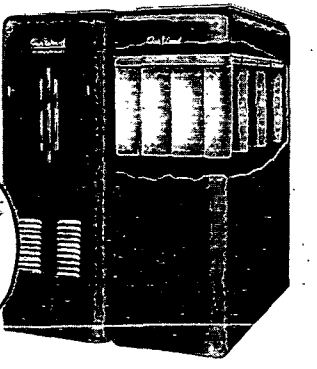


Large air blower delivers a generous quantity of air against the friction of duct systems.

DELUXE TEMPERED-AIRE WITH REMOVABLE, WASHABLE CLOTH FILTERS

Consisting of a series of bags similar to vacuum cleaner sacks, the filters are made of a special woven fabric and have a tremendous filtering area. These collector screens may be lifted out and cleaned in the washing machine. They can be removed and put back in a few minutes by lifting and replacing the retaining frames. DELUXE Tempered-Aire models, either oil or gas fired, are double the width of standard vertical models.

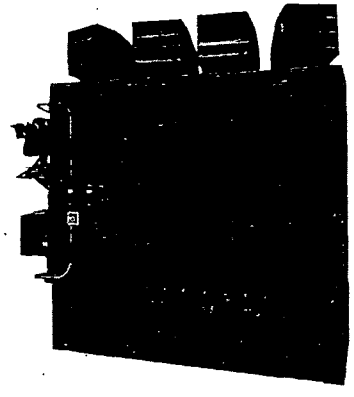




Airtherm Manufacturing Company

728 S. Spring Ave.
St. Louis 10, Mo.

THE ENGINEERED LINE OF UNIT HEATERS



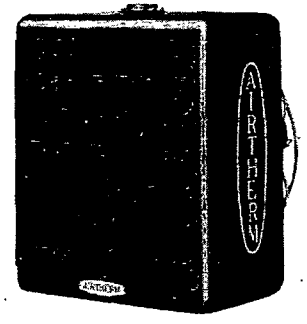
The New AIRTHERM
Direct-Fired
SPACE HEATER

HEATS LARGE AND SMALL PLANTS ECONOMICALLY!

The new Airtherm Direct Fired Space Heater can be installed either with or without duct work. It is available as a floor unit (illustrated) or in horizontal or vertical suspension types. For gas or oil fuel supply. 650,000 to 1,950,000 Btu capacities.

Airtherm Direct Fired Heaters give instant heat, when and where it is wanted, reflecting a saving in fuel costs. They are controlled automatically—require no maintenance.

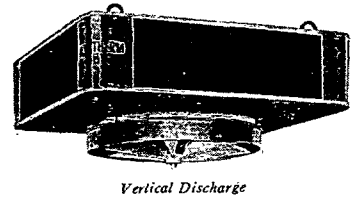
AIRTHERM
STEAM
UNIT HEATERS
(Copper Coils)



Horizontal Propeller Type

Airtherm Steam Unit Heaters move warm air quickly over wide areas into remote corners. They are controlled individually and give uniform heat throughout the room without waste of fuel. Wall or ceiling mounted to save floor space.

Complete range of sizes—send for bulletin giving complete details.

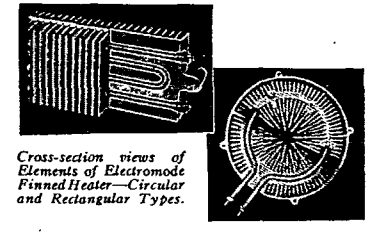


Vertical Discharge

ELECTROMODE CORPORATION
Div. American Foundry Equipment Co.

ELECTROMODE
Electric Unit Heaters

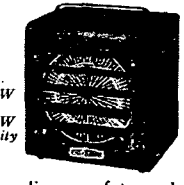
430 So. Byrkit Street, **Mishawaka, Indiana**



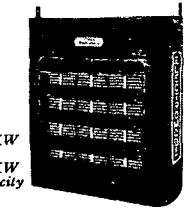
Cross-section views of Elements of Electromode Finned Heater—Circular and Rectangular Types.

All-Electric Air Heaters

Cast aluminum grids in electric heaters are a radically different innovation from the popular conception of electric heating units. Aluminum, a metal of highest thermal conductivity, is cast on a tubular element. This process seals the heating element, preventing oxidation and deterioration.



1.5 KW to 7.5 KW Capacity



10 KW to 60 KW Capacity

To the cleanliness, safety and convenience of Electromode Electric Unit Heaters, there are the further advantages of ease of installation, simplicity of control and economy.

The 1.5 to 7.5 KW sizes are for either portable use or suspension mounting, the larger sizes for suspension mounting only. By means of thermostat, heat can be automatically controlled. For complete data, see *Bulletin 45-U.*

1.5 KW to 7.5 KW Heaters

Model	KW	BTU	CFM Approx.	Shipping Weight Approx.	Price
[a]AA-15	1.5	5122	100	20 lb	$30.00
[b]AN-30	3	10245	200	40 lb	52.50
[c]BN-50	5	17075	350	50 lb	64.00
[d]CN-7½	7.5	25613	550	80 lb	91.00

Heaters 1.5 KW and 3 KW are furnished with ON and OFF switch, 10 feet of heavy duty cord and plug.

[a]Model AA-15 (1.5 KW) may be plugged into usual branch lighting circuit.—made for use on 115 volt circuits only.

[b]AN-30 is for 115 V or 230 V.— specify which.

[c]BN-50 heater is furnished with switch but without cord or plug, 230 volts only, single or three phase.

[d]CN-7½ is furnished without switch, cord or plug. 230 volts only, single or three phase.

Elements For Blast Coils and Air Ducts

Electromode Finned Heater Elements are made for a wide range of heating applications. Engineering help is at your disposal.

Bilt-in Home and Office Heaters

The production of Portable and Bilt-in Wall Heaters, restricted during war, has been resumed and complete details included in Bulletin 46-D.

10 KW to 60 KW Heaters

For 230 or 460 volts—specify which voltage when ordering.

Model	KW	BTU	CFM Approx.	Shipping Weight Approx.	Price
[a]14-10	10	34150	800	100 lb	$110.00
[a]14-12	12	40980	800	100 lb	125.00
[a]18-15	15	51225	1500	180 lb	138.00
18-20	20	68300	1500	180 lb	190.00
20-25	25	85375	1800	200 lb	245.00
20-35	35	119525	1800	200 lb	255.00
27-45	45	153675	4000	360 lb	380.00
27-60	60	204900	4000	360 lb	390.00

[a]Models 14-10, 14-12, 18-15 are for single or 3 phase,—others only for 3 phase.

Heaters are furnished with 460 single or 3-phase heater elements with 115 or 230 volt single phase motor. For 460 volt elements and motor, add 10 % to price.

Contactor

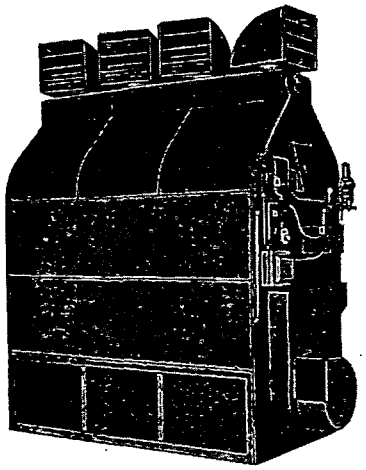
A contactor is necessary on any unit above 5 KW capacity and on 3 phase service and direct current in order to provide operation of the safety switch and thermostat. Automatic thermostat and contactors are available for any size unit.

DRAVO CORPORATION

HEATER DEPARTMENT

Dravo Bldg., 300 Penn Avenue, PITTSBURGH 22, PA.

Sales Offices in Principal Cities



View of standard model as used for gas or oil or gas and oil in combination. Combustion chamber is lined with plastic refractory material at factory, shipped ready for use. Fins, deflectors and corrugation increase heat transfer surface, economizer tubes reduce the temperature of flue gases. Entire design job tested and proved for highest efficiency.

Permanent savings in fuel consumption, cost of maintenance and cost of labor in operation are factors influencing favorable acceptance of Dravo Heaters.

Conditions often demand speed of installation and curtailment in the use of critical materials. Dravo Heaters are shipped with refractory in place, are installed by simply connecting to fuel and power supply and providing foundation breeching and vent stack, and their construction represents a saving in critical metals of 30 to 40 per cent over conventional steam plants with distributing systems.

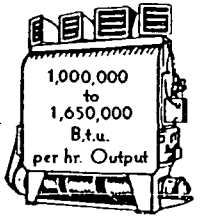
Each heater is complete and operated individually. A heating system for any size structure may be formed with a combination of one or more heaters. Dravo Direct-Fired Heating Systems save man-hours, money, transportation—all essential to the war effort.

Dravo Direct-Fired Heaters are used for permanent installation and also often used to supply temporary heat in new construction or plant expansion.

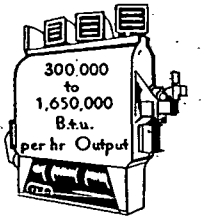
The exceptionally high heat transfer efficiency is the result of two fundamental design features—first, accurately controlled combustion of fuel and air, and second, highly effective transfer of heat to air. Standard stock sizes are obtainable ranging from production of 600,000 to 4,000,000 Btu output per hour. Dravo Bulletin No. 509-A with detailed description mailed on request.

Specification Data Sheets for any type or capacity are furnished on request.

GAS-OIL COMBINATION . . . Floor Set . . . Top Discharge . . . Front Fired . . . Standard V-Belt Drive—This series of heaters is equipped to burn either Heavy or Light oil with alternate gas burner. Equipment consists of a complete, separate burner for each fuel, each burner having a complete set of controls. The controls and wiring are so arranged that either burner can be put into operation by simply throwing a switch, and valves.

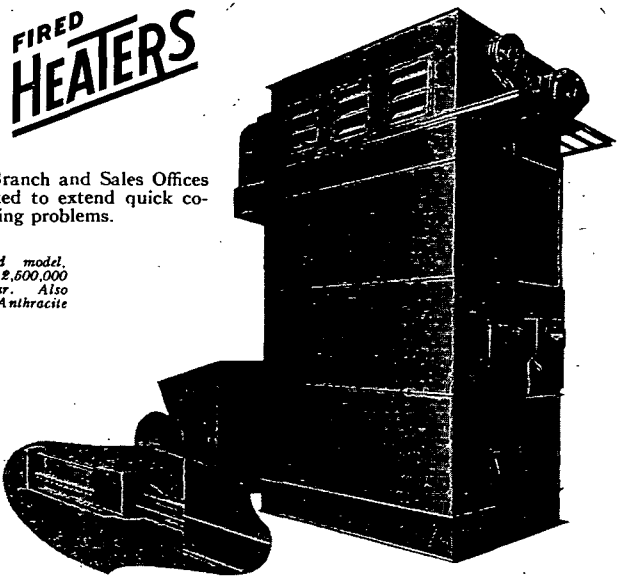


GAS . . . Floor Set . . . Top Discharge . . . Front Fired . . . Standard V-Belt Drive—This series as well as all others is designed to deliver the maximum output per square foot of floor space occupied. Because of their unique design they fit into the heated space and are dependable and efficient. They readily lend themselves to a wide variety of applications, using either natural or manufactured gas.



DRAVO DIRECT FIRED HEATERS

There are 47 Branch and Sales Offices strategically located to extend quick co-operation on heating problems.



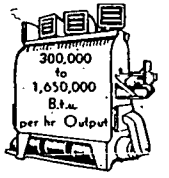
Right. Hopper feed model, capacities 750,000 to 2,600,000 Btu output per hour. Also available for bin-feed. Anthracite or Bituminous coal.

COAL FIRED SERIES—the Dravo coal-fired, self-contained Heater is particularly adaptable in areas where coal is the cheapest or easiest fuel to obtain. Coal fired heaters may be converted to either gas or oil firing should conditions change. The entire series of Dravo Direct-Fired Heaters has the utmost flexibility. Enough applications, combinations, and adaptations are available to meet any set of reasonable conditions. The same unique principle of welded combustion chamber is employed. The entire series has the DRAVO non-clinkering air-cooled setting incorporated in the design. Available for either bituminous, anthracite or lignite coal; equipped with either hopper feed or bin-feed stoker, installed under either end of the heater. Other series also available 2,750,000 through 10,000,000 Btu per hour output.

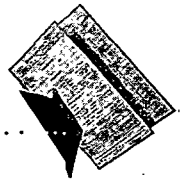
Overhead or underground duct systems may be used for air distribution.

Specification Data Sheets for any type or capacity are furnished on request.

OIL, Floor Set, Top Discharge, Front Fired Standard V-Belt Drive—This series obtainable in light oil or heavy oil fired models in all Dravo Direct-Fired Heaters. Economy of fuel consumption is the result of careful overall designing and constant improvement over many years. In summer they may be used for air circulation.



heating analysis . . .



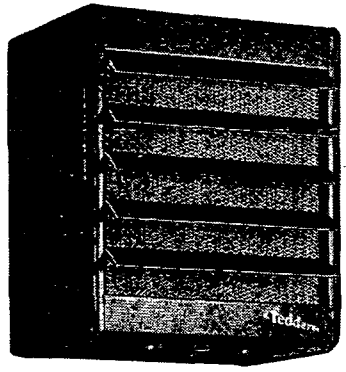
How much would an installation cost?

We have a form called "Dravo Building Survey" that we furnish free on request. You give us the vital statistics on this form and we'll do the rest. Write for your copy.

FEDDERS-QUIGAN CORPORATION

85 Tonawanda St.

Buffalo, New York



Fedders Series 15 Horizontal Unit Heaters with Streamline Copper Tubes and Fins

Fedders UNIT HEATERS

Horizontal Type

COPPER TUBES AND FINS

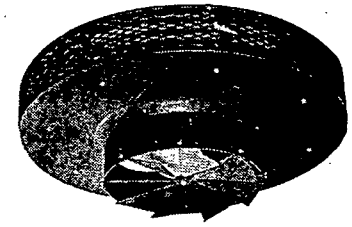
Fedders streamline tubes provide aerodynamic efficiency and the ample-area saddle of the fins gives generous bonded metal-to-metal contact between copper tubes and fins for high thermal efficiency. Handsome, rugged cabinets—complete relief of expansion stresses within the core as well as between core and cabinet—latest type broad blade fans provide large air volume, quiet operation and maximum efficiency. Large Btu capacity with low final temperatures assure ideal working conditions. Capacities, 100 to 1,000 EDR. Write for Bulletin 15C-3.

Fedders

DOWNBLOW

UNIT HEATERS

Used where requirements necessitate clearance for material handling equipment, tall machinery and other conditions requiring piping to be kept out of the way and for spot locations such as over frequently opened shipping room doors.



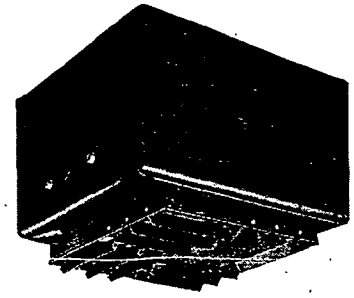
COPPER TUBES AND FINS

Straight downblow or diffusion of heat is accomplished by adjustable directional outlets to fit conditions.

Circular type for ceiling heights from 20 to 45 ft. 800 to 1800 EDR capacities.

Square models chiefly for medium ceiling heights of 15 to 25 ft.

Write for Bulletin 12C-2.



Grinnell Company, Inc.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence 1, R. I.**

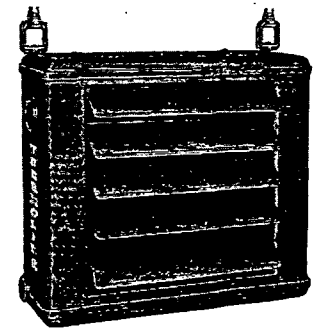
National Distributors of Thermoflex Traps and Heating Specialties

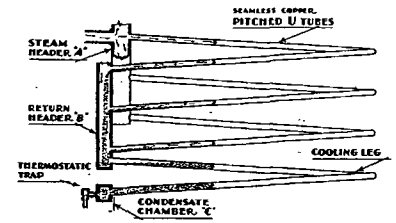
For data on other Grinnell Products, see pages 1112-1113, 1211

THERMOLIER

THE GRINNELL UNIT HEATER

Reg. U. S. Pat. Off.





Patented Construction of Internal Cooling Leg

In addition to its patented Internal Cooling Leg Thermolier has many other desirable features including its Single Header U-Tube construction which compensates for expansion and contraction strains. Radiation is from brass-finned seamless copper U-tubes rolled into a cast iron tube sheet.

Steam circulation and the removal of condensation in Thermolier are distinctly different than is usual in unit heaters. The actual cooling effect of this construction is equal to a run of more than 100 ft of ordinary, exterior cooling leg piping.

Steam is delivered into Chamber "A" of the header and circulates from there through the pitched U tubes, carrying its condensation with it into Chamber "B." By partitioning off the lower tube or tubes at the bottom of the Steam Supply Chamber "A" these tubes carry all condensation from Chamber "B" into Drain Chamber "C." In passage of this condensation through these tubes, the air from the fan is rapidly carrying off heat just as it does in the rest of the unit. The result is that these two bottom tubes form an efficient internal cooling leg, integral with the unit.

Thermolier is available in 10 Models. Catalog will be sent on request to Grinnell Company, Inc., 277 West Exchange Street, Providence 1, R. I., or to any branch office in principal cities listed on our page 1112.

CAPACITIES

60 F Entering Air Temperature—2 Lb Steam Pressure

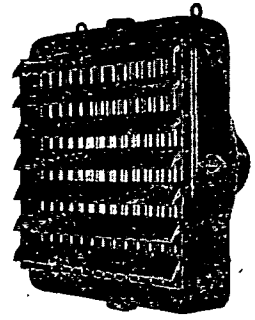
Model Nos.	Btu per Hour	Equivalent Direct Radiation	Model Nos.	Btu per Hour	Equivalent Direct Radiation
D21	35,600	148	D57	101,300	422
D31	48,700	203	D66	128,700	536
D37	62,200	259	D71	151,700	632
D41	71,000	295	D91	196,000	817
D44	84,100	350	D111	275,300	1147

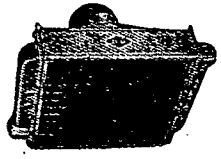
ILG Electric Ventilating Co.

2832 North Crawford Ave., Chicago, Ill.

Offices in 40 Principal Cities

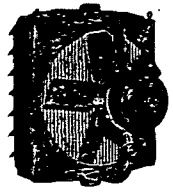
Horizontal Type Unit Heaters—have Self-Cooled Motor which counteracts coil heat—never "slow-roasts." Two piece header for "balanced" steam distribution—orifice bushings expand tubes uniformly in header plate. Fins are pressed into round tubes for permanent union—no brazing, soldering, or welding. Certified ratings. "One-Name-Plate" Guarantee. 40 capacities. Get Catalog No. 341.



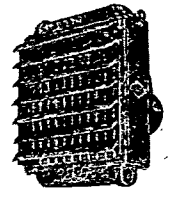


Vertical Type (left)—recommended for extremely high or extremely low ceilings.

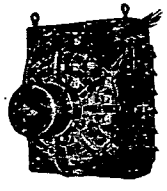
Textile Type (right)—for use where lint, etc., ordinarily adheres to fin surfaces and clogs up the coils. 10 capacities.



Low-Ceiling Type (right)—has side steam connections which make possible an extremely compact installation in locations where head-room is at a premium.



ILG Electric Unit Heaters

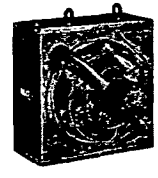


STANDARD TYPE

For instant, clean, safe, dependable heating. Coil is of black heat type which operates below 400 degrees. Protected patented automatic thermal cut-out and magnetic starter. Sizes, 5 KW to 48 KW. Get Bulletin No. 802.

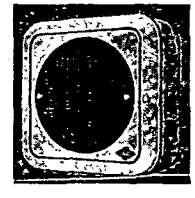
TYPE "HT"

For installations requiring a small volume of heat. Exceptionally efficient. Suitable for constant duty. Black heat type coil with individually interchangeable elements. Automatic circuit breaker protects ILG Self-Cooled Motor from overload. 1½ to 4 KW.



ILG Cooling and Air Conditioning Units

Ceiling type units for use singly or in multiple with remotely located refrigerating machine circulating direct expansion "Freon" or methyl chloride; or with cold water. For cooling, dehumidifying and recirculating. Also combination units for both cooling and heating. Write for Catalog.



For ILG Propeller and Centrifugal Fans, see pages 1060-1061.

Jaden

MANUFACTURING COMPANY

HASTINGS NEBRASKA

SINCE 1914

New and Improved Self Contained Air Conditioning Units—for use with cold water.

Capacities From 1 Ton to 20 Tons with Comparable Capacities for Heating.

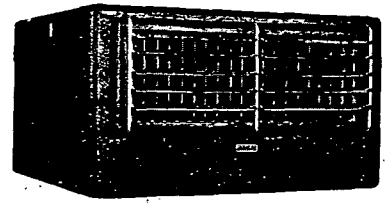
Models for Cooling Include Floor Type for Individual Rooms, Suspended Units and Central Plants for Duct Systems.

Centrifugal Unit Heaters, Package and Utility Type Blowers for Heating and Ventilating.

• **Central Plants**—capacities from 3 tons to 24 tons. Air delivery from 1100 cfm to 10,000 cfm. All central plants are assembled in sections to simplify installation. All models available with steam coils for combination heating and cooling.

• **Suspended Models**—capacities from 1 ton to 4 tons. Air delivery 585 cfm to 2200 cfm. Especially adaptable to individual rooms where duct system is undesirable. Completely self-contained with motor, coils, blower and filters housed in sturdy steel cabinet with attractive featherweave finishes.

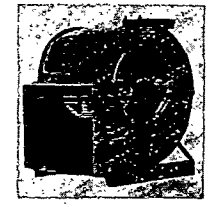
• **Floor Type Models**—capacities from 1 ton to 3 tons. Designed for use in individual rooms. All models self-contained with motor, blower coil and filters housed in well designed metal cabinets attractively finished or available in base coat for finishing in special colors.



• **Package Blowers**—capacities 1000 cfm to 4000 cfm. All models equipped with filters and available with or without fan controls. Cabinets of extra heavy steel construction with featherweave finish or with base coat for finishing in special colors.

• **Utility Blowers** both single—twin mounted double inlet type—capacity from 800 cfm to 10,000 cfm. Available in any discharge arrangement and with or without motors. Single blowers available in "base type" which includes heavy angle frame for special cabinet construction when necessary.

• **Unit Heaters**—capacities from 76,800 to 155,000 Btu/hr. std. rating, air delivery 1165 cfm to 2100 cfm. Centrifugal type for use with steam or forced circulated hot water. Unusually quiet in operation—highly efficient and easily adaptable to duct installation. Finished in attractive featherweave or base coat for special colors.



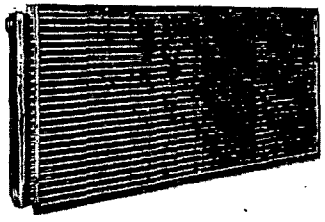
WRITE FOR CURRENT SPECIFICATION CATALOG

KENNARD CORPORATION

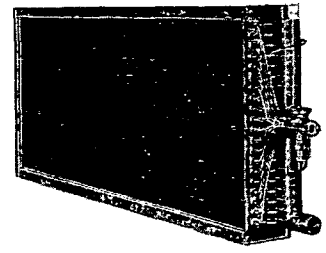
2019 So. Hanley Rd., **St. Louis 17, Mo.**

Manufacturers

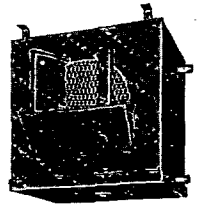
AIR CONDITIONING BLOWER UNITS—REFRIGERATION BLOWER UNITS
AIR CONDITIONING COILS—HEATING COILS



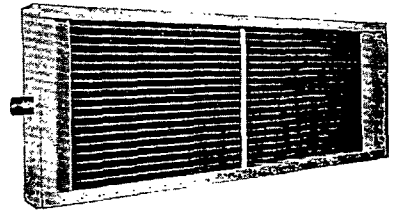
KENNARD Water Cooling and Heating Coils. Non-ferrous coils for cooling with cold water and for heating with hot water.



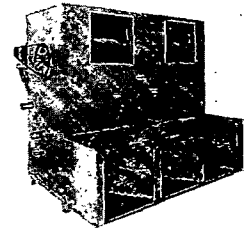
KENNARD Direct Expansion Coils. For all refrigerants. A complete range of sizes to meet all conditions and capacities.



KENNARD RS Unit. 100 per cent utilization of coil the Kennard Way. Full coil surface is utilized when air is drawn through coil. Units for brine, freon, ammonia and methyl-chloride. 9 sizes basic capacity ratings 410 Btu's to 1521 Btu's per degree temperature difference.



KENNARD Steam Coils. A complete range of copper steam heating coils. Can be furnished in non-freeze type with steam distributing tubes.

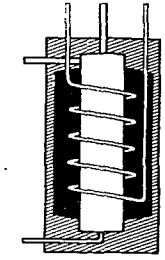


KENNARD Floor and Ceiling Type Air Conditioning Units. Ten sizes. Two to 35 tons cooling capacity range and 40,000 to 1,250,000 Btu's range in heating. Various arrangements of discharge, filter box, motor drive.

WRITE FOR COMPLETE INFORMATION

Kramer Trenton Co.

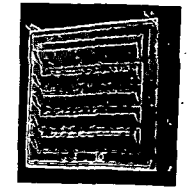
Manufacturers of
HEATING, COOLING AND REFRIGERATION DEVICES
Trenton, New Jersey



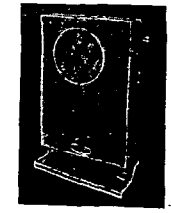
PATENT APPLIED FOR

Kramer THERMOBANK

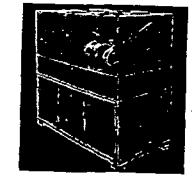
The completely automatic refrigeration system for temperatures below the frost line without the use of electric heaters, water spray or brine spray.



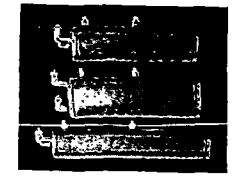
Kramer "Coolmaster" Product Coolers
An outstanding product cooler designed to give positive refrigerant distribution and maximum performance. Built-in heat exchanger. Adjustable louvers.



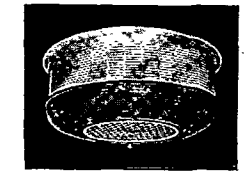
Kramer Panel Type Unit Coolers
Designed for application where space is at a premium. Easily accessible for adjustments and service without use of tools. All-copper evaporating coil. Built-in heat exchanger.



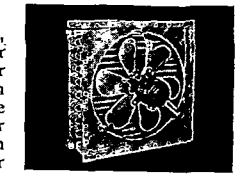
Kramer Floor Type Product Coolers
Designed for large cold storage installations. Capacities range from 3 to 10 tons. For all refrigerants. Top or front discharge.



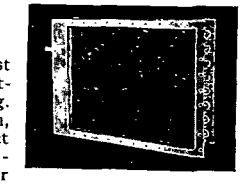
Kramer Fin-Tube Condensers
Combined water-cooled condenser and liquid receiver.



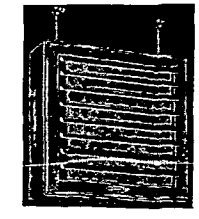
Kramer Radial Unit Cooler
Even air distribution in all directions. Low discharge velocity. High relative humidity. Oilless ball bearing motors. Maximum space economy. Pleasing appearance.



Kramer Unicon
Self-contained, remote type air cooled condenser complete with fan. May be used as booster in combination with air or water cooled systems.



Kramer Turbo-Fin Surface
All-copper blast surface, for heating and cooling. Used for steam, water or direct expansion refrigerants. Air side and refrigerant side flow-disturbers for increased heat transfer. Fin-tube ratio designed for maximum latent heat removal. Coil construction designed to allow thermal expansion. Electro-tin plate finish.



Kramer Unit Heaters
All-copper condensing coil with fused metallical fin-tube bond. Positive compensation for thermal expansion. High discharge air velocity. Moderate final air temperature. Silent in operation.

McQuay, Inc.

1602 Broadway, N.E., **Minneapolis, Minn.**

MANUFACTURERS OF AIR CONDITIONING EQUIPMENT

Sales Offices in all Principal Cities

- **Air Conditioners**
- **Air Conditioning Coils**
- **Blast Heating Coils**
- **Refrigeration Coils**
- **Convection Radiation**
- **Unit Heaters**
- **Unit Coolers**

PROVEN McQuay PRODUCTS

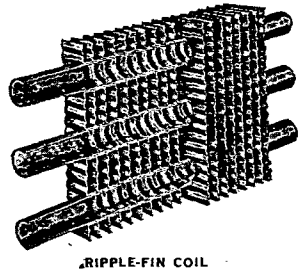
- **Comfort Coolers**
- **Blower Coolers (Suspended & Floor Type)**
- **Room Coolers (Cabinet Type)**
- **Ice Cube Makers**
- **Icy-Flo Accumulators**
- **Zeropak Low Temp. Units**

THE EXCLUSIVE McQUAY RIPPLE-FIN COIL ASSEMBLY

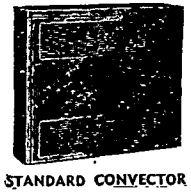
The McQuay Fin and Tube assembly in all McQuay coils and cores is one of the reasons McQuay products are considered "Tops in Over-All Efficiency" by many heating and refrigeration authorities.

Heat transfer efficiency primarily depends on three elements in coil construction. First, "Area of Contact," Second, "Contact Pressure" and finally "Quality of Contact" between collar and tube.

In McQuay coils *all three* necessary elements are found developed to their highest degree. The famous McQuay "Wide Fin Collar" plus Exclusive Hydraulic Expansion together with the polished surface, secured by "spinning" the fin collar, truly provides the last word in Heat Transfer.



RIPPLE-FIN COIL



STANDARD CONVECTOR

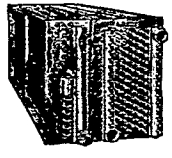
McQUAY STANDARD CONVECTORS

The Standard all purpose Convector has been designed to meet all heating requirements. They are available for free standing, partially recessed, fully recessed and wall mounting applications.

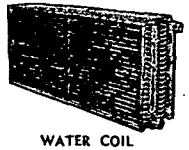
All enclosures are constructed from high grade steel, properly reinforced to make a sturdy cabinet.

The heating element is constructed of a series of round seamless copper tubes to which are attached die formed radiating fins, which are bonded to the tubes by an exclusive process.

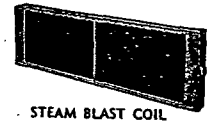
We offer the services of our Engineering and design department to help solve your heating problems.



COMBINATION COOLING COIL



WATER COIL



STEAM BLAST COIL

MORE THAN 1,000,000 STANDARD COIL TYPES AND SIZES

McQuay manufactures a complete line of Standard Coils for the Industry.

Coils for Heating—1 to 10 rows deep using low or high pressure steam or hot water.

Non-Freeze—(steam distributing tube) type coils 1 and 2 rows deep.

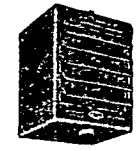
Removable Plug—(cleanable tube) type coils 1 to 12 rows deep.

Water Coils for Cooling—1 to 12 rows deep.

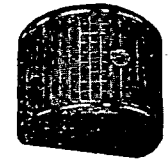
Direct Expansion Coils—for cooling 1 to 10 rows deep.

Refrigeration Coils—all types and sizes.

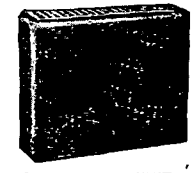
Special Coils—of various materials furnished on order for special applications.



STANDARD UNIT HEATER

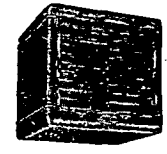


RADIAL UNIT HEATER

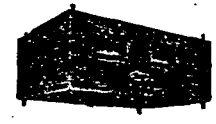


CABINET TYPE UNIT HEATER

COMFORT COOLER



COMFORT COOLER



AIR CONDITIONER (YEAR-ROUND)

UNIT HEATERS

The Radial Unit Heater, an exclusive McQuay development, is designed to do what no other unit heater can do—namely, distribute heat over a semi-circular area.

This Patented design, with the fins themselves acting as directional louvers, enables the radial heater to do the work of two or more conventional units. This naturally reduces installation costs, both from the standpoint of labor and material.

In addition to Radial Unit Heaters, McQuay also manufactures the Standard, or horizontal propeller type, the Down Flow, the Cabinet, as well as large capacity blower types.

For long life, efficiency, and for eye appeal specify McQuay Unit Heaters. They are available in a wide range of sizes to meet all applications.

McQUAY COOLERS

For Industrial Application

Made in two types—one for use with water or brine; another for Freon or methyl chloride. Eight sizes in each type—all with 4-speed motors.

AIR CONDITIONERS—COLD WATER AND FREON TYPES

For Industrial Application

Choice of recirculation of indoor air, entire intake of outside air, or a combination of both. Cold water or brine used in one type; Freon or methyl chloride in another. Modern "sound isolated" construction assures quiet operation. Capacities to 6 tons.

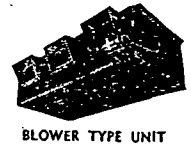
CENTRAL SYSTEM AIR CONDITIONING UNITS

Suspended and floor types, cools, dehumidifies, filters, and circulates air in summer, heats, humidifies, filters and circulates air in winter. Extreme flexibility and accessibility "built-in." Cooling capacities from 5 to 50 tons in both Suspended and Floor Type.

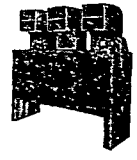
McQUAY ICY-FLO ACCUMULATORS

The new practical "Storage-Battery" for refrigeration effect is now available for handling heavy loads of short duration.

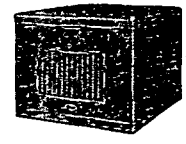
DOWN FLOW UNIT HEATER



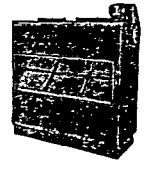
BLOWER TYPE UNIT HEATER



BLOWER TYPE UNIT HEATER



AIR CONDITIONER



AIR CONDITIONER (YEAR-ROUND)

ACCUMULATOR

Modine Manufacturing Company

Heating and Air Conditioning Division

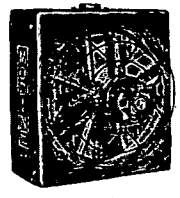
General Offices: 17th and Holburn Sts., **Racine, Wis.**

Factories at Racine, Wis. and La Porte, Ind.

Branches in all Principal Cities

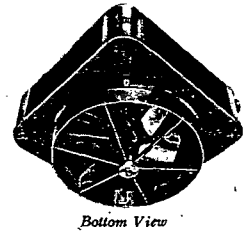
MODINE UNIT HEATERS

Front View



Back View

★ VERTICAL DELIVERY MODELS ★



Bottom View

HORIZONTAL DELIVERY MODELS

Condenser—Copper and copper alloy, inlet to outlet, for maximum resistance to internal and external corrosion. Pure copper fins are metallically bonded to round, seamless, heavy-gauge red brass tubes for permanent contact and to insure uninterrupted heat conduction from primary to secondary surface. Modine-patented expansion bend permits tubes to expand or contract individually as temperature requires. All steam and condensate carrying passages are brazed into an integral pressure-resisting unit.

Direct - Pipe - Suspension — Modine patented feature permits suspending unit directly from supply line without additional time and labor wasting supports; also permits complete rotation of unit for redirection of air stream.

Bonderized Casing—Casing protected from rust by Parker Bonderizing.

Safety Fan Guard—Staunch, steel safeguard built into unit protects against danger of unshielded fan.

CAPACITIES AND DIMENSIONS

Model No.	Btu Hr.*	Cfm	Rpm	Over-All Height	Width	Depth Less Motor
H-102	24,500	410	1580	14"	11⅛"	6"
H-144	34,600	585	1580	17¾"	14¼"	9"
H-184	44,200	685	1580	20⅛"	14¼"	9"
H-254	61,000	1020	1140	22½"	19"	11¼"
H-364	87,400	1500	1140	24"	19"	11¼"
H-474	113,800	1900	1140	27½"	23"	11½"
H-584	141,000	2400	1140	27½"	23"	11½"
H-704	169,000	2650	1140	29½"	24½"	11½"
H-894	214,500	3600	1140	33¾"	26½"	13¼"
H-1154	277,000	4300	1140	33¾"	26½"	13¼"

Condenser—All copper and copper alloy. Tubes are seamless red brass. Headers, inlet and outlet connections are heavy-walled copper pipes. Fins are pure copper. Brazing of all steam carrying passages and metallic bonding of fins to tubes insures rugged unit and continuously excellent heating. Hollow-square condenser literally grows as it heats up; retreats to normal size as it cools with complete distribution of stresses and absence of strains where injury might result. Parallel alignment of fins reduces possibility of dirt lodging between fins and clogging condenser.

Cone-Jet Deflectors—Verticals are regularly equipped with radial or spoke-like deflector assemblies illustrated above. Individually adjustable deflector blades make possible delivery of high, narrow jet-like cone of heated air from high elevation . . . or low, broad, softened-velocity cone from low elevation.

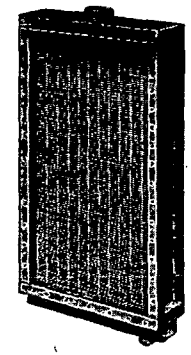
TrunCone Deflectors—Verticals can be furnished with special deflectors for use on units mounted at heights lower than recommended for Cone-Jet Deflectors.

CAPACITIES AND DIMENSIONS

Model No.	Btu/Hr*	Cfm	Rpm	Overall Dimension (Square)
V-152	36,500	610	1580	17½"
V-332	79,700	1370	1140	22"
V-604	145,000	2700	1140	33"
V-724	173,800	3400	1725	33"
V-794	190,200	3300	1140	33"
V-974	233,500	4300	1725	33"
V-1084	260,000	4550	1140	33"
V-1404	337,000	5600	1140	42½"
V-2004	481,000	8000	1140	42½"
V-2504	601,000	10,000	1140	42½"

*Btu at 2 lb steam, 60 deg Ent. Air.

BLAST HEATERS



For modern air conditioning, heating, ventilating, drying and processing systems. Condenser is pure copper and copper alloy, inlet to outlet, for resistance to corrosion, durability, light weight and high heat transfer. "Floating" condenser construction provides safe expansion and contraction. Fins are metallically bonded to tubes to prevent corrosion from reducing heat transfer capacity. Die formed fins promote greater heat transfer; permit use of smaller, lighter coils. Modine design permits use of ducts having cross-sectional areas no greater than those of condenser face. Wide range of capacities and sizes available in Standard and Booster Unit types.

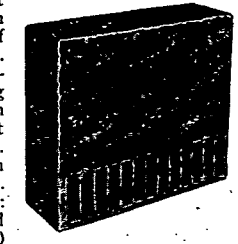
STEAM DISTRIBUTION TYPE BLAST HEATERS

Cross-section supply end, showing double tube construction

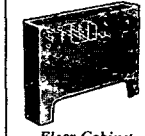
These coils are characterized by the uniform distribution of steam throughout entire heating surface . . . even when steam is partially throttled to meet system temperature demands. Inserted in each condenser tube is a steam-distributing tube with small, accurately-sized and spaced orifices along its entire length. Steam entering distributing tube is uniformly rationed through orifices into external or condenser tube . . . then flows as condensate into return header. Uniform steam distribution minimizes tendency of condensate freezing and tube damage; eliminates need for preheaters or tempering coils, thus simplifying system design and control; eliminates air stream stratification.

CABINET UNIT HEATERS

Designed for heating offices, lobbies, corridors, etc. . . . wherever quick, positive distribution of heat combined with quiet operation and gentle air movement are desirable. Has considerably greater heat output than convector of equivalent size. Employs copper heating coil for use on steam or hot water system. Mounted on wall or ceiling. Capacities: 105 Edr. and 310 and 450 Edr.



CONVECTOR RADIATION



Floor Cabinet Type

Built to meet modern residential, institutional, and commercial heating requirements, Modine convectors operate on the principle of *natural convection*. Cooler, heavier air is drawn in through the enclosure's bottom opening; comes in contact with copper heating unit that carries steam or hot water. As air is heated, it rises; is then circulated out into room through grille at top of enclosure.

Concealed copper heating unit combines compactness with high heat capacity—to give room added floor space as well as fast, even heating. Use of copper makes the heating unit almost instantly responsive to automatic control.

Enclosures are streamlined for simplicity and harmonize readily with any type of interior. Parker-Bonderized before being prime painted, the enclosures are less vulnerable to the formation and spread of rust.

Manually installed enclosure fronts can be installed in 30 seconds without tools. Easy removal of fronts assures convenience in cleaning and gives ready access to the heating unit.

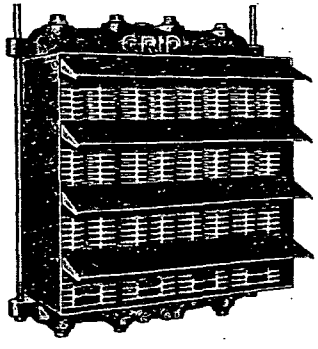
Available in recessed and cabinet types.

D. J. Murray Manufacturing Co.

Wausau, Wisconsin

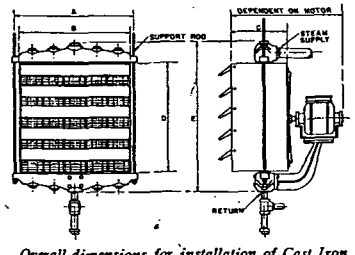
Offices in Principal Cities

MANUFACTURERS OF THE GRID UNIT
AND GRID BLAST COILS



One piece construction "fin" heating sections of high test cast iron—no soldered, brazed, welded or expanded connections. Patented.

Designed and tested to operate with steam or hot water systems—for steam pressures from 2 lbs to 250 lbs. Engineered along the same lines as the standard GRID Unit which had aluminum heating sections and has been on the market since 1929.



Overall dimensions for installation of Cast Iron GRID Unit Heater

CI (CAST IRON) SERIES GRID UNIT HEATER DATA

Model No.	Dimensions					Motor		Vol. Fan Cfm	Capacities 5 Psi Steam 60° F		Approx. Shipp. Wt. Lbs.	Pipe Size		Support Rod Diam.
	A	B	C	D	E	Hp	Rpm		Btu/ Hr	Final Air TMP°F		Supply	Return	
CI-1000	11¾	10½	12¼	10⅛	16	1/20	1750	675	30010	101	160	1¼	1¼	½
CI-1200	14⅛	13⅛	12¼	13⅛	19⅛	1/8	1750	1142	54000	103	230	1¼	1¼	½
CI-1500	17½	15⅜	12	16	23⅜	1/8	1750	1500	76500	107	300	1½	1¼	½
CI-2000	22⅛	20⅛	12	21⅛	28⅜	1/6	1150	2600	143000	110	500	2	1¼	½
CI-2504	27½	25⅜	13	25⅜	35¼	1/4	1150	3300	206000	117	675	2*	1¼	⅝
CI-2500	27½	25⅜	13	25⅜	35¼	1/2	1150	4350	224000	107	700	2*	1¼	⅝
CI-3000	32⅝	31	13	31	40½	1/2	850	6300	332000	108	1025	2½*	1¼	⅝
CI-3000	32⅝	31	13	31	40½	1/2	1150	8000	380000	103	1075	2½*	1¼	⅝

*Furnished also with 2 in. top supply connection inlet.

NO ELECTROLYSIS TO CAUSE CORROSION

Low maintenance expense.
More air changes per hour.
Positive "directed" heat.
No leaks—no breakdowns.
Lower outlet temperature.
Larger air volume.
No soldered, brazed or expanded joints.
Open design that keeps units clean.

Send for complete catalog information

Send for information on Blast coils and radiation.

National Heater Co.

401 Essex Bldg., **Minneapolis 2, Minn.**

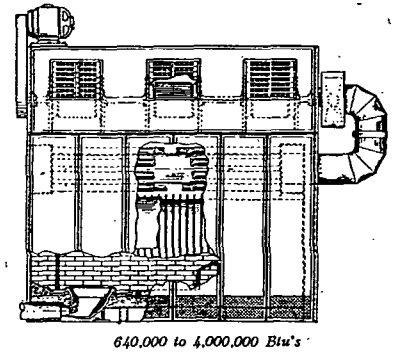
NATIONAL CHAMPION

STEEL S-P-A-C-E HEATERS

OIL—GAS—STOKER HAND FIRED UNITS

As shown in diagram these heaters are built as a complete unit consisting of heater, casing, firing equipment, multiple blower assembly, motor, drive, induced draft blower and controls. NATIONAL HEATERS are constructed of welded copper bearing hot rolled one quarter inch boiler plate steel instead of the usual lighter ten guage plate. This provides positive assurance of life-long gas and smoke tight operation with ample provision for absorbing all strains resulting from expansion and contraction.

The body of the heat generator is long and narrow in design, allowing complete fuel consumption with properly spaced radiating fins to facilitate maximum heat transmission from this primary heating surface.



640,000 to 4,000,000 Btu's

Casings are sturdy with insulated double metal walls to reduce wasteful radiation to a minimum. Panel type construction with rigid angle iron frames and structural iron reinforcement at both top and bottom assures a strong durable installation. Casings are designed for quick and convenient assembly.

ENGINEERING RATING CHART

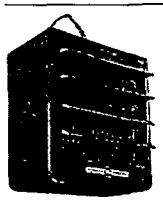
HEATER NUMBER	B.T.U. OUTPUT CAPACITY	½" S.P. C.F.M. MAX.	GRATE AREA SQ.FT.	SQ.FT. HEATING SURFACE	VOLUME CU.FT. INSIDE HEATER	BLOWER MAXIMUM	BLOWER MOTOR H.P. MAX.	BLOWER MOTOR R.P.M.	TUBES NO.	TUBES DIA.	APPROXIMATE WEIGHT LESS STOKER
300	640,000	8,200	6.5	320	75	18"TWIN	3	1750	2	8"	8,850
350	860,000	11,000	8.0	430	100	18"TWIN	5	1750	2	8"	14,420
500	1,215,000	18,150	12.0	600	130	21"TWIN	7.5	1750	10	4"	15,860
550	2,000,000	32,000	14.7	844	175	21"TRIPLE	10	1750	12	4"	16,610
600	2,500,000	45,000	16.0	1421	315	27"TRIPLE	15	1750	16	4"	20,850
650	3,000,000	52,000	20.0	1685	375	30"TRIPLE	20	1750	16	4"	24,090
700	3,500,000	56,000	24.0	1810	450	30"TRIPLE	25	1750	18	4"	26,340
750	4,000,000	60,000	28.00	1935	450	30"TRIPLE	30	1750	20	4"	28,470

THE HERMAN NELSON CORPORATION

General Offices and Factories at Moline, Illinois

Branch Offices and Product-Application Engineers in the Following Cities:

Albuquerque, N. M.	Dallas, Tex.	Louisville, Ky.	Richmond, Va.
Atlanta, Ga.	Denver, Colo.	Memphis, Tenn.	Saginaw, Mich.
Baltimore, Md.	Des Moines, Iowa	Miami, Fla.	St. Louis, Mo.
Birmingham, Ala.	Detroit, Mich.	Milwaukee, Wis.	Salt Lake City, Utah
Boston, Mass.	Duluth, Minn.	Minneapolis, Minn.	San Antonio, Tex.
Buffalo, N. Y.	El Paso, Tex.	Moline, Ill.	San Francisco, Calif.
Cape Elizabeth, Me.	Grand Rapids, Mich.	Nashville, Tenn.	Seattle, Wash.
Charlotte, N. C.	Houston, Tex.	New Orleans, La.	Spokane, Wash
Chicago, Ill.	Indianapolis, Ind.	New York, N. Y.	Springfield, Mass.
Cincinnati, O.	Jackson, Miss.	Oklahoma City, Okla.	Syracuse, N. Y.
Cleveland, O.	Kansas City, Mo.	Omaha, Neb.	Washington, D. C.
Columbus, O.	Los Angeles, Calif.	Philadelphia, Pa.	

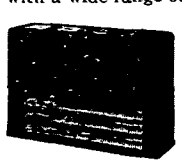


HERMAN NELSON HORIZONTAL SHAFT PROPELLER-FAN TYPE UNIT HEATERS

Designed for ceiling suspension, these unit heaters project warm air downward in an angular direction. Copper heating element for use with steam or hot water, incorporates patented stay tube which maintains proper relationship between headers without increasing strain on loops thus prolonging life of unit. Forty-eight models, sizes and arrangements.

HERMAN NELSON VERTICAL SHAFT PROPELLER-FAN TYPE UNIT HEATERS

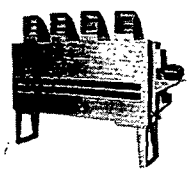
For high ceiling installations. Discharge air vertically downward, or at an angle to vertical in various directions. Long life copper heating element for use with steam or hot water incorporates patented stay tube. Units available with either high or low velocity discharge, each with a wide range of capacities.



HERMAN NELSON DE LUXE UNIT HEATERS

Efficient, economical, compact, quiet and attractive, these Unit Heaters provide the ideal method for heating offices, showrooms, corridors, markets, stores, etc. Copper heating element incorporates patented stay tube. Units may be placed on floor, wall or suspended from ceiling. Eighteen models, sizes and arrangements.

HERMAN NELSON BLOWER-FAN TYPE UNIT HEATERS

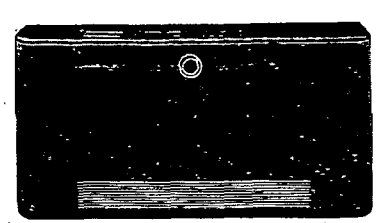


For efficient heating of large areas. Can be supplied with by-pass damper for introduction of indoor and outdoor air. Design of copper heating element assures durability and contributes to high velocity discharge. For floor, wall, ceiling or inverted wall mounting.

HERMAN NELSON UNIT VENTILATORS

Maintain desired air conditions for room or auditorium areas where large groups of people gather. Copper heating element for use with steam or hot water is sturdily constructed for long life and designed for greatest heating efficiency.

Exclusive "draw-through" design prevents unhealthful drafts and eliminates unnecessary overheating. Locating motor in end compartment provides additional space for fan assembly and use of larger fans running at slower tip speeds.



Herman Nelson Unit Heaters and Unit Ventilators are tested and rated in accordance with the Standard Test Code adopted jointly by the *Industrial Unit Heater Association* and the American Society of Heating and Ventilating Engineers.

THE HERMAN NELSON CORPORATION

General Offices and Factories at Moline, Illinois

HERMAN NELSON DIRECT DRIVE PROPELLER FANS

Provide most economical form of quality ventilation obtainable for industrial buildings of all types. Eighteen standard sizes available with wheel diameters from 10 to 36 in. and capacities from 655 to 12,400 cfm. There are sixteen high powered models to operate against static resistance of ⅜ in., with wheel diameters from 14 to 36 in. and capacities from 1260 to 22,690 cfm. Can be obtained in acid and explosion resisting models. Also three models especially adapted to small store and office applications.

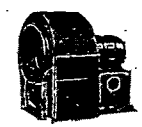
HERMAN NELSON BELT DRIVE PROPELLER FANS

For public and commercial building installations where slow speed, quiet operation are required. Twelve sizes of the standard model with wheel diameters from 24 in. to 54 in. Also six sizes of the High Powered model with the same wheel diameters. Capacities: 5650 to 36,150 cfm. Due to quiet operation of Herman Nelson Belt Drive Propeller Fans, use of two speed motor is unnecessary.

HERMAN NELSON DIRECT DRIVE UNIT BLOWERS

Designed for many applications, such as fume hoods, toilet ventilation, chemical laboratories, industrial processing and drying problems. Compact, direct connected, motor driven units have universal discharge and mount on floor, wall or ceiling. Available in five sizes with 14 speed combinations. Wheel diameters from 6½ in. to 15 in. and capacities from 360 to 4320 cfm.

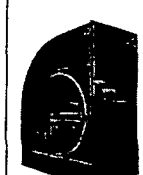
HERMAN NELSON BELT DRIVE UNIT BLOWERS



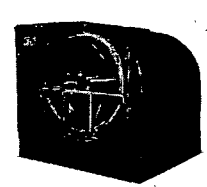
Fully self-contained unit including motor, drives and housing; forwardly curved blade wheel; adjustable motor pedestal with vibration dampers; universal discharge; nine sizes with 52 drive combinations. Available with any rotation and discharge. Wheel diameters from 11 in. to 30 in. and capacities from 1035 to 16,940 cfm.

HERMAN NELSON TYPE "H" AND TYPE "HB" CENTRIFUGAL FANS

Especially suited for heating, air conditioning and ventilating systems. Type "H" has forwardly curved blade wheels. Type "HB" has backwardly curved blade wheels incorporating non-overloading power characteristics. Seventeen wheel diameters from 12¼ to 73 in.; single or double width; Class I or II construction; and any speed or discharge requirement.



Type HB



Type H

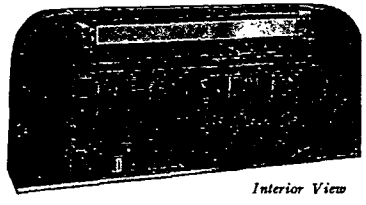
The Complete Line of Herman Nelson Propeller Fans and Blowers is tested and rated in accordance with the Standard Test Code adopted jointly by the *National Association of Fan Manufacturers* and the American Society of Heating and Ventilating Engineers.

John J. Nesbitt, Inc.

Holmesburg, Philadelphia 36, Pa.

11 Park Place, New York 7, N. Y.

Manufacturers of
THE NESBITT SYNCRETIZER Heating and Ventilating Unit, sold by John J. Nesbitt, Inc., and American Blower Corporation;
NESBITT HEATING SURFACE with Dual Steam-distributing Tubes,
NESBITT SERIES H HEATING SURFACE, and
NESBITT SERIES W COOLING SURFACE, sold by leading manufacturers of fan-system apparatus;
WEBSTER-NESBITT UNIT HEATERS (See page 1223), distributed in U. S. A. by Warren Webster & Company.



Interior View

NESBITT SYNCRETIZER—Series 400

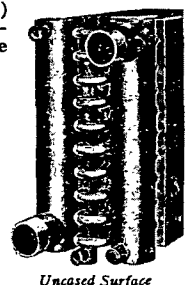
The last word in heating and ventilating units for schoolrooms, offices, etc., where the continuous introduction of outdoor air is desired. For engineering data, get Publication No. 225-2; for "The Story of Syncretized Air," Publication No. 231-2.

Nesbitt Series B Thermovent

For heating and ventilating auditoriums, gymnasiums, assembly halls, and similar gathering places. Publication No. 227-2.

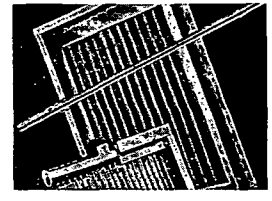
NESBITT COOLING SURFACE

Series W (Water) Surface with exclusive drain feature



Uncased Surface Showing Drain Header

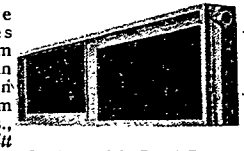
For air cooling and cooling and dehumidifying (with cold water) or air heating (with hot water). Constructed of copper tubes and plate-type aluminum fins. Available in either continuous or cleanable tube type, in single sections having one to eight rows of tubes deep, in three fin spacings, in eleven fin widths, and up to sixteen finned tube lengths. Sturdy galvanized casings. For particulars and engineering data send for Publications No. 233.



NESBITT HEATING SURFACES
With Dual Steam-distributing Tubes

Copper tube-and-fin surface for low-pressure applications. Perfectly adapted to close, continuous automatic control with modulating steam valves. Steam-distributing tubes within the condensing tubes carry the steam equally to the full section assuring UNIFORM discharge temperatures even under a throttled steam supply; eliminating temperature stratification; preventing tube freezing without preheaters; giving ideal system results.

Cased or uncased units of many sizes and capacities. For full particulars and engineering data, send for Publication No. 245.



For above advantages plus uniform distribution in extended fin lengths from 86 to 122 ins., specify *Nesbitt Duplex Heating Surface* with Dual Steam-distributing Tubes. Publication No. 245.

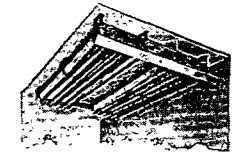
Nesbitt Series H Heating Surface

A lightweight, enduring, highly efficient blast-coil heating surface designed for use with steam pressures up to 200 lb gauge. Well suited to high-pressure as well as low-pressure applications. Seven types, each in eight fin widths and up to seventeen finned lengths—a total of 284 sizes from which to select. Send for Publication No 232 for complete engineering data.

Refrigeration Economics Co., Inc.

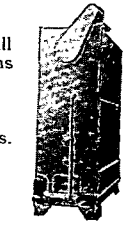
1231 Tuscarawas St. E., Canton 4, Ohio

RECOY PRODUCTS



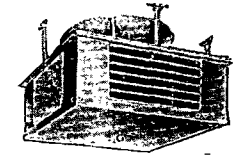
C. T. COILS

Continuous-tube down-draft fin-coils are still unsurpassed for meat coolers. Other forms available for practically any application.



EVAPORATIVE CONDENSERS

Evaporative condensers from 2 to 100 tons. Brine spray cooling to 25 tons.

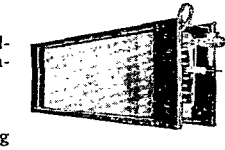


CEILING DIFFUSER

Ceiling diffusers distribute the cooled air across the ceiling, so the blast does not strike the products stored.

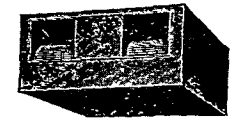
C. F. COILS

Continuous fin coils for unit coolers, blast heaters, air conditioning and condensers.



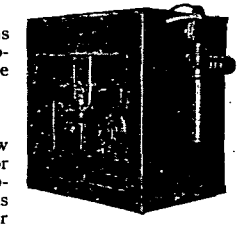
AIR CONDITIONING

Air conditioning units of ceiling or floor type in all capacities, for cooling, heating, or both.



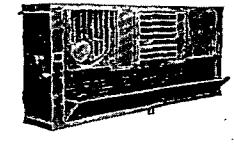
COOLANT COOLING

Coolant coolers from 1 to 20 tons are a necessity for modern production of accurate machine parts.



WALL UNITS

Wall units for long air throw discharge air horizontally. For meats and other products subject to shrinkage air flow is reversed to mix with room air before contacting stored products.

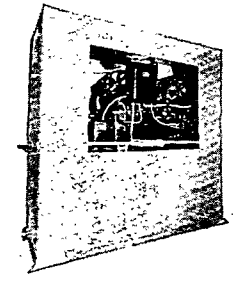


WATER COOLING

Self contained complete ice water and brine coolers complete with high and low sides, circulating pumps, controls, and insulation.

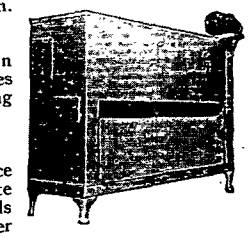
SHELL CONDENSER

Shell and tube, also shell and fin coil condensers. Both types have tubes arranged for cleaning with tube cleaner.



FLOOR UNITS

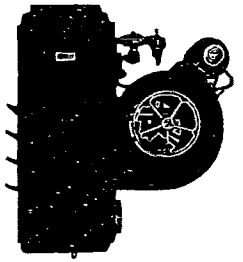
Floor units with cooling surface exposed to view have a definite advantage over those with coils hidden. Design permits water defrosting.



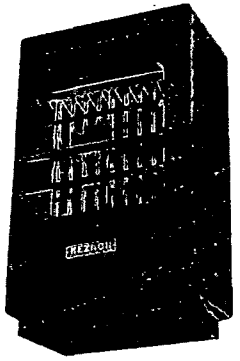
Reznor Manufacturing Co.

Specialists in Gas Heating Since 1888

Mercer **Pennsylvania**



Side view of Reznor Blower Type



Front view of Reznor Fan Type Suspended Unit Heater

Suspended Gas-Fired Heaters

A Reznor unit will solve most *any* heating problem . . . large or small areas. Complete in one package, Reznor gas-fired *suspended* unit heaters (fan and blower types) are literally self-contained heating plants. For installation all they need are a gas line, a flue, and electric supply. Reznor heaters are made in 9 sizes with input ratings ranging from 55,000 to 400,000 Btu's. They operate on natural and manufactured gases, butane and propane. Reznor heaters are finished in a seal brown wrinkle finish which retains its beauty for years.

Fan and Blower Types

Reznor offers the blower type unit heater in the same nine sizes as fan unit heaters. Blower heaters and fan heaters are identical in every respect (including standard and optional controls) except for the air moving power.

Blower heaters are frequently specified where *extreme* quietness is required, or where it is desirable to carry the cold air to the heater through a duct system.

Forced Air (floor type)

Entirely self-contained, *completely automatic forced air heaters* with every feature of the central plant except size but with the beauty of fine furniture—these new heaters represent an outstanding development in gas heating equipment.

Room temperature, gas pressure, air delivery and emergency shutoff are all automatically and accurately controlled without thought or effort on the part of the user. All controls are completely installed by factory.

Heater Number	HEATER INPUT RATING			OUTPUT (Vented)		AIR		MOTORS				FAN		Shipping Weight
	Btu. per Hr.	Cu. Ft. per Hr. Btu. 650	1000	Btu. per Hr.	Sq. Ft. Steam Equiv. Rad.	Free Del. in. CFM	Ave. Vel. in FPM	App. H.P.	Type	RPM	Amp.	Dia. in Ins.	Pitch in Deg.	
US55	55,000	100	55	45,650	190	800	540	1/50	Sh.Pl.	1000	.9	14	18	145
US75	75,000	138	75	62,250	259	1350	675	1/20	Sp.Ph.	1140	1.4	16	23	185
US90	90,000	164	90	74,700	311	1350	675	1/20	"	1140	1.4	16	23	244
US110	110,000	200	110	91,300	380	1700	850	1/8	"	1140	2.8	16	27	244
US130	130,000	238	130	107,900	450	1700	850	1/8	"	1140	2.8	16	27	295
US150	150,000	273	150	124,500	519	2000	850	1/8	"	1140	2.8	16	33	295
US200	200,000	364	200	166,000	692	2700	1110	1/6	"	1140	3.9	18	33	422
US300	300,000	546	300	249,000	1038	4000	850	1/6(2)	"	1140	2.8	16(2)	33	685
US400	400,000	728	400	332,000	1383	5400	1110	1/6(2)	"	1140	3.9	18(2)	33	850

All sizes are available with blowers instead of propeller fans, air deliveries, and other characteristics being the same.

Young Radiator Co.

Dept. 176, **Racine, Wis.**

Sales and Engineering Offices in Principal Cities

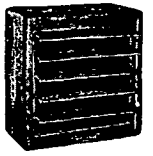
YOUNG

HEAT TRANSFER PRODUCTS

Oil Coolers • Gas, Gasoline, Diesel Engine Cooling Radiators • Heat Exchangers • Intercoolers • Engine Jacket Water Coolers • Unit Heaters • Convectors • Condensers • Evaporators • Air Conditioning Units • Heating Coils • Cooling Coils • and a Complete Line of Aircraft Heat Transfer Equipment.

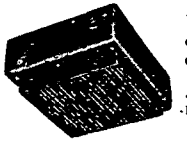
"STREAMAIRE" UNITS

"Streamaire" units are a development of Young's quarter century of experience in building heat transfer products. Personalized, "on-the-job" engineering service by field men—backed by modern research and manufacturing facilities—assures the practical, economical installation as required in your plans.



Type "SH" unit heaters for horizontal air discharge.

Available with capacities from 27,000 to 325,000 Btu per hour. *Catalog 2738.*

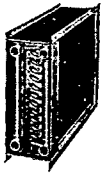


Type "V" or Vertiflow unit heaters for vertical air discharge.

Capacities from 59,000 to 520,000 Btu per hour. *Catalog 2541.*

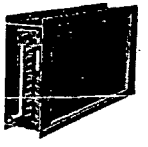
Type "FH" blower unit heaters for floor, wall or ceiling mounting.

Capacities from 163,000 to 1,300,000 Btu per hour. *Catalog 336.*



Type "W" water coils for cooling or heating with central plant systems.

Five widths, 11 to 35 in.; 2 to 8 rows of tubes; many lengths. Write for details.

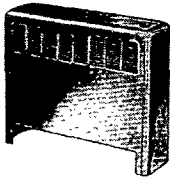


Type "E" evaporator coils for direct expansion cooling systems using Freon or Methyl chloride.

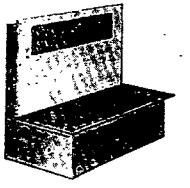
Four widths—two to six rows of tubes; many lengths. *Catalog 5039.*

Several types of convector units—circulate rather than radiate heat. Used with steam or hot water systems.

Styles to blend with architecture and room furnishings.

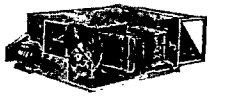


Type "DAC" units provide individual room, winter air conditioning—filter, humidify, heat, circulate. *Catalog 7041*



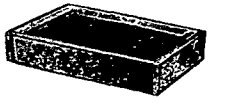
"YAC" units for year-round air conditioning. Also for winter or summer conditioning only.

Horizontal and vertical models. Capacities from 400 to 16,625 cfm. *Catalog 7541.*



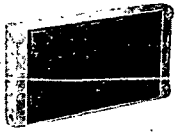
Types "B" and "A" blast coils for central plant heating and air conditioning systems.

Steam distributing tube type available. *Catalogs 4540 and 4542.*



Type "C" commercial heat transfer coils for use in factory built air conditioning units.

Steam distributing tube type available. One, two and three rows of tubes. *Catalogs 4540 and 4542.*



The Trane Company

2021 Cameron Avenue, La Crosse, Wisconsin

COMPLETE LINE OF HEATING, COOLING, AIR CONDITIONING AND AIR HANDLING EQUIPMENT

Over 70 U. S. Branch Offices

Albany, N. Y.; Allentown, Pa.; Amarillo, Texas; Appleton, Wis.; Atlanta, Ga.; Aurora, Ill.; Baltimore, Md.; Billings, Mont.; Birmingham, Ala.; Boston, Mass.; Buffalo, N. Y.; Canton, Ohio; Chattanooga, Tenn.; Chicago, Ill.; Cincinnati, Ohio; Clarksburg, W. Va.; Cleveland, Ohio; Dallas, Texas; Davenport, Iowa; Dayton, Ohio; Denver, Colo.; Des Moines, Iowa; Detroit, Mich.; Dodge City, Kansas; Flint, Mich.; Fort Wayne, Ind.; Gainesville, Fla.; Grand Rapids, Mich.; Greensboro, N. C.; Greenville, S. C.; Harrisburg, Pa.; Hartford, Conn.; Houston, Texas; Indianapolis, Ind.; Kalamazoo, Mich.; Kansas City, Mo.; LaCrosse, Wis.; Lake Charles, La.; Little Rock, Ark.; Los Angeles, Calif.; Louisville, Ky.; Memphis, Tenn.; Mexico, D. F.; Milwaukee, Wis.; Missoula, Mont.; Newark, N. J.; New Orleans, La.; New York, N. Y.; Oklahoma City, Okla.; Omaha, Neb.; Philadelphia, Pa.; Phoenix, Ariz.; Pittsburgh, Pa.; Portland, Ore.; Providence, R. I.; Richmond, Va.; Rochester, N. Y.; Salt Lake City, Utah; San Francisco, Calif.; Seattle, Wash.; Sioux City, Iowa; South Bend, Ind.; Spokane, Wash.; St. Louis, Mo.; St. Paul, Minn.; Syracuse, N. Y.; Trumbull, Conn.; Washington, D. C.; Wilkes Barre, Pa.; Wilmington, Del.

Sales Connections All Over The World

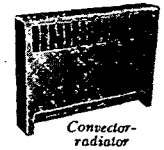
In Canada: TRANE COMPANY OF CANADA, LTD., Mowat and King Sts., W., Toronto, Ont. (15 Branches)

A COMPLETE LINE

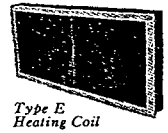
The Trane Company fabricates a complete line of heating, cooling, air conditioning and air handling equipment. Long years of experience with practical knowledge gained from close field contact, have developed products for every requirement.

Trane Convector-radiator—the modern successor to the old-fashioned radiator, the Trane Convector-radiator is a compact, light-weight, easy-to-install unit. Available for either steam or hot water heating system. It is equally suited to the spacious mansion or the smallest cottage, to the modern office, hospital, apartment house or even the industrial building. It combines attractive appearance with long life and economical service.

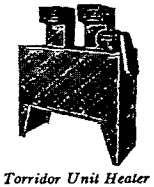
Currently easier to obtain and install than ever before, they are available in stock throughout the nation.



Convector-radiator



Type E Heating Coil

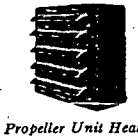


Torridor Unit Heater

Trane Coils—There are Trane Extended Surface Coils for every heating or cooling, comfort or process application, in all types and sizes. Types include coils for steam, hot water or booster heating, direct expansion or water cooling. In addition, a wide variety of coils is available for specialized or process work.

Trane Blower Type Unit Heaters—Better known as Torridors, Trane Blower Type Unit Heaters are available for large space or duct work applications. Manufactured in direct or belt-driven types and floor, ceiling, or wall models, the Torridor may be provided with thermadjust or face and by-pass dampers and a variety of discharge devices for greater flexibility. Ideal for heating large spaces, exposed areas requiring a blanket of heat, and for process applications.

Projection Heater



Propeller Unit Heater

Trane Projection Heaters—A Trane development, the Projection Heater taps the usually wasted heat reservoir at the ceiling bringing it down where it is needed. Installed at 8 to 50 ft heights, it projects warm air to the floor in a circular, vertical air stream. Ideal for low or high pressure systems in factories, warehouses, etc.

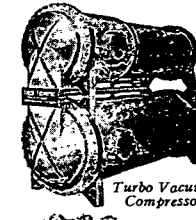
Trane Propeller Unit Heaters—Featuring a quiet operating, wide-bladed fan that pushes rather than bats the heated air through the coil, the Trane Propeller Unit Heater incorporates many unusual features, including adjustable louvers or grilled outlets directing heated air to the floor line, rugged motor supports, attractive appearance.

Trane Air Conditioning Manual—Trane offers the engineering profession a straight-forward and unbiased textbook covering the fundamentals of air conditioning, in a new enlarged edition. The Manual not only shows how to design every type of air conditioning system, but also clarifies underlying principles, enabling both student and engineer to reason out their own problems. Price—$5.00.

Trane Climate Changers—Trane Climate Changer, a unit type air conditioner, is designed for summer, winter, or year 'round air conditioning, commercial and industrial application, comfort or process installations. These units incorporate efficient Trane Coils, quiet blower fans in sturdily constructed casings in horizontal or vertical models. Available in various coil combinations with or without humidification equipment.

Climate Changer

Trane Refrigeration Equipment — Outstanding in the refrigeration field is the Trane Turbo-Vacuum Compressor, a completely self-contained hermetically sealed centrifugal type water chiller, available in 50, 70, 100 and 200 ton sizes. Year 'round efficiency in constant operation with a minimum of maintenance is assured by the scientific simplicity of this machine.

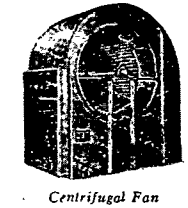


Turbo Vacuum Compressor

Trane also furnishes a complete line of Reciprocating Compressor and Condensing Units with capacities ranging from 3 to 100 tons. Also available are Trane Self-contained Air Conditioners for shop and office spaces.

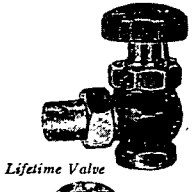
Reciprocating Compressor

Trane Roof Ventilators—The Trane Roof Ventilator can be used wherever an effective roof ventilator is required. This unit is available for either exhaust or supply purposes. It is not only light, but weather and bird-proof as well. Hoods of the units are hinged for easy maintenance.

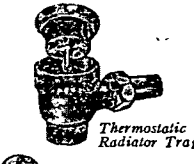


Centrifugal Fan

Trane Centrifugal Fans—Recommended for all types of heating, cooling, and air handling applications. In direct or belt-driven units, single or double widths, and all standard discharges in both backward and forward curved blade construction. Capacities 200 to 330,000 cfm.



Lifetime Valve



Thermostatic Radiator Trap

Trane Propeller Fans—The exclusive broad-bladed propeller fan is available in 2 or 4 blade construction. Can be used for vertical or horizontal installations for exhaust or supply applications. Direct or belt-driven. Capacities, 500 to 100,000 cfm.

Shell & Tube Heat Exchanger

Trane Steam Heating Specialties—There are over fifty valves, traps, vents, strainers, all allied specialties in the Trane Line. Among them are the famous Trane Hermetic Valve with the Lifetime Diaphragm that absolutely prevents steam leakage around the stem and the Thermostatic Radiator Trap which when used together provide an ideal combination for convectors and radiators.

Trane Hot Water Heating Specialties — Included among Trane Hot Water Heating Specialties are the Trane Circulator, Flo Valves, and Fittings. They combine with Trane Convectors or Unit Heaters to provide an ideal Warm Water Heating System for a great variety of applications.

Other Trane Equipment—The complete Trane Line also includes—1. Trane Unit Ventilators for schoolroom air conditioning; 2. Trane Condensation and Centrifugal Pumps for a large variety of uses; 3. Trane Dry Type Water Chillers; 4. Trane Evaporative Condensers to condense refrigerants in the air conditioning system with a minimum use of water; 5. Trane Cooling Towers; 6. Trane Force Flo Unit Heaters for quiet heat and neat appearance; 7. Trane Railroad and Bus Air Conditioning Equipment of all kinds; 8. Trane Shell and Tube Heat Exchangers for cooling and heating vapors or liquids in a closed system; 9. Evaporative Coolers for cooling fluids in a closed system; 10. Transformer Oil Coolers; 11. Air Washers.

Write today for Trane Products Bulletin PB290 which describes completely all of the products listed here as well as providing sufficient data for their selection.

Factories:
NEWARK, N. J.

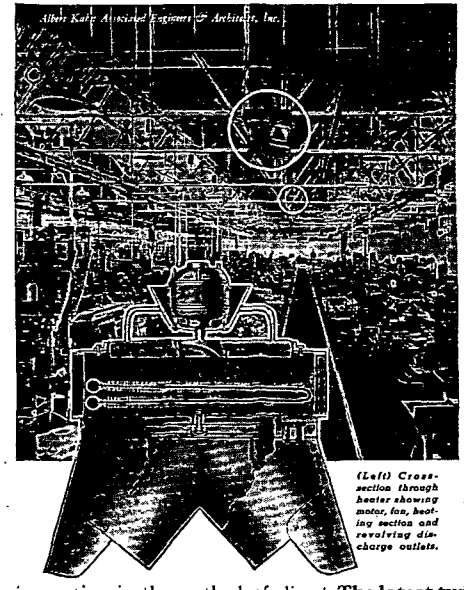
L. J. Wing Mfg. Co.

59 Seventh Avenue, **New York 11, N. Y.**

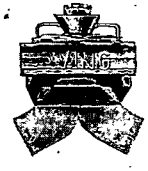
Canadian Factory:
MONTREAL

Branch Offices in **Principal Cities**

WING REVOLVING UNIT HEATERS



(Left) Cross-section through heater showing motor, fan, heating section and revolving discharge outlets.



This innovation in the method of distributing heat produces a sensation in heating comfort never before attained—a sensation of fresh, live, invigorating air.

The fact that the outlets revolve assures uniform and thorough distribution of comfortably warmed air throughout the entire working area, without drafts, hot spots or cold spots.

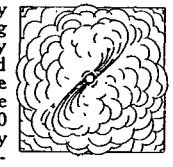
Such an unprecedented high efficiency in distributing heat is the result of nearly 20 years of constant study by Wing engineers to improve on the Floodlight System of heating pioneered by WING in 1921. This method projects the heated air vertically downward by means of light-weight, ceiling-suspended unit heaters.

It has needed only this latest refinement of slowly revolving discharge outlets to bring that method to perfection.

The WING Revolving Discharge type supplements the WING line of standard fixed discharge outlets, illustrated and described on the following page. *Bulletin HR-4.*

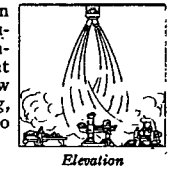
The latest type of WING Unit Heater—with Revolving Discharge Outlets—is just as great a contribution to the art of industrial heating as was the Ceiling-Suspended Unit Heater, announced by WING in 1921.

The area covered by a WING Revolving Unit Heater is slowly swept by the heated air discharged by the outlets which move through an arc of 360 deg. covering every direction of the compass successively.



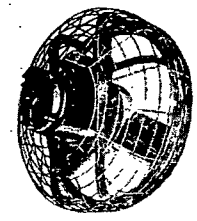
Top View

By maintaining an active, constant circulation of air throughout an industrial plant at all times, a new sensation of refreshing, invigorating comfort to workers is produced.



Elevation

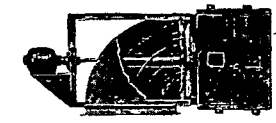
WINGFOIL SAFETY VENTILATING FANS



An axial flow fan that will deliver air against static pressure, quietly and efficiently.

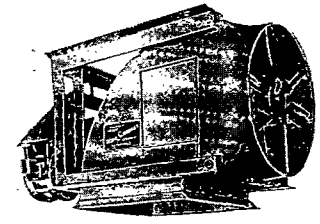
Moves the air forward in straight lines with minimum eddy. Capacities to 100,000 cfm. *Bulletin F-9.*

WING FEATHERFIN PROCESS HEATING UNITS



For manufacturing processes such as drying, aging, etc., requiring the recirculation of the heated air. Motor or turbine located outside air current. *Bulletin P-2.*

WINGFOIL DUCT FANS



For economically moving air wherever ducts are used. It combines the efficient WINGFOIL AXIAL FLOW Fan with a housing which places the motor entirely outside the air duct. Motor and drive remain cool and clean and are easily accessible.

The powerful WINGFOIL Fan delivers high air volume with low power consumption against any pressures for which duct systems should be designed. V-belt or direct drive.

Light, compact and easy to install. *Bulletin F-9.*

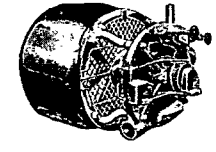
WING SYSTEM OF CONTROLLED COMBUSTION

For low pressure heating boilers and small power boilers. *Increases capacity and permits use of* lowest cost fuel. Includes Type EM Blower equipped with fully enclosed dustproof motor with speed regulating rheostat and automatic control. Eliminates necessity of frequent firing, allowing intervals as great as 24 hours even in zero weather. *Bulletin M-96.*

Installation of Wing System of Controlled Combustion in a large school

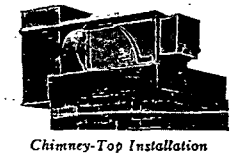
WING TURBINE-DRIVEN BLOWERS

Applied to hand, stoker, oil or pulverized fuel fired boilers, increase boiler capacity, maintain constant steam pressure and permit complete combustion of low-cost fuels. The exhaust steam, free from oil, can be used for heating or processes. *Bulletin T-98.*



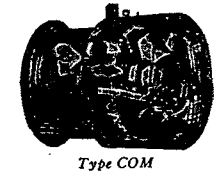
WING DRAFT INDUCERS

Installed in breeching or flue, or on chimney top; provide positive, exact draft regardless of weather conditions or inadequate chimney or breeching construction. Suitable for coal, oil, or gas-fired boilers; industrial furnaces and kilns. *Bulletin I-10.*



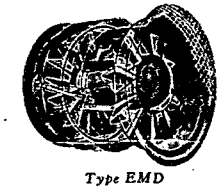
Chimney-Top Installation

WING MOTOR-DRIVEN BLOWERS

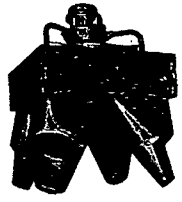


Type COM

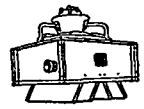
Type COM for static pressures over 5 in. and volumes up to 35,000 cfm. Type EMD for moderate static pressures up to 5 in. Both blowers have fully-enclosed dustproof constant speed motor and built-in adjustable control vanes. Type COM has double-staged axial flow fan; Type EMD, single stage fan. Extremely compact; discharge can be vertical, horizontal or inclined. *Bulletin CO-5.*



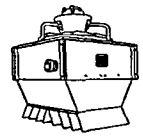
Type EMD



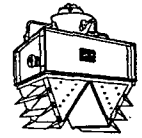
Type "HC" Fixed Discharge Design No. 7



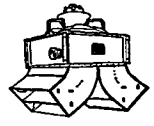
Design No. 1



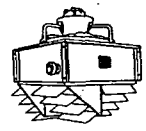
Design No. 1-HV



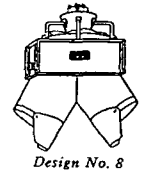
Design No. 3



Design No. 4



Design No. 6



Design No. 8

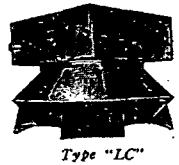
WING FIXED DISCHARGE UNIT HEATERS

The first light-weight, ceiling-suspended, unit heater. Eight different designs of outlets meet the requirements of every type, size and height of building or occupancy. Located near ceiling or roof, the accumulation of hot air in the upper spaces, with the accompanying costly waste of heat, is prevented. They project the air, comfortably warmed, downward to the working area. *Bulletin HR-3.*

DOOR HEATERS GARAGE HEATERS

WING developed this vertical cone-discharge heater in 1921 and today it is still applicable for heating the inrush of cold air at large doorways and for garage heating. Often cuts heating costs in half. *Bulletin HR-4.*

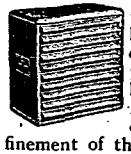
FOR LOW CEILINGS



Type "LC"

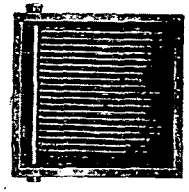
In this type of WING Unit Heater the position of fan and motor are reversed to meet conditions of ceiling or roof height, form and shape of building, coverage, etc. *Bulletin HR-4.*

WING UTILITY UNIT HEATERS



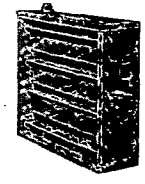
A lightweight suspended unit heater for delivering heated air in one general direction. Has the same powerful fan and rugged heating element as WING Featherweight Unit Heaters. This is the latest refinement of the original horizontal light-weight heater which was developed by WING. *Bulletin HR-4.*

FEATHERFIN HEATER SECTIONS



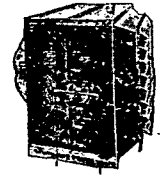
For heating or cooling air for any purpose by steam, hot or cold water or refrigerant. The heating element is extremely light and, for equal heat transfer, offers little resistance to air flow. Available for any desired final air temperature. *Bulletin HS-2.*

VARIABLE TEMPERATURE SECTIONS



Invaluable in supplying fresh air for space heating or process work. Close control of the delivered air temperature is obtained without danger of freezing. Manual or automatic control. *Bulletin HS-2.*

WING INDUSTRIAL FOG ELIMINATORS



Eliminate fog, odor and fumes in dyeing, bleaching and finishing plants, creameries, pasteurizing, bottling, canning and packing plants, chemical works, paper mills, steel pickling plants, etc. No ducts are required. *Bulletin FE-12.*

W. H. Wheeler, Inc.

7 East 47th Street, New York 17, New York

airkem

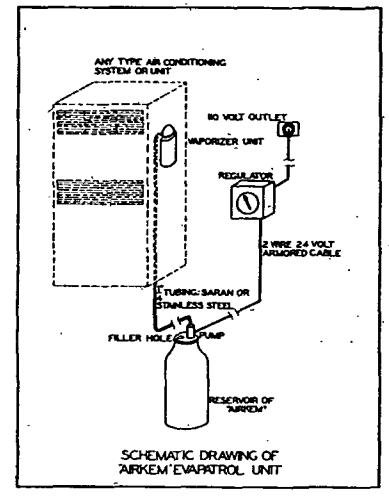
YEARS MARK REG. U. S. PATENT OFF.

CHLOROPHYLL AIR FRESHENER

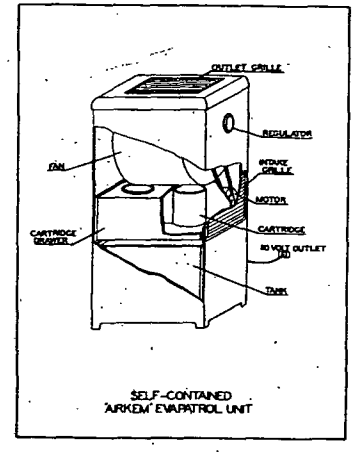
Air Quality Control for Air Conditioning—Airkem Chlorophyll Air Freshener reduces odors and provides a fresh indoor air quality. Manufactured under U. S. Patent 2,326,672, **Airkem** contains activated chlorophyll and is especially suited to industrial and commercial applications.

The efficacy of **Airkem** in producing a fresh indoor air quality has been demonstrated by its use in theaters, department stores, industrial plants, banks, hotels, restaurants, offices and in the most exacting of all fields—hospitals. In addition, **Airkem** permits reduction of the fresh air intake in air conditioning systems with corresponding reductions in both summer and winter load, and savings in fuel, water and electric power.

Easily Added to Systems—It is easy to engineer **Airkem** into new systems or add to existing air conditioning or ventilating systems, either large central systems or the smaller packaged units. The **Airkem** Evapatrol unit consists of a small vaporizing cartridge enclosed in an attractive casing, an **Airkem** reservoir with pump and an **Airkem** regulator. The vaporizing unit can be installed directly on the side of the fan section of a packaged type unit or on the fan section of the plenum chamber of any central system. The suction of blower in either case draws air through the cartridge which is saturated with **Airkem.** The saturation of the cartridge is maintained by means of a solenoid operated pump which is mounted directly on the **Airkem** reservoir, and pumps the **Airkem** to the distributor in the vaporizing unit. The operation of the pump is controlled by the regulator which permits adjusting the amount of **Airkem** by timing the pump strokes at the required intervals. Both the **Airkem** regulator and reservoir can be installed at remote locations.



SCHEMATIC DRAWING OF "AIRKEM" EVAPATROL UNIT



SELF-CONTAINED "AIRKEM" EVAPATROL UNIT

The Air-Maze Corporation

5200 Harvard Avenue, **Cleveland 5, Ohio**

AIR-MAZE

ENGINEERS AND MANUFACTURERS OF AIR FILTERS

DIRECT FACTORY REPRESENTATIVES IN ALL INDUSTRIAL AREAS

DISTRIBUTORS IN PRINCIPAL CITIES THROUGHOUT THE UNITED STATES

Sound Engineering—As specialists in air filtration, Air-Maze has spent over 20 years of constant research in engineering over 3000 types of filters. The enviable reputation enjoyed by Air-Maze—the result of long experience in the air filtration field with varied dirt and dust problems—assures soundly engineered filtration.

Viscous Impingement Type—Air-Maze viscous impingement, cleanable type air filters are permanently efficient in industrial, commercial and residential air conditioning systems. The filter media consists of alternate layers of crimped, galvanized screen cloth of various mesh, arranged to break up the air stream into minute, swirling currents. Dirt is impinged on the wire baffles, previously coated with an adhesive.

Dirt Arrestance—Efficiency of dirt arrestance varies, depending on filter design, type of dust, dust feed, etc. In regular service, practically all so-called "nuisance" or "abrasive" type dust is eliminated.

Resistances — Filters available with initial resistances as low as 0.045 in. water at 300 fpm. (Graphs showing resistance at various velocities are available for each type of filter.)

Face Velocities—Various Air-Maze filters are designed to operate efficiently at any velocity, including 106, 300, 500, 1500, and up to 2000 fpm, depending upon application; pressure drop available, etc.

Sizes—Air-Maze filter panels are rectangular in shape and can be made in any reasonable size and thickness. 1 in., 2 in. and 4 in. thicknesses are standard. Specially shaped filters are built to your specifications.

Holding Frames—Holding frames, complete with felt seal and choice of locking devices, are supplied when requested. For filter banks, Air-Maze furnishes either assembled frames or will drill frames for assembly on the job.

Cleaning Data—Air-Maze filters are easily cleaned with hot water, steam or any commercial solvent. They are charged by immersing in recommended adhesive or S.A.E. 30-50 oil.

Write for Details—Write for complete information and let Air-Maze engineers work with you on your air filtration problem. See your classified telephone directory for Air-Maze factory representatives or distributors.

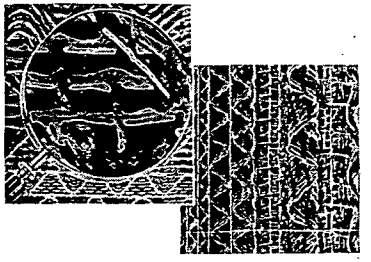
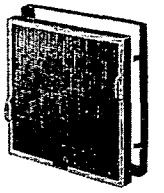


Photo at extreme right shows cross-section of selective filtration principle and large dirt holding capacity in typical Air-Maze filter panel.

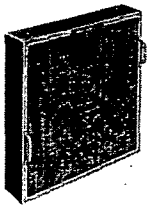
In the left-hand photo, a magnifying glass shows the smooth round wire baffles collecting dirt in "streamlines", thus presenting low resistance to air flow despite heavy collection.

The Air-Maze Corporation

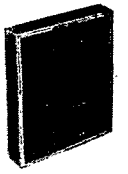
5202 Harvard Avenue, **Cleveland, Ohio**



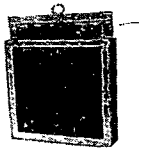
TYPE "A" with holding frame



TYPE "B"



Humidifier, P-5 (4")



Oil Separator (R66-B)

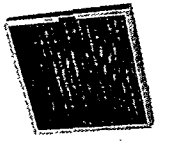
Type "A"—Built for heavy-duty service. Scientific fabrication of media results in open construction, large dirt-holding capacity, without undue increase in restriction.

Type "B"—Recommended for average dirt conditions, or where shorter servicing intervals are acceptable.

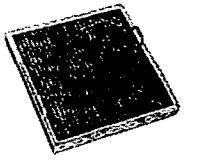
Kleenflo*—For home unit conditioners and recirculated air systems.

Greastop*—Prevents grease nuisance and fire hazards in kitchen ventilating ducts, and reduces blower and motor maintenance by collecting entrained grease in exhaust cooking vapors. Greastops eliminate duct cleaning expense.

*Trade Mark Registered



Kleenflo

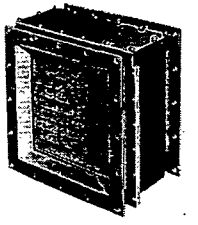


Greastop

SPECIAL FILTERS FOR SPECIAL APPLICATIONS

Humidifier—Increases humidity for many industrial processes where dry atmosphere may be harmful or present a fire hazard. Effective for evaporative cooling installations. Panels also available for separation of liquids from air stream. Bronze construction.

Flame Arrester—Panel open enough to permit air to pass freely, yet will prevent explosion on one side from igniting air-gasoline mixture on other side. Excellent for mines, grain elevators, liquid storage installations, etc.

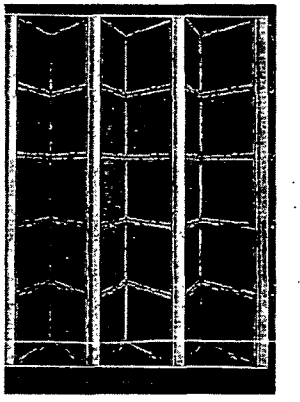


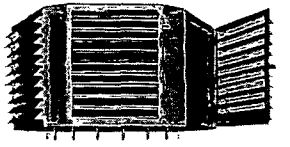
Flame Arrester

Oil Separator—Removes entrained oil, kerosene and other liquids from air. Oil collecting media is removable. Filters can be applied to exhaust systems in connection with coolants, cutting oils, degreasers, etc.

TYPICAL APPLICATIONS

Air-Maze panel filters arranged to save space with a maximum amount of air flow.





A "doghouse" installation for compressors and engines using Air-Maze panel filters. Furnished with or without weather louvres. One louvre is shown open.

AMERICAN AIR FILTER COMPANY INC.

673 Central Avenue, **Louisville 8, Ky.**

In Canada: Darling Brothers, Ltd., Montreal, Quebec

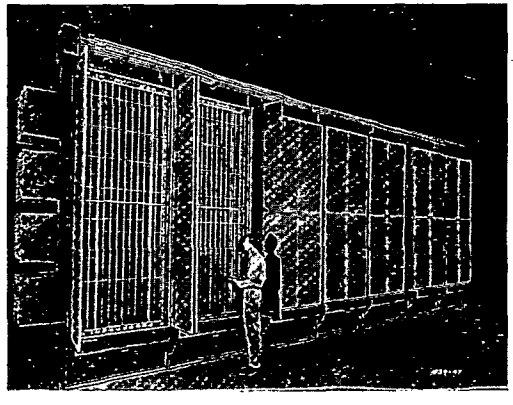
PRODUCTS: The American Air Filter Company, Inc. manufactures a complete line of Air Filtering Equipment including Electronic Air Filters, Automatic Self-Cleaning Filters, Viscous Unit Filters, and Dry Process Filters to be used for the removal of dust, soot, smoke, dirt, bacteria and other foreign matter from the air.

Because air filtration has proved to be the economical, practical and efficient method of cleaning the air, the modern air filter is now considered an absolute necessity in building ventilation or air conditioning, in the maintenance of health and personal efficiency, protection of interiors and furnishings and valuable merchandise and equipment. Shown here are but a few American Air Filter products in most general use.

The American Air Filter Company is recognized as an authority on dust problems and their satisfactory solution. Products offered embody the knowledge accumulated from twenty-five years of intensive research devoted exclusively to the study of dust problems and the development of air cleaning apparatus; the experience gained from designing, building and applying thousands of air filters; are backed by ample technical and financial resources; and may be relied upon as the most modern equipment in their field of service.

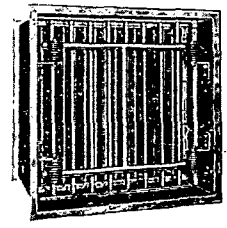
THREE TYPES OF ELECTRONIC AIR FILTERS

For more than ten years, research and experimentation with electronic air filtration have been in progress by AAF engineers. Today's complete line includes the self cleaning Electro-Matic introduced in 1939, the washable Electro-Cell with removable collector plates, and the Electro-Airmat with the replacable Airmat paper medium. Here for the first time is high efficiency air cleaning in three types of electronic filters to meet any requirements for super clean air.

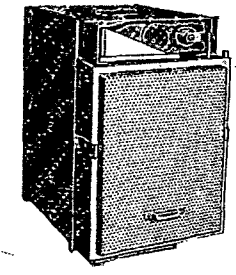


Electro-Matic Self-Cleaning Electronic Filter—The Electro-Matic filter is a self cleaning electric precipitator combining the most advanced principles of electronic air cleaning with notable improvements in construction details and method of operation. The self-cleaning feature is an exclusive advantage of the Electro-Matic filter. 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.

Electro-Cell Electronic Filter—Incorporates all of the new developments of major importance in electronic air filtration using collector plates. Installation has been simplified, performance improved and maintenance advantages provided. Built in vertical sections of two widths—2 ft and 3 ft over-all. Collector plate assemblies are removable; ionizers are hinged and extend the full height of the sections, preventing current loss. A choice of washing collector plate assemblies while in place, or removing assemblies for individual cleaning. Installation is simplified because filter is built in vertical sections rather than assembled of small units. Parts to be lifted and aligned are light in weight and electrical connections simplified. Write for Bulletin No. 252.

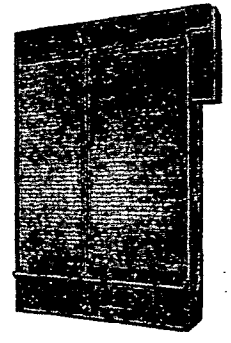


Electro-AIRMAT Filter



Electro-CELL Filter

Electro-Airmat Electronic Filter—The application of an electrostatic charge to a dielectric filtering material is an exclusive AAF research development which began early in 1934. Airmat paper is composed of a number of plys of porous tissue-like cellulose sheets. When electrically charged the plys tend to separate and each individual fibre becomes a collecting electrode which attracts and holds the dust and smoke particles. Ease and convenience of maintenance is a desired advantage of the Electro-Airmat. Requires neither water nor sewer connections for cleaning, nor spraying with oil to maintain its efficiency. When Airmat paper has accumulated its dust load it is removed and replaced with clean material by means of a mechanical loader. In case of power failure, filter media provides best mechanical air filtration known. Send for Bulletin No. 253.



American Multi-Duty Self-Cleaning Filter

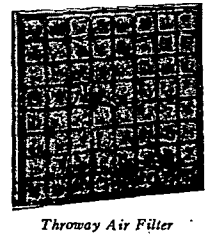
American Multi-Duty Automatic Filter provides outstanding features of performance and design and will accomodate either armored screen panels or die stamped louver panels available in three types. Offers advantage of uniformly constant air supply, fixed operating resistance and automatic operation. Ideal for ventilation and air conditioning service. Available in any size or capacity. Send for Bulletin No. 241-A.

AAF UNIT FILTERS

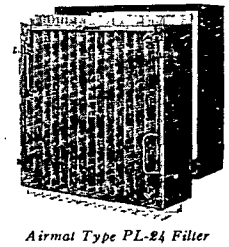
Throway Air Filter—Throway filters are inexpensive and designed to be discarded after accumulating dust load. Send for Bulletin No. 117-E.

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-C.

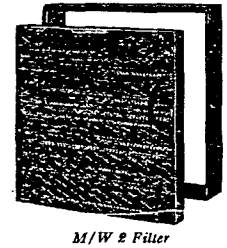
M/W Filters—The M/W comes in 2 in. and 4 in. thicknesses. Ideally suited to air cleaning problems encountered in general ventilation and commercial air conditioning. Permanent type, washable. Send for Bulletin No. 202.



Throway Air Filter



Airmat Type PL-24 Filter



M/W 2 Filter

W. B. CONNOR ENGINEERING CORP.

114 East 32nd Street
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Canadian Representative: Arthur S. Leitch Co., Ltd., Toronto, Ont.

Air Recovery **DOREX** **Equipment**

DOREX Activated Carbon Adsorbers recover the freshness of vitiated air by extracting air-borne odorous, gaseous and vaporous impurities. When applied to recirculated air in air conditioning, DOREX Adsorbers convert used, stale, *conditioned* air for ventilation thus reducing to a minimum the volume of *unconditioned* outdoor air make-up otherwise necessary.

With DOREX Air Recovery, the cooling and heating capacity of a proposed air conditioning system is decreased by the saving in outdoor air load.

With DOREX Air Recovery, an existing air conditioning system will serve a larger space or satisfy a greater conditioning load without increase in cooling or heating equipment or the consumption of more fuel or energy.

The highly active, specially processed and impregnated carbon employed will remove from the air passed through it and retain 95 per cent of all entrained gaseous impurities and maintain approximately this efficiency for from 6 months to 2 years depending upon the air contamination. Upon exhaustion the carbon may be reactivated for re-use.

DOREX Air Recovery equipment is available in several types to suit individual requirements.

TYPE H—FOR COMPLETE DECONTAMINATION OF ALL AIR PASSED THROUGH IT

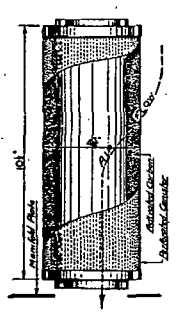


Fig. 1

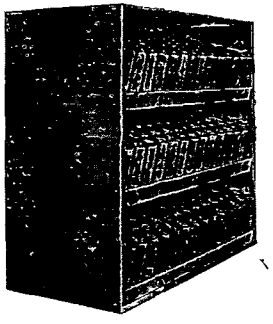


Fig. 2

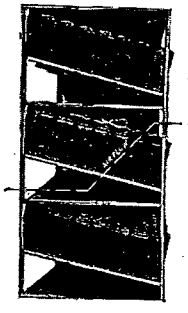


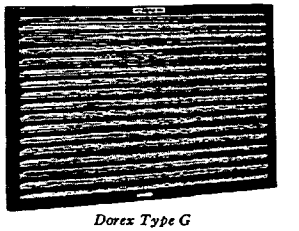
Fig. 3

DOREX Type H Equipment consists of light-weight, removable, perforated, activated carbon filled, adsorption canisters. These are mounted in multiple on one or more supporting manifold plates in such manner that all air to be treated will pass uniformly through the granular carbon media. The assembly arrangement is flexible to suit the space limitations. The resistance to air flow averages only 0.15 in. wg.

Fig. 1 shows a typical canister. It is closed at the top and the inner cylinder is open at the bottom, which opening registers with a corresponding hole in the supporting manifold plate. Fig. 2 is a photograph of a typical arrangement of canisters as installed. In this instance, three manifold plates each support 98 canisters arranged in 7 rows of 14 canisters each. Fig. 3 is a sectional view of this arrangement with the side removed and indicating direction of air flow. Each canister decontaminates from 25 to 35 cfm of air.

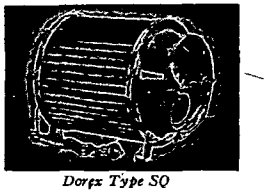
TYPE G—FOR CONTINUOUS DECONTAMINATION OF RECIRCULATED AIR

The DOREX Type G Adsorber and Air Recovery Unit is designed to remove accumulated odors and gaseous impurities from air recirculated by air conditioning units and systems. Its extremely compact design and the ease with which it can be installed makes it equally adaptable both to existing ventilating systems without necessitating expensive alterations, and to newly designed conditioning systems, particularly where space is at a premium. It is also adapted to incorporation in *package* conditioners, unit heaters and cold diffusers.



Dorex Type G

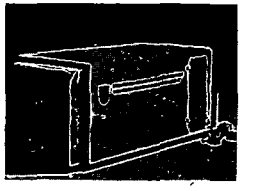
The Type G Adsorber consists of sturdy metal frames forming panels, each housing a battery of exposed perforated metal tubes which contain the granular carbon filter media. These panel units are available in stock sizes, suitable for arrangement in air ducts, or for attachment to standard dust filters, outlet or inlet grilles. They are designed to expose a maximum of carbon adsorption surface to the air stream with a minimum of resistance to air flow. Standard units are of one, two or three tube rows in depth, designated as G-1, G-2 and G-3, respectively.



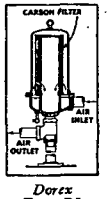
Dorex Type SQ

The DOREX Type SQ Odor Adsorber is an entirely self contained unit suitable for either portable or stationary installation. It consists of a cage of closely spaced, perforated, granular activated carbon-filled tubes housing a quiet operating, circulating fan and motor. It is furnished assembled, equipped with starting switch and complete with extension cord and plug ready to connect to electric socket. **The DOREX Type SQ Unit** keeps the room air in continuous motion while, at the same time, completely decontaminating one-third of the total air circulated. It, therefore, provides both air circulation and ventilation without the use of outdoor air.

At the right is an illustration of the **Type A-100-B,** smallest of the DOREX self-contained *package* recirculating Odor Adsorbers. In its attractive enameled wood cabinet are contained a dust filter, four carbon gas adsorbing canisters, circulating fan and motor. It has a host of practical uses—in homes, offices, doctor's rooms, walk-in refrigerators, etc.



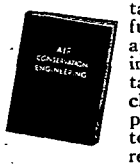
Dorex Type A



Dorex Type PL

The Type PL DOREX Vapor and Gas Adsorber is designed especially to extract all vapors, fermentation odors and other gaseous impurities from compressed air. It is the only device which effectively removes air-entrained gaseous odors and impurities not eliminated by commercial filters, separators, after-coolers or receivers.

"AIR CONSERVATION ENGINEERING"



A complete and authoritative Text Book on the function, engineering and application of Air Recovery in air conditioning. Contains valuable tables and charts. Price $2.00. Complimentary copies available to recognized engineers upon request.

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Air Filters for Building Ventilation, Air Conditioning, Engine Intakes, Etc.

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STAYNEW FILTERS

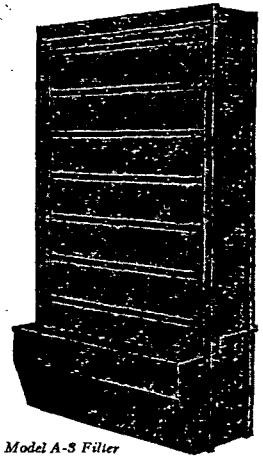
STAYNEW MODEL A-3 AUTOMATIC FILTER

A New and Improved High Efficiency Automatic Air Filter

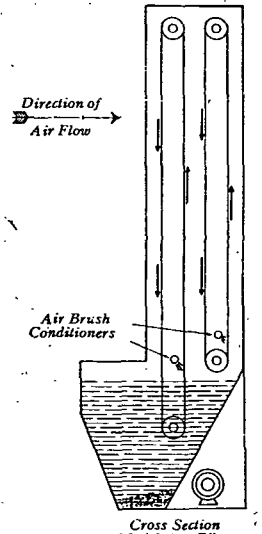
An endless curtain type oil-bath filter outmoding the previous *Model A* design. In addition to offering all exclusive features of the earlier model, the new A-3 maintains higher efficiency, greater dirt-handling ability, improved air brush conditioning and other qualities. Meanwhile the original low resistance has been retained.

Operation and Features: Two endless curtains are carried on heavy roller chains which are driven by sprockets keyed to the shafts of the curtain rollers. These rollers float on ball bearings for quiet, frictionless operation. Curtains consist of removable panels made of a single layer of bronze screen cloth to which are attached layers of woven copper mesh. The first of the curtains is the denser, having about twice the impingement surface of the second or rear curtain. This first curtain acts as the filter and travels through the oil reservoir. The second curtain does not enter the reservoir but acts only as a safeguard against oil entrainment. This design permits the front curtain to be rotated so that the panels on the air-entering side move downward into the oil bath and are *cleaned* before rising on the return or clean side. Thus no dust can be carried across the back or return side of the front curtain to be blown off and carried on by the flow of air.

Patented Staynew Air Brush Conditioners prevent excessive amounts of oil being carried upward on the curtain panels and being entrained in the air stream. This is *important*, since natural drainage of even the finest grades of filter oil cannot be depended upon due to changes in viscosity.



Model A-3 Filter

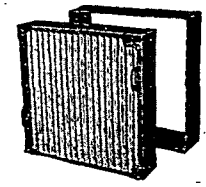


Cross Section Model A-3 Filter

Specifications

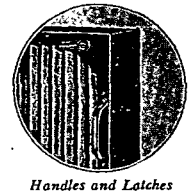
The Model A-3 filters are sectional and may be bolted together to obtain any required capacity. Sections come in two widths: 4 ft 3 in. and 2 ft 9 in. The filter drive and control mechanism includes a ⅙ hp motor driving through a reduction gear, a Telechron Time Switch, and a momentary contact by-pass switch (providing manual control when required)—all mounted on a common base plate located on clean air side of filter. The drive motor and the Air Brush Conditioners operate simultaneously for a few seconds at 15-minute intervals. Each control mechanism handles up to and including 4 filter sections.

STAYNEW PANEL TYPE FILTERS



Model WKE Panel and Frame

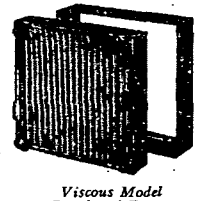
New Model WKE: Dry-type finned panel filter for use in ventilation and air conditioning systems. Extremely large filtering area in relation to overall size. Adaptable to wide variety of filtering media—in fact, almost any medium obtainable in sheet form that can be crimped. Steel mesh on both sides of medium prevents sagging and makes the WKE *fire-resistant* 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.



Handles and Latches

Filter cells are held in rigid box-type supporting frames of heavy gauge 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.

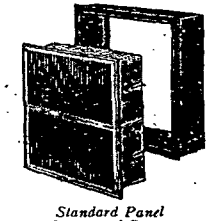
Frames are drilled so that they can be riveted together to form a flat bank, or by the addition of angle uprights into a "V" or staggered arrangement.



Viscous Model 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 above. Also, frames are drilled as in Model WKE.

Both Model WKE and DPV cells are furnished in 2 in. and 4 in. depths in various standard sizes.



Standard Panel Insert and Frame

Standard Panel: Dry-type Fin Construction, high filtering efficiency. Heavy steel Panel Insert and Frame. Cleanable. Forty-two square feet of filtering area. Thousands of Staynew Standard Panel Units have given satisfactory service for years. Available in 20 in. x 20 in. x 7 in. size only.

Representatives in Principal Cities

Complete Information on Request

FILTERS ALSO MADE FOR INTERNAL COMBUSTION ENGINES AND COMPRESSORS, AIR AND LIQUID LINES, ETC.

Badger CORPORATION

Milwaukee 2, Wisconsin

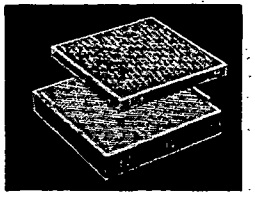
Manufacturers of PERMANENT and Replaceable AIR Filters for Heating, Ventilating, Air Conditioning, Aircraft and Industrial Use

Badger designers and engineers have contributed many of today's most valuable developments in air filtration, dust, grit, and grease elimination in a wide variety of industries. BADGER Air Filters are well known for their high efficiency, low air resistance, greater dust-holding capacity, and reduced maintenance cost on an increasing range of modern applications.

BADGER Type P Air Filter . . . For Air Conditioning and Ventilating Systems

Permanent type, easy to clean, sturdy and strong, with special BADGER filter mesh design that holds dust through complete depth while providing minimum air resistance. Dual purpose expanded metal screen deflects air stream and protects filter media. Vents in frame aid in quick cleaning, thorough draining. Standard 1 in. and 2 in. sizes from 640 C.F.M. to 1000 C.F.M. Special sizes to meet all applications. APPROVED BY UNDERWRITERS' LABORATORIES, Inc.

BADGER Type HD Industrial Filter for Extra Heavy Duty and Industrial Air Cleaning Services



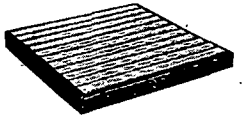
Specially designed for use on industrial applications, Badger Type HD is made in 2 in. and 4 in. sizes, and capacities from 640 cfm to 1000 cfm. Permanent type, with high dust-holding capacity, effective, sturdily constructed filter media protected by deeply crimped layers of expanded metal screen, and heavy arc welded frame.

BADGER HOLDING FRAMES

Badger's special "Slip-Groove" holding frame design provides greater filtering capacity with a minimum of space. Compact, sturdy, extremely strong and rigid, this frame construction also simplifies filter removal and cleaning. Available in V-type, in straight banks, or adaptable to special requirements.

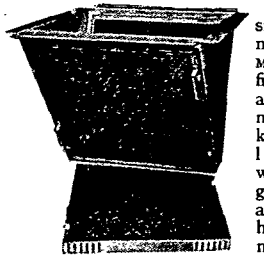
BADGER REPLACEABLE AIR FILTER

for domestic and commercial forced warm air and air conditioning units.



A new, dry replaceable filter. Media is odorless, fire-resistant glass-fibre. Ribbed construction on both sides of filter insures complete air filtration with 25% greater filtering surface.* Long life assured by sturdy cardboard frame cemented to rib structure. Standard 1 in. and 2 in. sizes.

BADGER Grease Filter



A new design in all-metal PERMANENT type filter that is an absolute necessity in kitchens, galleys, and wherever grease-laden air is a fire hazard and a maintenance problem. Filter media assures greater efficiency in air circulation and ventilation, increased working comfort and safety, and protection to motors, brushes and mechanical equipment. Available in 2 in. thickness, adaptable to a wide variety of applications.

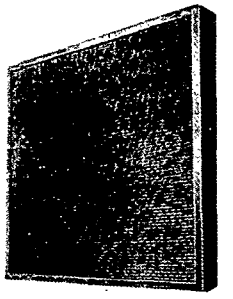
Complete bulletins, specifications, test and performance charts may be obtained on all BADGER Permanent and Replaceable Filters, as well as ready engineering assistance on special problems.

Farr Company

Manufacturing Engineers

Los Angeles • California

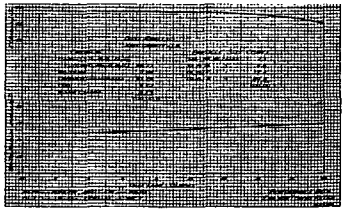
FAR-AIR FILTERS* FOR ALL TYPES OF INDUSTRIAL USE



FAR-AIR STANDARD PANEL FILTER
For Ventilation For Grease

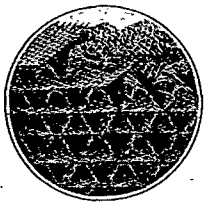
Face velocity in f.p.m.	Capacity in c.f.m. per sq. in. of face area	Starting static pressure	C.F.M. CAPACITIES OF STANDARD FILTERS BASED ON VELOCITY USED			
			16"x20"	16"x25"	20"x20"	20"x25"
346	2.40	.06"	625	795	800	1020
390	2.70	.07"	700	895	900	1145
433	3.00	.09"	780	995	1000	1275
476	3.30	.10"	860	1095	1100	1400
519	**3.60**	**.12"**	**935**	**1195**	**1200**	**1530**
563	3.90	.14"	1015	1290	1300	1655
606	4.20	.16"	1090	1390	1400	1780
650	4.50	.19"	1170	1490	1500	1910
693	4.80	.20"	1250	1590	1600	2035

Based on our recommended face velocity of 519 ft per minute, Far-Air filters deliver 50% more air with lower pressure drop. Filters are all metal, permanent construction . . . Installation of Far-Air Filters, effects a saving not only in initial costs but, maintenance as well—you need only ⅔ of the filter area, with a Far-Air filter.



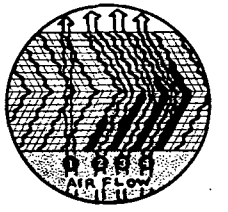
***CERTIFIED PERFORMANCE**

You can depend on Far-Air Filters to deliver the performance claimed for them . . . Farr *Certifies* filter performance data.



HERRINGBONE CONSTRUCTION

Low Pressure Drop . . . More Air Delivery . . . High Efficiency Large Dust Holding Capacity . . . Easy Cleaning—these Far-Air features are the result of Farr's radically different crimped wire screen filter construction. Of paramount importance to heating and refrigeration engineers is the fact that Farr Filters provide a filter area equal to the fin coil area. This special feature eliminates clumsy V-shaped additions on vent ducts usually necessary with other type filters.



PROGRESSIVE LOADING

As entering orifices of Far-Air Herringbone Construction Filters are loaded, air direction changes, flowing past the front loaded surface and progressively loading the clean screen mesh that remains. Greater free area permits a larger dust load with lower pressure drop. The slight difference in pressure drop when a Far-Air Filter is loaded from when it is clean maintains a constant balanced air flow in the air distribution system.

For full information about other types of FAR-AIR FILTERS, write: FARR CO., Dept. HVG, 2615 South West Dr., LOS ANGELES 43, CALIF.

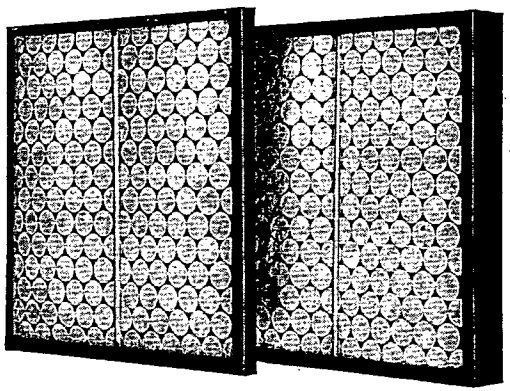
Owens-Corning Fiberglas Corporation

Toledo 1, Ohio

DUSTOP* AIR FILTERS

—a FIBERGLAS* product

*T.M. REG. U.S. PAT. OFF.



Dust-Stop Air Filters are replaceable impingement-type filters for use in all systems in which air is moved mechanically—central heating, ventilating and air-conditioning systems, and forced-warm-air furnaces.

Dust-Stops provide high air filtering efficiency. They are constructed of packs of glass fibers (Fiberglas*), coated with an adhesive, faced with a metal grille, and bound on the edge with a fiberboard frame.

The Fiberglas fibers, packed to proper density, form an exceptionally effective medium for air filtration. Being glass, they are inorganic, chemically stable, resistant to heat and corrosive vapors. And being of glass, they do not absorb the nonodorous, nonevaporating adhesive with which they are coated. Each impinged particle of dust is quickly soaked, acting as a wick to carry adhesive to other particles. Thus, the adhesive remains effective until the filter is so heavily loaded with dust that resistance to air flow calls for replacement.

A minimum of manpower and time is required in replacing economical Dust-Stops—and they can be obtained quickly from near-by suppliers. Dust-Stops are made in two standard types: No. 1 (1 in. thick) and No. 2 (2 in. thick). Both are available in several sizes.

Owens-Corning Fiberglas Corporation Offices

ATLANTA, GA.
BOSTON, MASS.
BUFFALO, N. Y.
CHICAGO, ILL.
CINCINNATI, OHIO
CLEVELAND, OHIO
DALLAS, TEXAS
DETROIT, MICH.
LOS ANGELES, CALIF.
NEW YORK, N. Y.
PHILADELPHIA, PA.
PITTSBURGH, PA.
ST. LOUIS, MO.
SAN FRANCISCO, CALIF.
SEATTLE, WASH.

DUSTOP AIR FILTERS AND FRAMES

TWO TYPES OF FRAMES—

Dust-Stop Air Filters may be installed in banks of either "L" Type or "V" Type Dust-Stop Air Filter Frames. Both types of frames are designed, patented and manufactured by Owens-Corning Fiberglas Corporation for the convenient handling of Dust-Stop Air Filters.

The choice between "L" and "V" Type frames is determined by the frontal area available. The "L" Type frame requires less depth within the duct or plenum chamber but takes a larger face area than the "V" frame for the same cfm capacity. However, it can be set in various arrangements that reduce the required face area by increasing the depth.

The "L" Type frame two filters deep is designed to hold two No. 1 Dust-Stop Air Filters in each cell. The "L" Type four filters deep holds four No. 1 Dust-Stop Air Filters (20 in. x 20 in.). "L" Type frames can be provided in any size from a single unit having one cell at a rated capacity of 800 cfm at a velocity of 300 lineal feet per minute to a unit consisting of 91 cells with a capacity of 72,800 cfm.

The "V" Type frame contains two "cells" each forming one side of the "V." Each cell will hold up to four No. 1 or two No. 2 Dust-Stop Air Filters. "V" Type frames take only the 20 in. x 25 in. filters. The "V" frames are available in any size from a single unit having a capacity of 2000 cfm to a 98 cell unit with a capacity of 98,000 cfm.

All frames are 13-gauge cold rolled steel and are shipped knocked down and crated, complete with all parts and instructions necessary for easy and rapid assembly.

"L" Type Frame

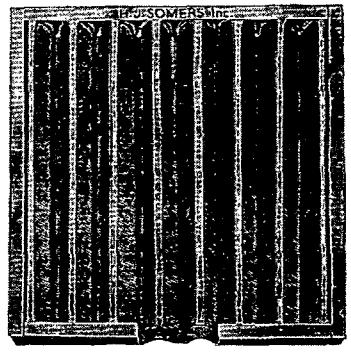
"V" Type Frame

H. J. Somers, Inc.

6063 Wabash Ave., **Detroit 8, Mich.**

Agents in All Principal Cities

SOMERS Heavy Duty Industrial Filter



All Welded Vee Type
Patent No's. 2008800, 2130107

Somers Hair Glass Filters provide everything required in an efficient air-cleaning system.

Consider These Features:

- **High rating for dust, soot and bacteria separation.**
- **Require no adhesive, coating or impregnation.**
- **Indestructible in normal service.**
- **Minimum low-pressure drop.**
- **Odorless and non-absorptive.**
- **Fireproof.**
- **Washable.**
- **Permanent—Do not rot nor disintegrate.**
- **All welded zinc-plated 20 ga. steel frame.**
- **Metal protection strip on apex.**
- **Glass cloth between hot-dipped hardware cloth.**
- **Glass ribbon seal so air cannot short circuit.**

Somers Hair Glass Filters consist of a 20 gauge hot galvanized frame holding galvanized wire cloth packed with hair-spun glass strands. The glass strands are flexible, do not break up and cannot be drawn into air stream.

Hair Glass being chemically inert, has no facility of absorption; it cannot rust and lasts indefinitely in service. Water either hot or cold may be used to clean it, without impairing its efficiency.

These filters eliminate the necessity, the expense and the inconvenience of periodic replacement.

SOMERS WASHABLE AIR FILTERS All Welded Vee Type Stock Sizes

Frame Size Height and Length	Frame Depth	Filter Surface Square Inches	For Average Dry Filter Installations	Wet Application
8″ x 12″	3 1/8″	288	288 C.F.M.	144 C.F.M.
12″ x 12″	2 7/8″	288	288 C.F.M.	144 C.F.M.
12″ x 20″	3 1/8″	720	720 C.F.M.	360 C.F.M.
15 1/2″ x 24 1/2″	3 5/16″	1023	1023 C.F.M.	511 C.F.M.
15 1/2″ x 24 1/2″	3 1/8″	1674	1674 C.F.M.	837 C.F.M.
15 5/8″ x 24 7/8″	2	480	480 C.F.M.	240 C.F.M.
15 5/8″ x 24 7/8″	3 1/8″	1110	1110 C.F.M.	555 C.F.M.
16″ x 20″	2	384	384 C.F.M.	192 C.F.M.
16″ x 21 1/2″	3	816	816 C.F.M.	408 C.F.M.
16″ x 25″	2	480	480 C.F.M.	240 C.F.M.
16″ x 25″	2 1/2″	624	624 C.F.M.	312 C.F.M.
16″ x 25″	3″	864	864 C.F.M.	432 C.F.M.
16″ x 25″	3 1/8″	1344	1344 C.F.M.	672 C.F.M.
16″ x 25″	3 1/8″	1440	1440 C.F.M.	720 C.F.M.
16″ x 25″	3 1/8″	1632	1632 C.F.M.	816 C.F.M.
16″ x 25″	3 1/8″	1056	1056 C.F.M.	528 C.F.M.
18″ x 18″	3 5/16″	864	864 C.F.M.	432 C.F.M.
18″ x 18″	3 1/8″	1134	1134 C.F.M.	567 C.F.M.
18″ x 24″	3″	1080	1080 C.F.M.	540 C.F.M.
19 1/2″ x 19 1/2″	2″	480	480 C.F.M.	240 C.F.M.
19 1/2″ x 19 1/2″	3″	819	819 C.F.M.	409 C.F.M.
19 1/2″ x 19 1/2″	3″	936	936 C.F.M.	468 C.F.M.
19 1/2″ x 19 1/2″	3″	995	995 C.F.M.	497 C.F.M.
19 1/2″ x 19 1/2″	3 1/8″	1053	1053 C.F.M.	526 C.F.M.
19 1/2″ x 19 1/2″	3 1/8″	1170	1170 C.F.M.	585 C.F.M.
19 1/2″ x 19 1/2″	3 5/16″	1696	1696 C.F.M.	848 C.F.M.
20″ x 20″	2″	480	480 C.F.M.	240 C.F.M.
20″ x 20″	2 1/2″	600	600 C.F.M.	300 C.F.M.
20″ x 20″	2 3/4″	780	780 C.F.M.	390 C.F.M.
20″ x 20″	2 3/4″	840	840 C.F.M.	420 C.F.M.
20″ x 20″	3″	960	960 C.F.M.	480 C.F.M.
20″ x 20″	3 5/16″	1020	1020 C.F.M.	510 C.F.M.
20″ x 20″	3 1/8″	1200	1200 C.F.M.	600 C.F.M.
20″ x 20″	3 1/8″	1320	1320 C.F.M.	660 C.F.M.
20″ x 20″	3 1/8″	1680	1680 C.F.M.	840 C.F.M.
20″ x 25″	2″	600	600 C.F.M.	300 C.F.M.
20″ x 25″	2 7/8″	1020	1020 C.F.M.	510 C.F.M.
20″ x 25″	3 1/8″	1560	1560 C.F.M.	780 C.F.M.
20″ x 25″	3 5/16″	1800	1800 C.F.M.	900 C.F.M.
20″ x 30″	3 1/8″	1800	1800 C.F.M.	900 C.F.M.
20″ x 30 1/2″	3 5/16″	2400x	2400 C.F.M.	1200 C.F.M.
23″ x 20″	3 1/8″	1656	1656 C.F.M.	828 C.F.M.
23 1/2″ x 23 1/2″	3 1/8″	1621	1621 C.F.M.	810 C.F.M.
23 3/4″ x 17 3/4″	3″	1068	1068 C.F.M.	534 C.F.M.
24″ x 25 1/2″	3 1/8″	1872	1872 C.F.M.	936 C.F.M.
25″ x 20″	3 1/8″	1800	1800 C.F.M.	900 C.F.M.
26″ x 23 1/2″	2 1/2″	936	936 C.F.M.	468 C.F.M.
26″ x 23 1/2″	2 1/2″	936	936 C.F.M.	468 C.F.M.
26″ x 34″	3 1/8″	2652	2652 C.F.M.	1326 C.F.M.
28″ x 33 1/2″	2 1/2″	1428	1428 C.F.M.	714 C.F.M.
29″ x 33 1/2″	3 1/8″	3045	3045 C.F.M.	1520 C.F.M.
30″ x 15″	3 5/16″	1800	1800 C.F.M.	900 C.F.M.
30″ x 20″	3 1/8″	1800	1800 C.F.M.	900 C.F.M.
30″ x 24″	3″	1800	1800 C.F.M.	900 C.F.M.
31″ x 23 1/4″	3 1/4″	3162	3162 C.F.M.	1551 C.F.M.

Other sizes also available. Send for complete stock size list.
Frames zinc plated for 100 hour salt water spray test. Refill may be inserted if necessary.
Quotations and further engineering data, including master holding frame drawings will be sent on request.

Just a *few* users of Somers Filters

Chemical Plants
American Viscose Co.
American Zinc & Chemical Co.
Davison Chemical Corp.

Automotive
Frederick Sterns Co.
Cadillac Motor Car Co.
Chevrolet Motor Car Co.
Chrysler Corp.
Fisher Body Corp.

Refrigeration and Air-Cond.
Frigidare Corp.
Norge Div.—Borg Warner
Kelvinator Corp.
York Ice Machine Co.

Ships
Amer. Shipbuilding Co.
U. S. S. Saratoga
U. S. N. Lake City, Fla.
U. S. N. Daytona Beach, Fla.
U. S. N. Vero Beach, Fla.
U. S. N. Jacksonville, Fla.

Utilities and Municipalities
City of Kenosha
Michigan Consolidated Gas Co.
Detroit Edison Co.
New York Edison Co.
Westchester Lighting Co.

Dep't. Stores
S. S. Kresge Co.
S. H. Kress & Co.

Food Processing
Awrey Bakeries
Gilbert Chocolate Co.
Kellogg Co.

Manufacturers
Buffalo Forge Co.
Burroughs Adding Machine Co.
Clarage Fan Co.
Curtiss-Wright Airplane Co.
Glensder Textile Co.
Hoover Co.
International Heater Co.
Kearney & Trecker Corp.
Killian Mfg. Co.
National Carbon Co., Inc.
Pittsburgh Plate Glass Co.
Rockford Machine Tool Co.
Sunstrand Machine Tool Co.

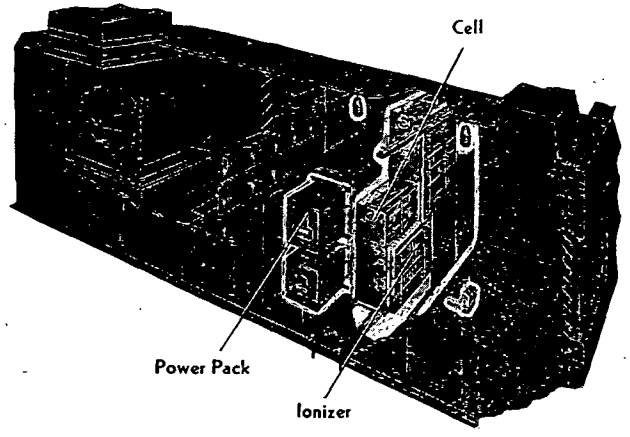
B. F. Sturtevant Company

Division of Westinghouse Electric Corporation

Hyde Park, Boston, Mass.

PRECIPITRON*

Westinghouse Precipitron—the electronic air cleaner—now makes possible greater efficiency in removing dust, dirt, smoke and other foreign particles from the air. The Precipitron cells are simply installed in the forced-air or air-conditioning system and sealed so the air must pass through them (see illustration). Dirt, smoke and soot pass through an electrostatic field, receive a positive charge, and are drawn to a collection plate of opposite polarity where they remain until flushed down the drain.



ADVANTAGES

The following major advantages make Precipitron the complete answer to mass air cleaning in all buildings using forced ventilation or air-conditioning duct systems.

Unmatched Efficiency — Precipitron removes particles as small as 1/250,000 of an inch in diameter, and removes 90 per cent of all air-borne dust particles in the air stream.

Easily Cleaned—New vertical collector plates make it still easier to clean the Precipitron. Collected dirt is flushed harmlessly down the drain.

Safe—Precipitron is approved by Underwriters' Laboratories without installation of additional fire protection apparatus.

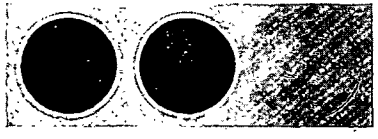
Nothing to Clog—Precipitron creates only negligible resistance in the forced air system under continuous operation.

No Moving Parts—Precipitron has no more moving parts than a storage battery.

*Trademark registered in U.S.A.

TESTS PROVE EFFICIENCY

Accepted testing methods previously conducted on a weight basis have been obsoleted since Precipitron entered the field of air cleaning. The Blackness Test reproduced here provides greater accuracy, approximates the actual particle count and is made without the addition of artificial dust. The efficiency of Precipitron by Weight Test showed over 99% and by the Blackness Test as high as 90% at 300 FPM per 36 in.



Uncleaned Air — *Mechanically Cleaned Air* — *Electronically Cleaned Air*

Test Comparisons Proving High Efficiency of Precipitron

MAJOR COMPONENTS OF INSTALLATION WITH AUTOMATIC WASHER

CELLS

The cell itself, known as the Collector, is composed of two duplicate sets of high voltage and ground plates mounted between two load-bearing end plates laced together by steel bars, angles and channels to form a rigid structure.

Precipitron is built in sizes to handle from 1200 cfm (for a single 24 in. cell) to any desired volume through multiple cell arrangements. Cells are available in two sizes: 24 in. x 24 in. x 26½ in. and 36 in. × 24 in. × 26½ in. For 90 per cent efficiency, the 24 in. and 36 in. cells are rated at 1200 and 1800 cfm respectively. For 85 per cent efficiency, ratings are 1500 and 2250 cfm.

IONIZERS

Made of formed sheet steel, top, bottom and side plates welded together to form a rigid frame. The ground tubes run vertically between the top and bottom plates. Two sets of Ionizing wires are used; one, the usual vertical set, and another set stretching horizontal at the ends to prevent leakage around the ends of the wires.

POWER PACKS

The Power Pack is employed to change the a-c supply to d-c of suitable voltage to charge the Ionizers and Collector Cells. They furnish approximately 6 kv (6000 volts) d-c to the plate circuit of Collector Cells and 13 kv (13,000 volts) d-c to the Ionizer circuit.

The Power Packs contain rectifier tubes, Inerteen impregnated capacitors, and an indicator system and panel. Cabinet doors, opened by a thumb screw, provide time delay, simultaneously breaking the primary circuit, if this has not already been done at another point.

Three types of Packs are available, differing principally in the number of cells they will energize. These types are:

Type L—Suitable for 4 to 16-36 in. cells.

Type S—Suitable for 1 to 5-36 in. cells.

Type M—Suitable for 1-24 in., 2-24 in., 1-36 in., or 1-24 in. and 1-36 in. cell.

All Power Packs are designed to operate from 115 volts ± 5 volts on single phase a-c systems.

TRAVELING HEADER

The basic element of the automatic washer is a set of spray nozzles which spray either water, compressed air or adhesive on the Precipitron cells. Water nozzles are mounted on a horizontal header and a mechanism is provided to move this header vertically up and down over the face of the cells. The secondary element of the washer is the control of its automatic operation which is described under "Control Panel."

Illustration below shows complete washer with back vanes closed, the washing being done with vanes in this position as air is prevented from going through this group and forced through the balance of the cells where it is cleaned.

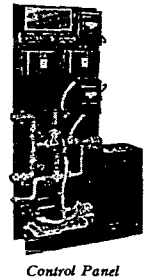
Header is moved by means of roller chains driven through a gear reducer by a ½ hp, 220-440 volt, 3 phase motor. A limit switch controls up and down travel of header.

CONTROL PANEL

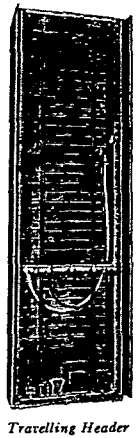
This panel operates the two traveling headers of the washer automatically once they have been placed in position on either side of the cell group to be washed and the momentary contact pushbutton on the panel has been pushed.

Control Panel comes fully assembled, is internally wired and ready to be set in place. The control circuit requires 110 volts single-phase (25 or 60 cycles) of sufficient capacity to carry a ¼ hp motor. The power circuit requires either 220-volt or 440-volt, 3-phase, 25 or 60 cycle power of sufficient capacity to handle two ½ hp motors in reversing service.

FULL DETAILS—For complete information on this new principle of cleaning air electronically, write Westinghouse Electric Corporation, Box 868, Pittsburgh 30, Pa. or contact your nearest Westinghouse office.



Control Panel



Travelling Header

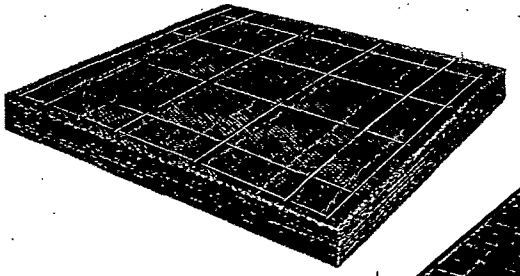
Research Products Corporation

Madison 3, Wisconsin

RESEARCH AIR FILTERS FOR HEATING AND VENTILATING

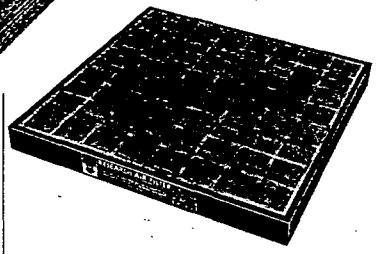
U. S. Patent 2070073 — U. S. Patent 2294478

RESEARCH AIR FILTERS



No. "200" Series

←No. "200" Series. When this Re-Fil-Able filter becomes dirt clogged, the wire grids are unhooked, the filter pad replaced. Pads fit snugly, sealing against air leaks. (Can be Rip-Cleaned).



No. "100" Series

Self-Sealing Edge. The "200" series filter is made ¼ in. "oversize" so it extends over the sides of the wire grids thus sealing without the need of sealing felts. This prevents the by-pass of air around filter.

Re-Fil-Able. When the ultimate dirt holding capacity of the pad is reached it can be replaced with one 2 in. pad or two 1 in. pads. The wire grids are permanent —can be used over and over again.

Rip-Clean. When the filter pad becomes surface clogged with particles of dirt, lint, etc., the wire grids are unhooked, the two top layers of the pad rolled off and removed, thus Rip-Cleaning for further use. The removal of surface dirt in this manner may be repeated 5 times—thus extending the useful life of the filter.

No. "100" Series. This fiber framed filter is designed so the entire unit is easily replaced when its dirt holding capacity is reached. Offers high dust removing efficiency. Especially adaptable to heating, ventilating and air conditioning systems.

Research Air Filters are used extensively wherever air is moved mechanically in forced warm air heating, ventilating and air conditioning systems because of their superior capacity for catching and holding dust, dirt, soot and pollen. Tests for both the No. 100 and No. 200 air filters show a 91% dust removal efficiency (with 80-20 dust) and 99% pollen removal efficiency.

Research Filter Banks. Research Air Filters are used extensively in filter banks—both flat and V types. Write for technical data booklet.

American Moistening Company

ESTABLISHED 1888

Providence 1, R. I.

ATLANTA 2, GA. — BOSTON 9, MASS. — CHARLOTTE 1, N. C.

AMCO

UNIT HUMIDIFYING AND AIR CONDITIONING EQUIPMENT

A few of many AMCO products with a Long Record of Dependable Performance

Self-cleaning Atomizers
Sectional Humidifiers.
Amtex Humidifiers.
Hand Sprayers.
Mine Sprays.
Fabric and Paper Dampeners.
Mechanical Psychrometers.
Electro Psychrometers.
Sling Psychrometers.
Hygrometers.

The Amco line of devices for the supply, maintenance and control of humidity is complete in its ability to meet any presented problem of applied humidification. Used independently or as an adjunct to Central Station equipment, these devices automatically maintain any required humidity condition in a capable uniform performance.

AMCO ATOMIZER—No. 5

Quality and quantity of spray are maintained even under adverse conditions because this atomizer is automatically self-cleaning. When the compressed air supply is shut off, either manually or in response to a humidity control, *both* air and water nozzles are thoroughly cleaned.

AMCO HUMIDITY CONTROLS

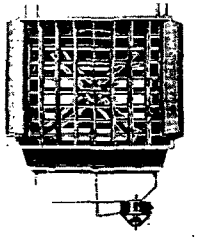
Compressed Air Operated

An extremely accurate and active device operated by compressed air which assures a regulation of humidity within exceedingly close ranges.

AMCO HUMIDITY CONTROL

Electrically Operated

Similar in principle to the Compressed Air Type except that the hygroscopic element operates electrical contacts which control the units.



AMCO EVAPORATIVE COOLING UNIT

The Amco System of evaporative cooling contributes to smooth production at high speeds in two ways; it maintains the percentage of relative humidity best suited to the fibre and process involved, and at the same time promotes the comfort and efficiency of personnel by obtaining the maximum practical cooling effect from evaporation. It does this by introducing outside air into the room in varying amounts, regulated in accordance with climatic conditions and inside requirements.

A ductless system—very flexible and portable. Can be applied in conjunction with an existing humidifying system.

April Showers Company

4126 Eighth Street, N.W. **Washington 11, D. C.**

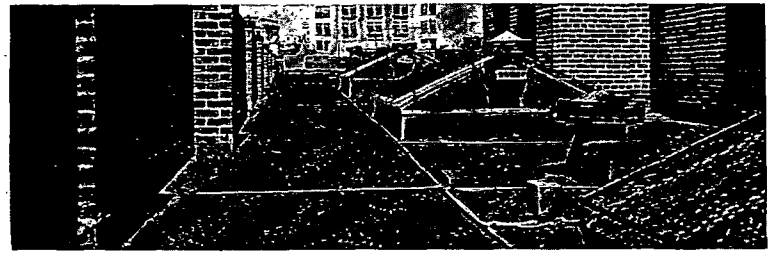
April Showers

(Trade Mark Reg. U. S. Pat. Office)

AUTOMATIC EVAPORATIVE ROOF COOLING

Distributors and Dealers in Principal Cities

SPRAY PROTECTION AGAINST SOLAR RADIATION



APRIL SHOWERS is the trade name of an EFFICIENT fool proof method of preventing roof heat penetration or solar infiltration causing excessive heat in upper floors and in factories, stores, theatres, shops, etc.

City water under normal city pressure is usually adequate to serve your APRIL SHOWERS system. The sun operates it. Roofs LAST LONGER, greater comfort is assured, production during the summer's heat is kept apace, errors through fatigue are avoided, and MANAGEMENT acclaims APRIL SHOWERS a GOD-SEND.

DEALERS and CONTRACTORS—AN ANNOUNCEMENT

Any established air conditioning engineer, contractor, dealer or any plumber and kindred workmen can obtain APRIL SHOWERS supplies, plans and instructions for erecting APRIL SHOWERS for you or your clients. Write for "INTRODUCING APRIL SHOWERS" which is a loose-leaf bound book for Installers and Dealers.

Contractors, Dealers and Installers be sure to share in the APRIL SHOWERS post-war prosperity. You are not required to inventory any merchandise. You purchase only such equipment as is needed for each installation. You set your own SELLING PRICE. You share PROFITS with NO ONE.

Hundreds of installations, from Boston to Los Angeles have been made.
Write for information and address of nearest distributor.

Inquiries will be answered promptly. Estimates free upon request.

APRIL SHOWERS controlled roof cooling is protected by U. S. Patents; beware of imitators or infringers.

APRIL SHOWERS steam humidifying heads now ready.

Developed in 1933-34.

Lilie-Hoffmann Cooling Towers, Inc.

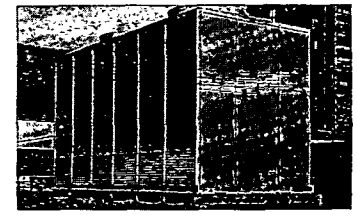
Exclusive Builders of Cooling Towers for 46 Years

4239 Duncan Ave., St. Louis 10, Mo.

Two Modern Plants—St. Louis, Mo. and Plainview, Texas

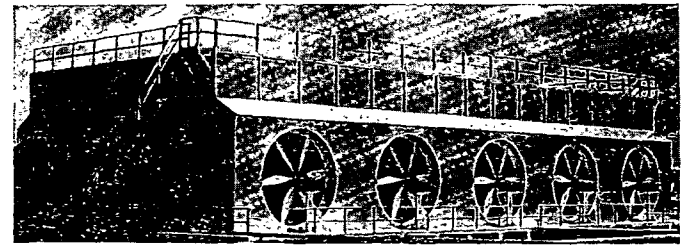
INDUCED DRAFT TOWERS

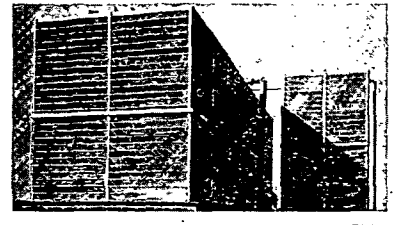
New type filling reduces static pressure; uniformly distributes air and water over effective tower area. Ring gravity distribution system requires only one riser pipe. Each cell main trough equipped with Weir gate, giving individual cell control. Teco timber connectors develop 100 per cent working stress of members.



FORCED DRAFT TOWERS

Three-tier zigzag pattern spray eliminator system cuts drift loss to a minimum. Fan opening covered by galvanized screen wire. Built in single or multiple cells. Continuous design improvement, based on installation studies, presents record of unfailing operation.



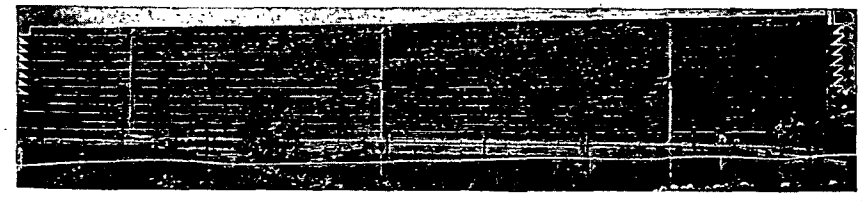


ATMOSPHERIC SPRAY TYPE

Generally offered in capacities up to 3000 gpm. Distribution by galvanized pipe header with smaller lateral arms, equipped with non-clogging spray nozzles. Louvres fit into mast slots—no nails required. Recommended where cost is prime factor; and close approach to wet bulb is unnecessary. Rigidly braced and tied together to prevent warping and buckling.

ATMOSPHERIC DECK

Distribution by either wood trough, or wood pipe header system. Removable deck grids. Special L-H drift baffles on deck supports prevent entrained water loss. Generally 12 ft wide—can be made any length or height.

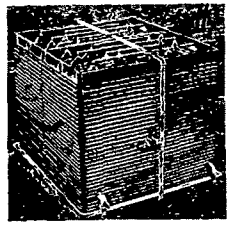


The Marley Company

(Fairfax and Marley Roads,) **Kansas City 15, Kansas**

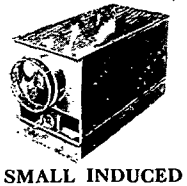
Branches or Agents in Principal Cities

Spray Nozzles and a Complete Line of Water Cooling Equipment



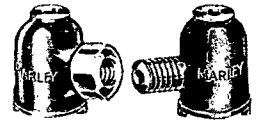
MARLEY NATURAL DRAFT TOWERS

Practically unlimited range of closely graduated sizes, entirely shop fabricated. Minimum initial, maintenance and operating costs. Many exclusive MARLEY advantages. **Bulletins 201 and 202.**



SMALL INDUCED DRAFT TOWERS

Small, self-contained, steel units for 2 to 170 ton service, to go indoors or out. Horizontal air flow. Smaller sizes shipped all assembled. Larger sizes are entirely shop fabricated for fast, simple assembly at the location. **Bulletins 504 and 505.**



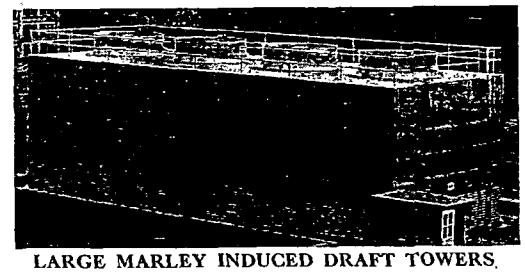
MARLEY Small 2-Piece Nozzles for Brine Spraying, Air Washing and Similar uses.

MARLEY 1-Piece Nozzles for Cooling Towers, Spray Ponds, etc.

MARLEY Humidifying Nozzle adds moisture to air in open rooms or duct system.

MARLEY PATENTED NON-CLOG SPRAY NOZZLES

Made in scores of types and sizes. Practically any metal or alloy the purpose may demand. **Bulletins 101, 102 and 103.**



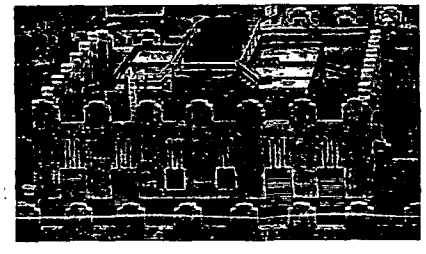
LARGE MARLEY INDUCED DRAFT TOWERS

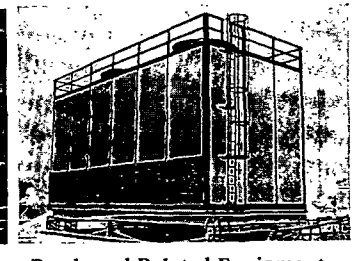
Both Conventional and Double-Flow types, for heavy duty water cooling services of all kinds. Any capacity, with one fan or many, individually engineered to the exact requirements of each installation. MARLEY patents cover a variety of important features for extreme operating flexibility, high efficiency and economy.

Redwood is the standard structural material. Steel, transite and other special materials require much higher priorities for the war's duration.

Conventional Induced Draft (pictured above in steel construction; below right, of Redwood) well adapted to most large capacity services. **Bulletin 603.**

Double-Flow Induced Draft (below left) for unusually large capacity services. **Bulletin 700-A.**





Also Many Other Types of Towers, Spray Ponds and Related Equipment

J. F. Pritchard & Co.

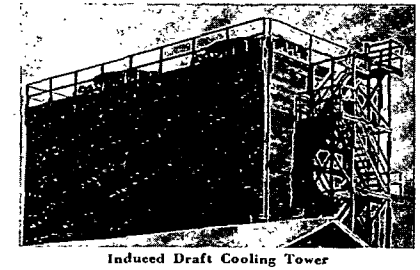
Fidelity Building, **Kansas City 6, Missouri**

New York, Pittsburgh, Houston, Atlanta, Tulsa, Denver, Salt Lake City, El Paso, Omaha

Engineers—Constructors—Manufacturers for the Refrigerated Industries
Specialized Equipment for Use with Refrigeration and Air-Conditioning

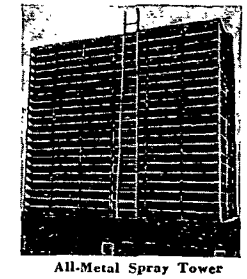
ENGINEERING-CONSTRUCTION

Many years of experience, covering large plants and small, all types of cold storage and refrigeration processing, qualify Pritchard to handle complete plants, revamps or additions, from plans to operation, on a full-responsibility turnkey basis that saves time and money. Any *part* of the complete Pritchard service (process designing, economic studies, plant layout, unit installations, etc.) is available separately.



Induced Draft Cooling Tower

AIR TREATER—Renders Air Germicidally Pure—Removes Dust, Pollen, Smoke and Other Solid Particles and Microorganisms—Controls Humidity—Refreshes Air by Adding Oxygen—A compact "package" unit for simple installation in any duct system.



All-Metal Spray Tower

Indoor Forced Draft

AIR & GAS DRYING EQUIPMENT—Total or Controlled Dehumidifying—Adaptable for Various Solid or Liquid Adsorbents—Standard units, all capacities, automatic or manual regenerating.

HEAT EXCHANGERS AND CONDENSERS—All types of heat transfer apparatus, high or low pressure, shell and tube, atmospheric or submerged, standard or special designs. Pritchard has the facilities and application knowledge to meet any need with high-efficiency units.

COOLING TOWERS—Pritchard has long been one of the foremost suppliers of cooling towers of all types and capacities. Specifications, prices, ratings and application recommendations furnished promptly.

Small Mechanical Draft Towers—Forced or induced, for indoors or out. Self-contained steel-cased units, easy to install, 2-ton capacity and up.

Natural Draft Spray Towers—With or without deck filling, with or without basins. Simple to assemble; slip-fit louvers; Redwood or all-metal; any capacity.

Natural Draft Deck Towers—Celebrated Pritchard "drift retrievers" minimize drift loss without restrictive secondary louvers. Free air flow gives higher efficiency. Rugged construction; low-pressure, uniform water distribution.

Induced Draft Towers—Extra-sturdy structural members and joints; double-wall casing. Valuable exclusive features include "Featherweight Fans" of wrought monel, superior air delivery per horsepower; trouble-free gear drives, either right-angle type with flex-coupled shaft to motor outside of fan ring or the famous "Sealdflow" unit streamlined to the fan hub, mounting motor and gears vertically on the fan shaft; low pumping head, uniform water distribution.

EVAPORATIVE CONDENSERS AND COOLERS—Complete welded, galvanized, self-contained condenser; also for cooling jacket-water and lube oil of engines, and industrial processing. Maintain accurate temperatures, eliminate freeze-ups, save space, simplify piping. All prime surface coils, readily cleanable.

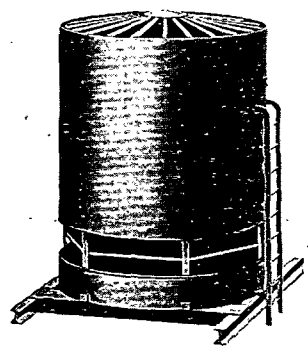
"AIRDFIN" AIR-COOLED HEAT EXCHANGERS—New efficiency and economy for many cooling and condensing functions, i.e., steam and other vapors, jacket water, oils, gases, quenching baths, etc. High-efficiency fan propels air through banks of finned tubes. No water problems.

Rogers Diesel and Aircraft Corporation

manufacturers of

COEY FLOATING FILM COOLING TOWERS

1120 Leggett Avenue — **New York 59, N. Y.**



Patented Spray Eliminator . . . sloping blades discharge accumulated water droplets back into water distribution trough. Spray is never evident.

Fan Drive Motor . . . protected against moisture, this motor will stand up under severest conditions.

Light in Weight . . . the Coey tower is lighter in weight than other towers of similar capacity because its interior is free of slats and baffles.

Readily Accessible . . . for cleaning and maintenance due to open interior construction.

Fireproof . . . Coey towers are fireproof because of all metal construction.

Low Noise Level . . . its quiet operation is outstanding.

PATENTED "FLOATING FILM" PRINCIPLE INSURES PREDICTABLE, ACCURATE COOLING

Water, pouring in from the top of the Coey Cooling Tower, is met by a counterflow of air at a velocity of approximately 1500 ft per minute. The water flows across this air in a *floating film*. At five platform stages this entire *floating film* of water is broken into droplets substantially all the same size and each droplet is surface cooled. Size is important because when droplets are too large they do not cool enough, when they are too small they evaporate completely.

Five times during their downward course, the droplets are collected into floating film and cooled surfaces are mixed with warmer droplet centers for an overall lower temperature. This five stage operation insures precise control . . . maximum cooling.

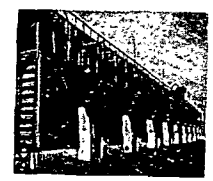
There is a Coey tower for every cooling requirement. Write Department A, Cooling Tower Division, Rogers Diesel and Aircraft Corporation, for complete information.

Water Cooling Equipment Company

8613 New Hampshire Avenue

Affton Station, St. Louis 23, Missouri

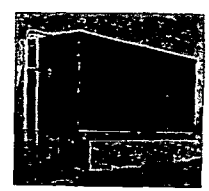
MANUFACTURERS OF MECHANICAL DRAFT AND ATMOSPHERIC COOLING TOWERS



FORCED DRAFT COOLING TOWER

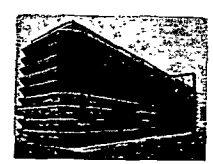
41,500 g.p.m. twenty-two cell, heavy duty, double forced draft cooling tower . . . equipped with twenty-two 12-foot diameter adjustable pitch, propeller type fans.

REDWOOD "WATERFALL" ATMOSPHERIC SPRAY COOLING TOWER



Factory fabricated and shipped knocked-down with all hardware, spray headers and spray nozzles. Complete erection instructions and drawings are furnished to assist in the assembly of the tower.

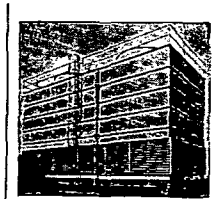
These towers are portable and can be knocked-down and moved simply by removing the louvres and bolts. Shipment can be made from stock. Write for Bulletin 125-A.



REDWOOD ATMOSPHERIC DECK TYPE COOLING TOWER

PRODUCTS

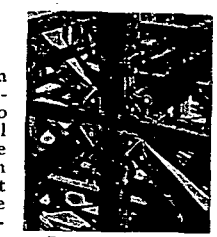
Heavy duty forced and induced draft cooling towers, standard induced and forced draft towers, induced draft coil towers, atmospheric deck cooling towers, atmospheric spray towers, spray ponds and spray nozzles.



INDUCED DRAFT

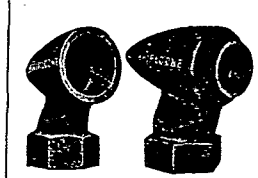
This type of cooling tower is recommended for installations where noise is a prime factor of consideration. The noise of the mechanical items is carried upward and discharged into the air. The mechanical draft cooling tower assures a positive cooling of the water to a specified temperature independent of wind velocity.

TIMBER JOINT CONNECTOR



Special cast iron timber joint connector used to develop the full strength of the timber joint, an accomplishment almost impossible to achieve by bolting.

Patent No. 2,280,121



Patent No. 2,123,697

SPRAY NOZZLES

"Whirlcone" non-clogging, low pressure, centrifugal type spray nozzles for spray ponds, spray towers and other uses. Write for Bulletin 76-A.

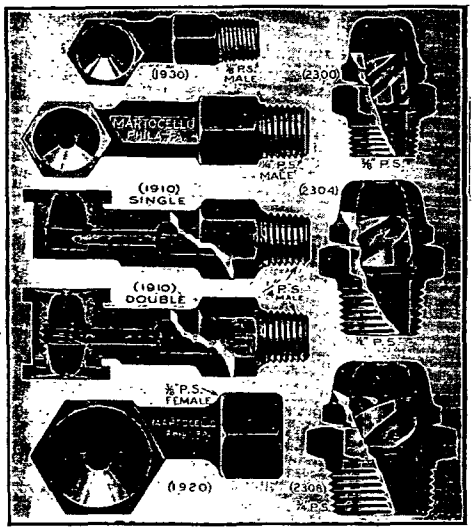
ENGINEERING

Design and construction are based on sound engineering principles to meet specific requirements for cooling performance and structural strength. Redwood, steel or other suitable materials are used.

Jos. A. Martocello & Company

229-31 North 13th Street, **Philadelphia, Pa.**

SPRAY POND AND ATOMIZING SPRAY NOZZLES



MARTOCELLO

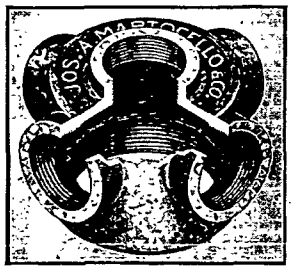
Atomizing Spray Nozzles produce a uniform, good, wide spray with less friction and at minimum pressure requirements.

Nozzles illustrated at the right are manufactured with precision in Brass Forgings and Bar Stock. Their design has been thoroughly tested for results and durability and will give you satisfaction.

Successful, Efficient Results depend largely upon selecting the proper number and type of Nozzles with Brass or Monel cap suitable for your job. Therefore we suggest you send us your specifications as we also have several Types other than illustrated and we will gladly assist you to obtain the most efficient application.

MARTOCELLO

Spray Pond Nozzles of a sturdy one-piece construction —cast of High Grade Red Brass with Inlet and Outlet accurately machined, are less clogging, offer less friction and give unsurpassed overall efficiency.



MARTOCELLO CLUSTER CASTINGS

Sturdy Grey Iron Construction with large area for reduced friction and even distribution. They are Hot Dipped Galvanized after fabrication.

Furnished with **Nozzles** and continuous Standard Steel **Long Sweep** Galvanized Pipe **Spray Arms** and **Center Nozzle** Nipple in accordance with layout required.

Sizes Carried in stock for **Prompt Shipment** 1½ in. P. S. Outlet, 3 in. P. S. Inlet and 2 in. P. S. Outlets, 4 in. P. S. Inlet Cluster Castings.

WRITE, PHONE OR WIRE

For Bulletin listing Capacities and Prices. Prompt shipments from stock.

Monarch Manufacturing Works, Inc.

2731 E. Westmoreland 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.

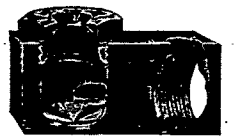


Fig. 631

Capacities: Gallons per Hour

Sizes		Lb Operating Pressure			
Pipe	Orifice	10	20	30	40
¼"	69			5.3	6.15
	61		6.25	7.5	8.3
	61 x 53	5.7	7.5	8.8	10.0
	53	6.4	9.0	10.7	11.8
	3/32	14.1	18.7	22.5	25.0
	⅛	23.2	32.0	40.	47.3
	11/64	31.0	43.9	54.5	63.0
	3/16 x 3/32	45.0	64.2	75.0	90.0
	7/64" S	73.4	105	137	155
	1/16	85.8	116	140	164

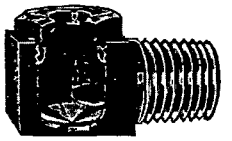


Fig. 629

AIR CONDITIONING AND OIL BURNER NOZZLES

Water Capacity in Gallons per Hour

Fig. F-80

Nozzle No.	Lb Operating Pressure				
	25	40	60	80	100
1.35		.57	.69	.83	.92
1.65		.75	.89	.99	1.12
2.00		.94	1.14	1.23	1.40
2.50		1.13	1.45	1.64	1.86
3.00	1.03	1.39	1.62	1.85	1.95
3.50	1.36	1.77	2.11	2.46	2.80
4.00	1.56	2.00	2.42	2.77	3.16
4.50	1.86	2.32	2.77	3.21	3.68
5.00	2.20	2.88	3.57	4.09	4.59
5.50	2.22	2.96	3.75	4.31	4.78
6.00	2.55	3.35	4.01	4.78	5.23
7.00	2.90	3.91	4.60	5.17	6.00

Produce finest breakup possible 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 Nozzle No. 60.00.

Furnished of all Brass for Water—Stainless Steel tip and disc for Oil. Standard with ⅛ in. or ¼ in. female pipe Brass adapter and Monel gauze strainer.

SPRAY POND NOZZLES

For recooling condenser water, etc. Operate on pressures from 5 lb upward. Made of Cast Red Brass and in pipe sizes 1 in., 1½ in., 2 in., and 2½ in. Capacities from 4.1 to 88 gpm at 7 lb pressure.

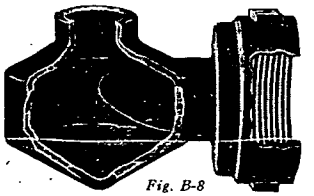


Fig. B-8

Write for Detailed Catalogs

Acme Industries, Inc.

Jackson, Michigan

Representatives in Principal Cities

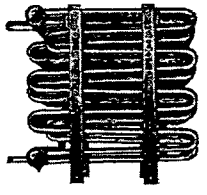
REFRIGERATION AND AIR CONDITIONING EQUIPMENT

FREON CONDENSERS
AMMONIA CONDENSERS

Shell and Tube Type for use with Ammonia, Freon or other refrigerants. Standard or special designs to meet varying water temperatures available and condensing temperatures desired.

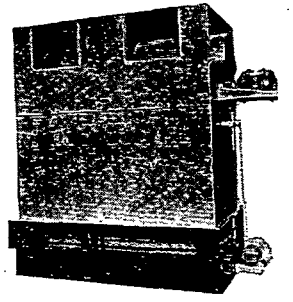
DRY-EX COOLERS

Refrigerant in tubes, solution baffled through shell. For cooling water, brine, glycols or alcohols by direct expansion of refrigerant.



PIPE COILS

Fabricated in all shapes and sizes from ⅛ in. IPS to 2 in. IPS. Special assemblies available for custom built evaporative condensers, raceway coils.

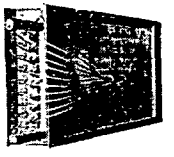


EVAPORATIVE CONDENSERS

All prime surface for Freon or Ammonia refrigerants. Heavy galvanized iron casing, inside treated with rust resistant paint. Capacities to 100 tons.

HEAT INTERCHANGERS

Shell and coil units for small capacities. Shell and tube units for large installations. 16 standard models from one ton to 180 tons capacity.



FIN COILS

Direct expansion coils with liquid distribution for Freon. Low temperature, wide fin spacing all steel coils hot dip galvanized for flooded ammonia.

Acme Industries, Inc. also manufactures flooded water and brine coolers, receivers, oil separators and lowside specialties.

WRITE FOR CATALOG ON ANY PRODUCT

AEROFIN CORPORATION

410 So. Geddes Street

Syracuse 1, N. Y.

AEROFIN

Standardized Light-weight Heat Exchange Surface

Branch Offices

CLEVELAND, CHICAGO, NEW YORK, PHILADELPHIA, DETROIT, DALLAS, TORONTO

Aerofin is the modern Standardized Light-Weight Encased Fan System Heating and Cooling Surface originated by *Fan Engineers* to meet the 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, protecting the Unit from all the 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 together adjacent Units, or for duct connection.

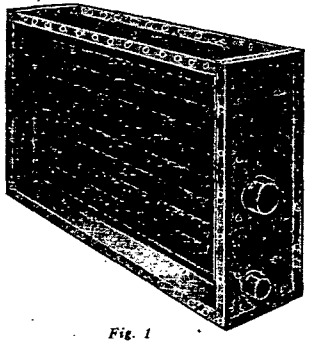


Fig. 1

Aerofin Non-freeze heater (Fig. 1) is non-freeze, non-stratifying spiral fin coil built into casing for air conditioning units or for installing in ducts. May be installed horizontally or vertically. Used on any two-pipe steam system for preheating or reheating. Modulating control on preheaters.

Available in **13** lengths and **3** widths, from net face area of 2.76 sq ft to 26.28 sq ft.

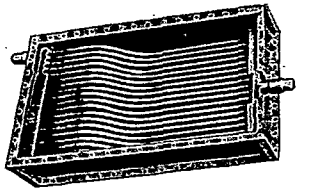


Fig. 2

Flexitube Aerofin (Fig. 2) is distinguished from all other developments by its off-set tubes, so arranged as to absorb all expansion and contraction strains.

Headers—Steel.

Tubing—⅝ in. O.D. copper, admiralty or aluminum.

Joints—Where admiralty or copper tubes are used together with bronze or steel headers tubes are brazed to headers. Where both aluminum tubes and headers are used tubing is welded to headers.

Casings—Copper, aluminum or galvanized iron.

Design—Constructed with headers on opposite ends making possible installation of units with tubes horizontal or vertical.

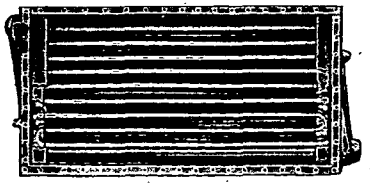


Fig. 3

Universal Aerofin (Fig. 3) is distinguished by its "S" bend construction of tubing, units designed with steel headers on opposite ends, the ends of the "S" bends being connected thereto by compression nuts, the bends taking care of the expansion and contraction of the tubing.

Recommended where close control is desired.

Headers—Pressed steel.

Tubing—1 in. O.D. Copper or admiralty.

Casings—Copper, aluminum or galvanized iron.

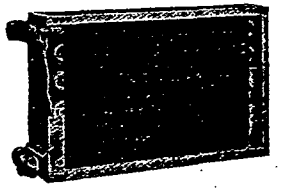


Fig. 4

High Pressure Aerofin (Fig. 4) is of continuous tube design, being recommended where extremely high pressures of steam are used.

Headers—Pressed steel.

Tubing—1 in. O.D. Copper or admiralty.

Casings—Copper, aluminum or galvanized iron.

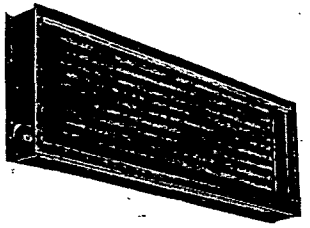


Fig. 5

Booster Aerofin—straight tube type, single pass construction for pressures from 1 to 200 lb gauge.

Headers—cast bronze.

Tubing—⅝ in. O.D. copper.

Casings—copper, aluminum or galvanized iron. Recommended where small coils are needed or to raise the air temperatures in branch ducts.

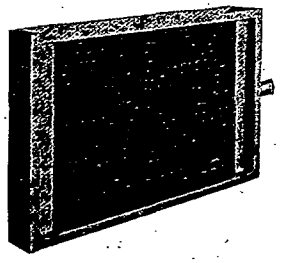


Fig. 6

Narrow Width Aerofin: (Fig. 6) recommended for water cooling or for flooded Freon systems. Made in straight tubes only with headers on opposite ends, joints between headers and tubing being brazed. Construction similar to Flexitube AEROFIN.

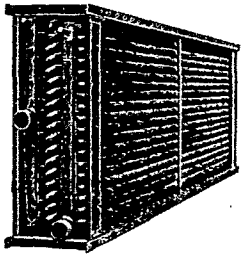


Fig. 7

Aerofin Continuous Tube Water Coils (Fig. 7) are designed for air cooling by circulating cold water through the AEROFIN and air over extended fin surface. Made for either horizontal or vertical air flow.

Tubes and fins are copper, completely tinned with permanent metallic bond between fin and tubes. Headers are made of one-piece cast bronze and casings of heavy galvanized iron or copper.

Units tested to 1000 lb hydrostatic pressure.

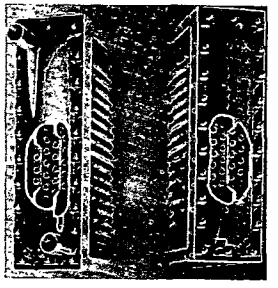


Fig. 8

Aerofin Cleanable Tube Units (Fig. 8) for cooling only made with headers removable to permit cleaning tubes. Recommended for use where sediment or scale forming chemicals are present in the cooling water.

Headers—Cast iron.

Tubing—Copper or admiralty.

Casings—Copper or glavanized iron.

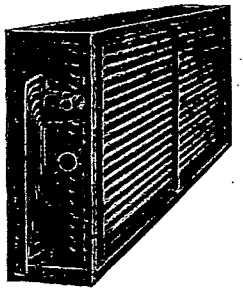


Fig. 9

Aerofin Direct Expansion Units: (Fig. 9) Centrifugal Header Type—Recommended where control of rows in direction of air flow is not required.

AEROFIN Sizes

Flexitube: 13 standard lengths, three widths, one and two rows deep.

Narrow: same as Flexitube.

Universal: 17 standard lengths, two widths, one and two rows deep.

Continuous Tube: 13 standard lengths, three widths, 2-3-4-5 and 6 rows deep.

Cleanable Tube: 17 standard lengths, one width, 2 and 4 rows deep.

Direct Expansion: Centrifugal Header—11 standard lengths, three widths, 2-3-4-5-6 rows deep.

Steel Supporting Legs: 18 in. and 24 in. high. Punched same bolt hole centers as standard casings. Quickly attached. No other foundation required.

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus. List upon request.

Write Syracuse for Heating Bulletin G-32; Direct Expansion Bulletin DE-34 on refrigeration type units; Continuous Tube Bulletin C. T. 34 for Water Cooling Coils; or phamplet on Cleanable Type AEROFIN for cooling.

Condenser Service & Engineering Co., Inc.

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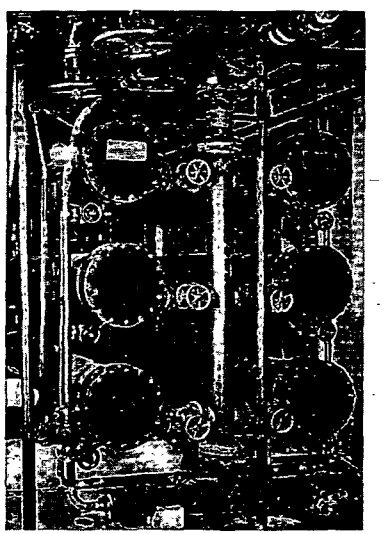
Twenty years of specialized experience. *designing*, building, fabricating and a special feature—a designing organization which also services heat exchangers—assures sound design, quality construction and economical operation. Conseco builds heat exchangers for all types of air conditioning systems using, Freon, CO_2, ammonia, methyl chloride and SO_2.

Conseco maintains a highly-trained service organization equipped with special tools developed by us to save time, labor and material, ready for action any time of day or night, anywhere on the continent. Heat exchangers are retubed or repaired in the shortest possible time, and at low cost. Tube sheets, tubes, ferrules, all types of packing, and other necessary materials are always stocked, ready for instant shipment.

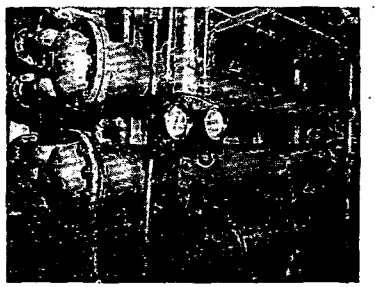
The keynote of our service organization is fast, accurate, dependable service performed by competent, properly equipped men working under qualified engineering supervision.

Every detail of a Conseco heat exchanger is designed to profit from good and bad features uncovered by years of service work on units of every type and make. This has resulted in the production of simple, efficient units, low in original cost and inexpensive to maintain.

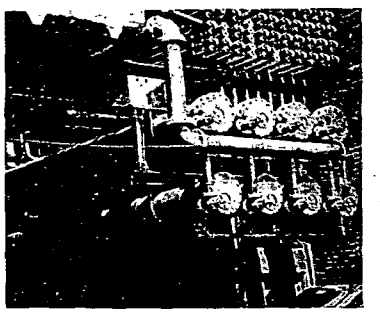
Conseco engineers will be glad to serve you. If you want fast action, write, phone or wire.



Conseco CO_2 heat exchangers operating at 1200 psi in various Sanger chain theatres.



Conseco double-bank heat exchanger installation aboard ship.



Conseco ammonia heat exchanger installation, Stanley Theatre, Philadelphia.

The G & O Manufacturing Company

138 Winchester Avenue **New Haven, Connecticut**

G&O

SQUARE FIN TUBING

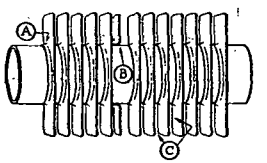
STRAIGHT LENGTHS—U-BENDS—CONTINUOUS COILS

THE use of INDIVIDUAL fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface.

Fins of any size or shape may be obtained giving any desired proportion of primary and secondary surface.

A square fin has about 30 per cent greater surface than a round fin of a diameter equal to one side of the square.

Individual fins permit of any fin spacing; also, of using fins in groups at intervals along tubes.



A—Generous Fin Collar provides large contact area between Tube and Fin.
B—Tube expanded against Fin Collar; insures mechanically tight joint, made permanent by bond of high temperature alloy—complete thermal contact.
C—Free air-flow passages; non-clogging.

STANDARD SIZES

O.D. of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
⅜"	⅞" sq.	6	0.80 sq. ft.
⅜"	⅞" r'd.	6	0.60 sq. ft.
⅝"	1⅛" r'd.	6	0.87 sq. ft.
¾"	1½" r'd.	6	1.55 sq. ft.
¼"	1⅜" sq.	6	2.40 sq. ft.
1"	2⅛" sq.	6	4.00 sq. ft.
1⅜"	2⅛" r'd.	4	2.33 sq. ft.

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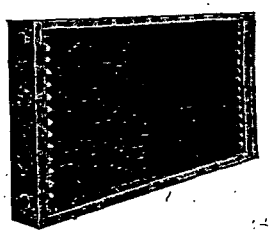
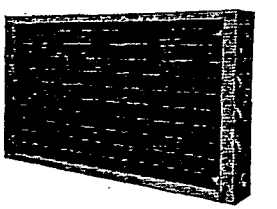
G&O Finned Radiation Coils for industrial applications are available in a wide range of sizes.



Universal U-108



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The Rome-Turney Radiator Co.

Rome, N. Y.

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Helical Fin Tubing

Over 100 sizes—for all air heating and air cooling equipment—straight lengths—U bends—Continuous Coils.

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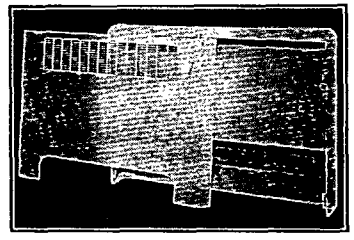
Helical Fin Tube Core Assemblies—for heat transfer equipment.



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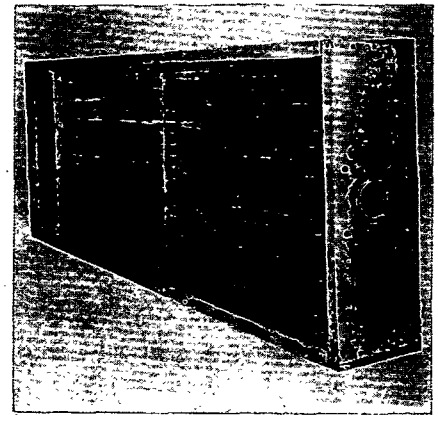
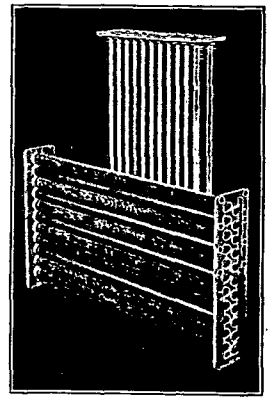


Blast Air Heating and Air Cooling Coils



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Rome products are "backed" by 40 years of experience.

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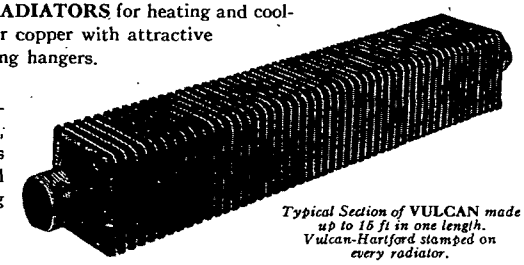

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Heat Output for 2 in. IPS is 5.25 sq ft per lineal foot with 1 lb steam and air at 70 deg, for 1¼ in. IPS—4.25 sq ft.

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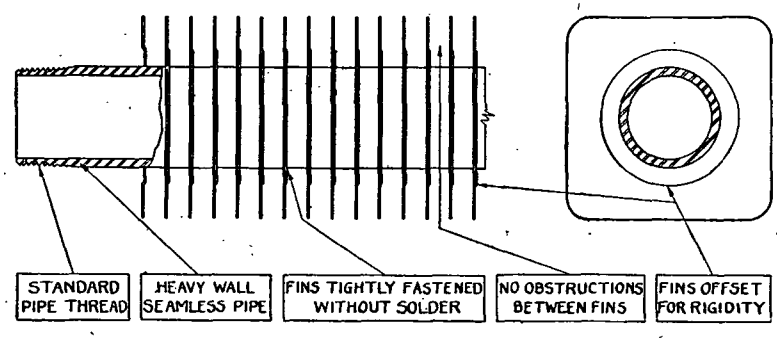

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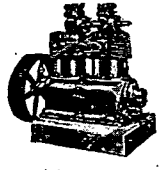
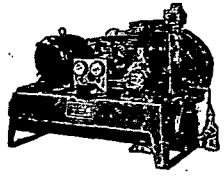
BUILDERS OF DEPENDABLE REFRIGERATION EQUIPMENT SINCE 1905

• BAKER builds compressors, condensing units, compressor units, shell and tube equipment, and cooling units for all commercial and industrial air conditioning applications. BAKER engineers are free to select the equipment and refrigerant best suited to your specific requirements. Write for information, descriptive literature on units shown below.



Methyl Chloride or "Freon-12" Compressors

• Four-cylinder type, available in sizes from 3 hp to 60 hp. Semi-steel cylinders and pistons. Counter-balanced crankshaft, precision ground. Timken roller main bearings. Full force feed lubrication is furnished by gear type oil pump.



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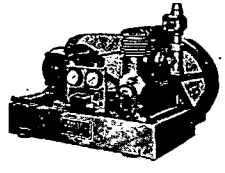
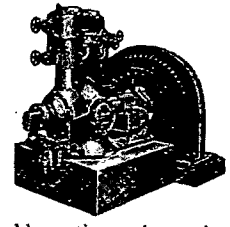
• Complete line of self-contained, automatic units. Sizes range from ½ hp to 60 hp capacity. Two- and four-cylinder types. Shell and tube condenser-receiver.



Methyl Chloride or "Freon-12" Compressor Units

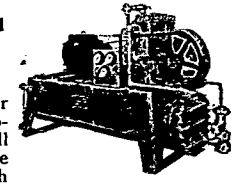
• Arranged for use with evaporative type condenser or separately mounted shell and tube condensers. Sizes range from 3 hp to 60 hp. Two- and four-cylinder types. Automatic controls.

Ammonia Compressors (3 to 100 hp)



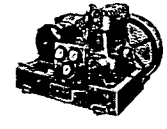
• Vertical enclosed, single-acting type. Can be installed in multiple installations. V-belt drive or direct connection to motors or engines. Double suction and capacity reduction in larger sizes.

Ammonia Water-Cooled Condensing Units (3 to 15 hp)

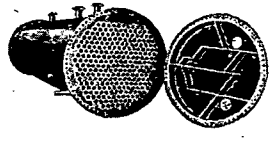


• Excellent for all industrial applications. Shell and tube type condensers with pressure-operated water control valve. Compact design economizes floor space.

Ammonia Compressor Units (3 to 20 hp)



• Designed for quick installation and ready accessibility to all unit parts. For use with evaporative or separately mounted shell and tube type condenser. Full enclosed automatic motor control with overload and low-voltage protection. Two- or four-cylinder compressor, V-belt drive.



Baker Shell and Tube Condensers and Liquid Coolers for Use with Ammonia or "Freon-12" (1 to 250 tons capacity)

• Made in sizes up to 2500 square feet of cooling surface in single shells. Available for multi-unit installation with special stands to allow compact installation. Supplied in either horizontal multi-pass or vertical type. Heads may be removed quickly and easily for complete cleaning. Tubes are spaced to provide even gas distribution throughout the shell. All valves of maximum size for greatest efficiency.

Baker Also Manufactures a Complete Line of Industrial-Type Cooling Units, Ammonia Valves, Screw-End Fittings, Capped Valves, Flanged-Type Fittings

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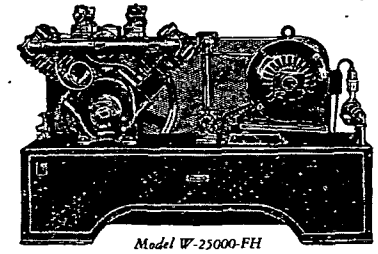
BRUNNER

COMMERCIAL REFRIGERATION



Model W-25000-FH

The Brunner Line of Refrigeration Equipment includes Air Conditioning models up to and including 25 hp for all types of high temperature applications within their capacity, using either "Freon-12" or Methyl Chloride as refrigerant.

BRUNNER DEPENDABILITY is based on time-proven features of design and manufacture . . . all parts are precision machined with extremely close tolerances . . . bronze bearings throughout . . . extra large fin surface on cylinders and heads . . . bellows seal . . . silent eccentric drive (except on 20 hp and 25 hp models, which employ crankshaft) . . . suction and discharge valves in "all-in-one" plate assembly . . . heavy-duty motor with high starting torque . . . adjustable motor base . . . multiple V-belt drive. Throughout, Brunner Refrigeration Units are geared to the demands of *heavy-duty* service.

SPECIFICATIONS					CAPACITIES Air Conditioning Units Based on 75° F Water Temperature "Freon-12" Refrigerant	DIMENSIONS
Model No.	H. P.	Cyls.	Bore and Stroke	R.P.M.	B. T. U. per Hr. 40° Evap. Temp.	L. W. H.
W 300-FH	3	4	3¼ x 2¼	260	38547	50″ 24″ 28¾″
W 500-FH	5	4	3¼ x 2¼	420	62270	50″ 24″ 28¾″
W 750-FH	7½	4	4¼ x 3	260	91526	71″ 29½″ 38½″
W 1000-FH	10	4	4¼ x 3	350	123211	71″ 29½″ 38½″
W 1500-FH	15	4	4¼ x 3	525	184815	71″ 29½″ 38½″
W 20000-FH	20	4	4¼ x 5	435	255046	73″ 33¼″ 48¾″
W 25000-FH	25	4	4¼ x 5	540	316652	73″ 33¼″ 48¾″

Additional air and water cooled models from ¼ h.p. for commercial and industrial applications.

The Brunner Field Sales Organization is available in all parts of the country, backed by outstanding achievements in engineering, and adoption of modern methods and design of air conditioning equipment.

Installation of Brunner refrigerating units is insurance of fine quality materials and workmanship—plus the exceptional efficiency possible in modern design and manufacture.

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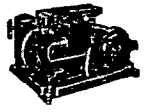
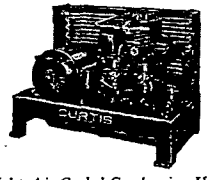
Unit Coolers and Evaporator Coils

PRODUCTS: Refrigerating Machinery; Forced Draft Cooling Units; Cooling Coils, Condensers, Shell and Tube Coolers, Valves, Fittings and Accessories, Complete Refrigerating Equipment for Dairies, Creameries, Ice Cream Cabinets, Ice Cream Making Plants, Cold Storage Locker Systems, Walk-in Coolers, Drinking Water Systems, Commercial and Low Temperature Cooling, Processing and Air Conditioning Installation, Packaged and Remote Types.

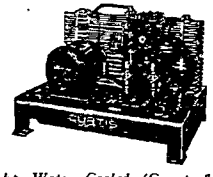
Commercial Refrigeration



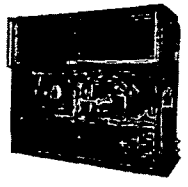
¼ to ½ hp Self-Contained Condensing Unit.



1½ hp Air Cooled Condensing Unit. Other sizes from ¼ to 3 hp.



5 hp Water Cooled (Counterflow) Condensing Unit. Other sizes from ½ to 5 hp.



7½-10-15 ton Remote or Central Type Air Conditioner.

Air cooled condensing units from ¼ to 5 hp, inclusive, and water cooled units from ½ to 30 hp, inclusive. All models available for either Freon (F-12) or Methyl Chloride. Mechanical advantages include Timken Bearings, Centro-Ring Positive Pressure lubrication.

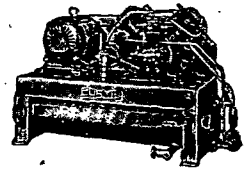
Special models are available for ice cream, frozen food cabinets and for the dairy industry.

Saturated Air Condenser

For condensing refrigerant vapors economically and efficiently. Saves approximately 95 per cent water cost. Used for air conditioning or commercial refrigeration installations up to 5 ton Capacity.

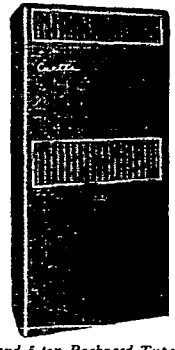
Air Conditioning

For today's essential Air Conditioning requirements Curtis offers complete packaged, refrigerated air conditioning units, requiring only water and electrical connections to install. Cools, dehumidifies, circulates and filters the air. Eliminates costly installation expense. Adaptable for heating.



15 hp Cleanable Shell and Tube Condensing Unit. Other sizes from 3 to 30 hp.

Saturated Air Condenser



3 and 5 ton Packaged Type Air Conditioner.

ALBANY
ATLANTA
BALTIMORE
BOSTON
BUFFALO
CHARLOTTE
CHICAGO
CINCINNATI
DALLAS
DETROIT
KANSAS CITY

Frick Company

(Incorporated)

Air Conditioning, Refrigerating and Ice-Making Equipment

Waynesboro, Penna.

LOS ANGELES
MEMPHIS
NEW ORLEANS
NEW YORK
OKLAHOMA CITY
PALATKA
PHILADELPHIA
PITTSBURGH
ST. LOUIS
SEATTLE
WASHINGTON

Distributors in 150 FRICK **Principal Cities**

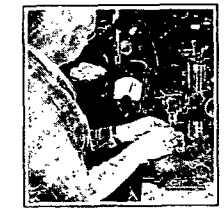
AIR CONDITIONING

Ask for Frick Bulletins 503, 504, 505 and 520 on Air Conditioning

Complete Frick systems; also refrigeration for use with equipment supplied by others. Over 1000 installations attest the value of Frick air conditioning systems and those using the Auditorium patents. Successful experience with important Government and industrial war jobs enable us to solve your problems.

AMMONIA REFRIGERATION

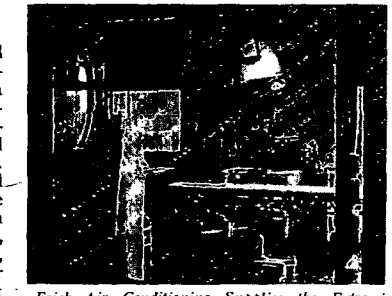
Combined units and vertical enclosed compressors, with two or four cylinders, in sizes from ½-ton up. Widely used for air conditioning, with material savings. Ask for Bul. 503 on this subject.



Pratt and Whitney use Frick Ammonia Refrigeration for Precision Air Conditioning Service

FREON-12 REFRIGERATION

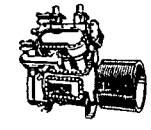
Frick "Eclipse" and the larger F-12 compressors provide a complete and efficient line. Coils, coolers, condensers and controls to suit. Patented Flexo-Seal at shaft, pressure lubrication from submerged pump, capacity controls, and other superior features make Frick machines your logical choice.



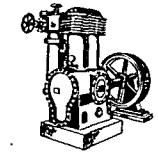
Frick Air Conditioning Supplies the Extra Humidity, Steady Operation, Spark-proof Controls, Quietness, and Freedom from Drafts Necessary in Hospital Operating Rooms

LOW-PRESSURE REFRIGERATION

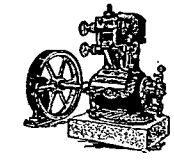
Commercial and industrial units in sizes from ¼ hp. up. Charged with either Freon-12 or methyl chloride. Air and water cooled condensers. Coils, coolers, and air conditioners. Get in touch with your Frick Distributor; ask for Bul. 97. Our service includes estimates, layouts, manufacture, installation, maintenance.



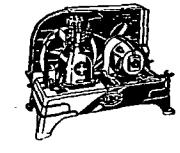
"Eclipse" Freon-12 Compressor. Bulletin 100.



Enclosed Freon-12 Machine. Bulletin 508.



Enclosed Ammonia Compressor Bulletin 112.

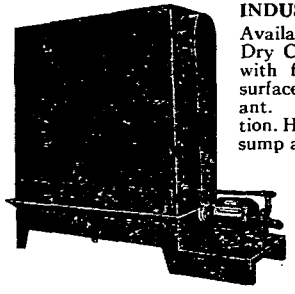


Low Pressure Refrigerating Unit. Bulletin 97.

Marlo Coil Co.

6135 Manchester Ave., St. Louis 3, Mo.

Manufacturers of Heat Transfer Equipment
Evaporative Coolers—Industrial Coolers—Unit Heaters—Unit Coolers —Evaporative Condensers—Low Temperature Units—Air Conditioning Units—Heating and Cooling Coils.

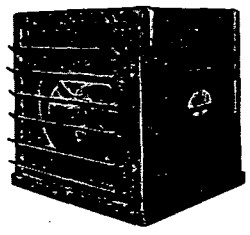


INDUSTRIAL COOLERS

Available in 15 sizes, both Dry Coil and Spray Type; with finned coil or prime surface. For any refrigerant. Sectional construction. Hot galvanized frames, sump and base.

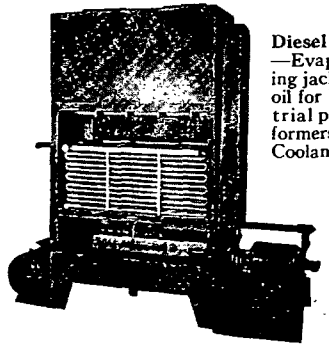
EVAPORATIVE COOLERS

Evaporative Condensers—A complete line of condensers for all refrigerants having single motor "Unidrive" for fans and pump. All prime surface coils. Frame electric welded and galvanized after fabrication. All internal surfaces covered with corrosion resistant mastic.

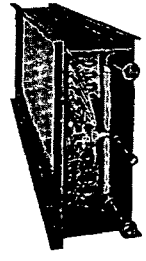


UNIT COOLERS and HEATERS

Sturdily built, for all refrigerants and heating mediums. Coolers in "Pull-Through" and "Blow-Through" types.

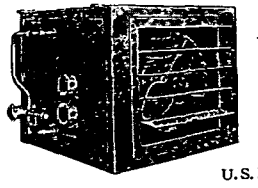


Diesel and Industrial Coolers—Evaporative coolers for cooling jacket water and lubricating oil for Diesel engines and industrial processing; also for Transformers, Quenching Oils, Cutting Coolants, etc.



AIR CONDITIONING COILS—BLAST COILS

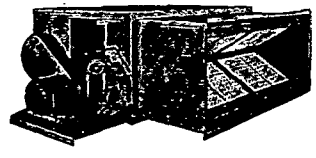
Extended surface—"Ball-Bonded"—coils for all cooling or heating mediums, suitably constructed of any conventional metal—all sizes, duties and capacities.



U.S. Patent

LOW TEMPERATURE UNIT

Designed for maintaining sub-zero temperature. Defrosted manually or automatically by electric heater units.



AIR CONDITIONING UNITS

Air Conditioning Units in either ceiling suspended or floor type. Capacities from 900 cu ft to 12,000 cu ft. Sturdily built on welded angle iron frames of sectional design for easy installation.

"MARLO MEANS HEAT TRANSFER EQUIPMENT"

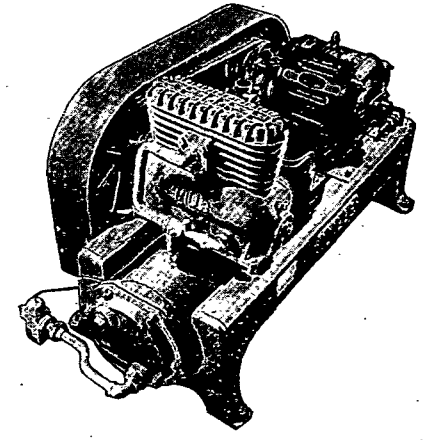
MILLS *Industries, Incorporated*

Refrigeration Division 4100 Fullerton Avenue - Chicago 39, Illinois

Mills Compression Equipment
for Air Conditioning • Commercial and Industrial Refrigeration

COMPRESSOR

2½ in. x 3 in. four cylinder, vertical, single acting, reciprocating type . . . heavy, one piece alloyed semi-steel cylinder block and crankcase . . . oil sight glass . . . alloy semi-steel pistons, two compression rings, one oil ring . . . hardened piston pins . . . drop forged steel connecting rods . . . bronze piston pin bearing . . . bronze main bearings . . . exclusive Mills design shaft seal.



CONDENSING UNIT

Base of cast iron ends and formed steel sides . . . shell and tube type receiver condenser with removable heads and leak alarm fittings . . . complete with automatic water valve, starting box, low pressure control with high pressure safety cut-out . . . complies with Underwriters' specifications . . . dynamically balanced and non-vibrating, light in weight, compact, requiring a minimum of floor space. Mills Condensing Units are economical to operate and maintain. Condensing units are shipped ready to operate. No special foundations are necessary.

APPLICATIONS

Refrigeration in air conditioning applications for comfort, food preservation, and industrial uses for processing and testing. Compactness of condensing units make them ideal for self-contained applications.

ENGINEERING

Mills Industries, Incorporated maintain an engineering test laboratory as a service to their customers, equipped with three hot rooms, calorimeter, indicating and recording, pressure and electrical instruments and all apparatus necessary for a thorough analysis of the use of condensing units to any manufacturer's product or field applications.

CAPACITIES AT 40° F SUCTION GAS

Size	RPM	Displacement CFM	Capacities in BTU	Width	Depth	Height	Net Weight
5 hp	480	16.37	65000	63"	29½"	32"	895 lb
7½hp	700	23 86	90500	63"	29½"	32"	945 lb
10 hp	890	30 34	125000	63"	30"	34"	1050 lb

Air cooled models from ¼ to 3 hp.
Water cooled models from ½ to 10 hp.

All models are designed for use with Freon-12 or Methyl Chloride refrigerants for low, standard, and high back pressures, for operation throughout the complete range of temperatures.

Compressors are designed for low head temperatures and high volumetric efficiency.

Condensers are matched to the performance ranges of each compressor.

A Member of the Mills Field Organization Is Near You And at Your Service

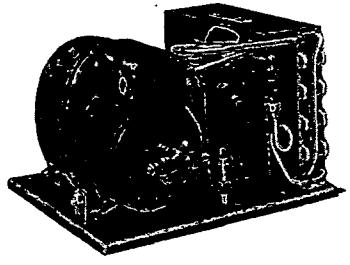
Servel, Inc.

Electric Refrigeration Division

Evansville 20, Indiana, U. S. A.

"POWERED by Servel" is the hall mark of a quality product

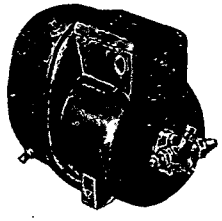
CONDENSING UNIT SPECIALISTS



Servel Supermetic Condensing Unit

The Electric Refrigeration Division of Servel, Inc., has for over 25 years, specialized in producing top quality low pressure condensing units. As the first American manufacturer to use oil soluble chlorinated hydro-carbon refrigerants, Servel has pioneered a large part of the most important engineering developments in this field.

Servel maintained its condensing unit production throughout the War Years, adding many important features and developing improved designs and technical processes.



Steel Cased Supermetic Power Unit Fractional H.P.

Participation in many of the most critical applications for war production materially accelerated the improvement of Servel products.

New machine tools and processing methods have been installed and these facilities will permit a continuous increase in production during 1946. Emphasis will be placed on volume production of SERVEL SUPERMETIC UNITS. These units, developed before the war, and suspended because of critical material shortages have been refined and improved to provide added performance, efficiency, and dependability.

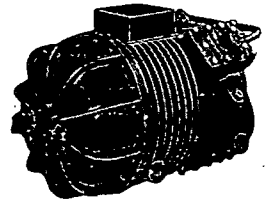
The new line of units will serve the requirements of more than 90 per cent of the refrigeration field from sub-zero **Freezers** up to air conditioning and comfort cooling. In addition to these **Supermetic** units, Servel will offer a companion line of belt driven units and compressors for special applications, D.C., odd frequency, Gas Engine drive, etc.

FOR DISTRIBUTORS AND CONTRACTORS—Servel sells its condensing units throughout the World through regularly franchised dealers and distributors serving a specific trading area. Full selling, Engineering, and service assistance is extended these customers by Servel's field force and factory staff. Servel's National advertising assures acceptance everywhere.

FOR MANUFACTURERS—We especially invite inquiries at this time from manufacturers who contemplate the production of window units, room coolers, store coolers, etc.

Our Engineering Department will be glad to cooperate in a selection of components and extend testing facilities for completed units. Samples of our manufacturing products are available to responsible manufacturers and contractors without obligation. Our field sales representatives and engineers will be glad to make appointments for consultations on these matters upon request.

Address Servel, Inc., Electric Refrigeration Division, Evansville 20, Indiana.



Cast Case "Field Serviceable" Supermetic Power Unit—1 H.P. and above

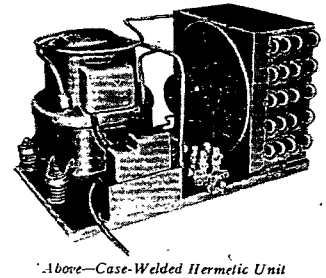
UNIVERSAL COOLER

Division

INTERNATIONAL DETROLA CORPORATION

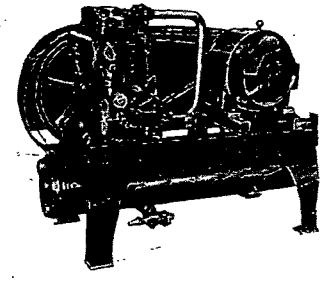
MARION, OHIO
and in Canada
BRANTFORD, ONT.

WE SELL TO MANUFACTURERS ONLY



Above—Case-Welded Hermetic Unit

Below—Universal Cooler Remote-Type Unit



Manufacturers and designers of *hermetic*, open type self-contained and remote types Refrigerating Units in wide range of capacities.

⅛ to 15 hp for . . .

- **Household Refrigerators**
- **Frozen Food Cabinets**
- **Food Storage Refrigerators and Display Cases**
- **Ice Cream Cabinets and Soda Fountains**
- **Water and Beverage Coolers**
- **Air Conditioning**
- **Commercial Refrigeration Equipment**
- **Truck Refrigeration Equipment**
- **Vending Machines**
- **Machine Tool Cooling**
- **Special Applications**

Low-cost, trouble-free performance—that's what every fixture manufacturer is looking for. And that's the kind of performance you can expect from Universal Cooler's compact, soundly designed and precision-built refrigerating units.

From the experience gained through a quarter century devoted exclusively to the manufacture of refrigerating units, Universal Cooler has developed advanced design and exclusive construction features, proved in actual field service, that have established new performance standards in the refrigeration industry.

For the economical, dependable service that will provide the answer to your particular refrigeration problem, look to Universal Cooler! An analysis of your particular requirements will be made by Universal Cooler specialists without obligation; write today for complete information on products and production.

Automatic Refrigeration Since 1922

Worthington Pump and Machinery Corporation

Air Conditioning and Refrigeration Division

General Offices: HARRISON, NEW JERSEY

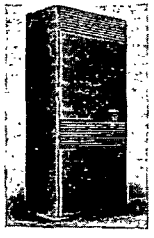
ALBANY	CHARLOTTE	DETROIT	LOS ANGELES	PITTSBURGH	SAN FRANCISCO
ATLANTA	CHICAGO	EL PASO	MILWAUKEE	PORTLAND, ORE.	SEATTLE
BALTIMORE	CINCINNATI	FORT WORTH	NEW HAVEN	PROVIDENCE	SPRINGFIELD, MASS.
BIRMINGHAM	CLEVELAND	GALVESTON	NEW ORLEANS	ST. LOUIS	SYRACUSE
BOSTON	DALLAS	HOUSTON	NEW YORK	ST. PAUL	TULSA
BUFFALO	DENVER	KANSAS CITY	PHILADELPHIA	SALT LAKE CITY	WASHINGTON, D.C.
					WILMINGTON, DEL.

Representatives in all Principal Cities

REFRIGERATION SYSTEMS FOR AIR CONDITIONING

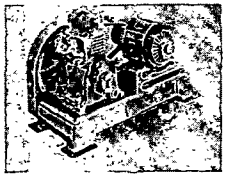
Complete refrigerating systems for use with Freon-11, Freon-12, Methyl Chloride, Ammonia, or Carbon Dioxide, either direct-expansion or water cooling applications. A complete line of refrigeration compressors, permitting impartial recommendations. A nation-wide organization of Distributors in major cities to provide sales and engineering service and plan complete air conditioning systems of the central or unit type. Architects, Engineers, and Contractors are invited to consult with us. Write to Harrison, N. J., or any branch office, for bulletins on these products.

"Packaged" Unit



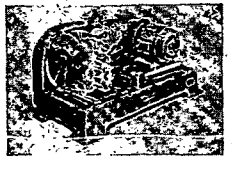
Cooling (or heating, if desired), dehumidification, ventilation, circulation and air cleaning for commercial and small industrial applications. 3 hp and 5 hp sizes. Two-cyl. reciprocating compressor, finned copper tubing condenser, large-surface finned cooling coils, quiet low-speed fan, throw-away filters, concealed temperature and air volume controls. Freon-12. Capacities up to 5 tons.

Small Self-contained Units



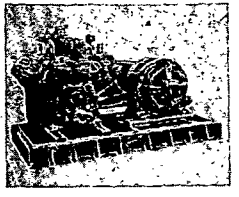
Freon-12 or methyl chloride condensing units; motors ¼ to 2 hp with air or water-cooled condensers. Used in small air conditioning systems, and in commercial refrigeration. Capacities up to 2 tons.

Medium Self-contained Units



Freon-12 or methyl chloride compressor units for use with "shower" condensers or water-cooled condensers. Features: FEATHER (Pat'd.) Valves; automatic capacity control. Capacities 3 to 30 tons.

Large Self-contained Units



Freon-12 or methyl chloride compressor units for use with "shower" condensers or water-cooled condensers. Features: Worthington FEATHER (Pat'd.) Valves; automatic capacity control. Capacities up to 100 tons.

Centrifugal Refrigeration

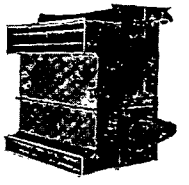
Water Cooling Systems

Freon-11 centrifugal compressor, water cooler and water-cooled condenser in compact unit assembly. Electric motor or steam turbine drive. 56 unit sizes . . . 150 to 1200 tons.

Worthington Pump and Machinery Corporation

Evaporative Type Jacket—Water Cooler

(With By-Pass Section For Automatic Temperature Control.)



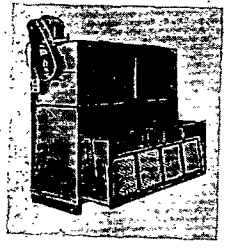
Cooling jacket water for diesel and gas engines, air compressors, etc. Ideal for the cooling of quenching oil for tempering steel products. Also for cooling transformer oil to reduce core loss.

Miscellaneous

High and low side equipment for every purpose.

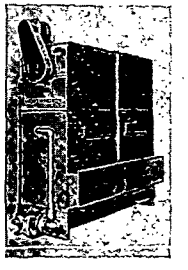
Air Conditioning Units

For Direct Expansion Freon-12 or Chilled Water Circulation



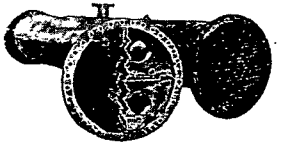
Vertical and horizontal; 1500 to 12,000 cfm; large air passages; slow speed, quiet, rugged fans; separable sections; readily accessible. The design permits flexibility in installation arrangements.

Shower Condensers



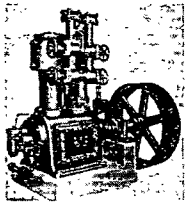
A combined condenser, receiver and modified cooling tower, in one assembly, for Freon-12 or methyl chloride systems; 10 to 60 tons refrigeration; built in separable sections; all parts easily accessible. Saves 90 to 95 per cent in cost of water.

Horizontal Condensers



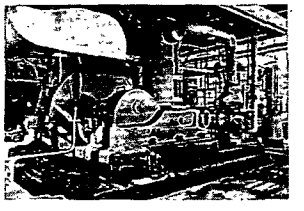
Atmospheric drip type, for warm corrosive waters. Double-pipe for closed systems, can be retubed without shutting down. Multi-pass, as illustrated above, for closed systems and space saving.

Vertical Ammonia Compressors



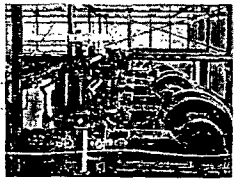
Pressure-lubricated; roller main bearings; safety heads; patented Feather Valves; belt drive, or direct-connected to electric motor, diesel or gas engine; sizes 3 x 3 to 10 x 10 2 cylinder.

Horizontal Ammonia Compressors



Single and duplex; single-stage and two-stage; belt drive, or direct-connected to electric motor, diesel, gas or steam engine; patented Feather Valves; ratings from 60 to 750 tons. Automatic capacity control features are easily applied. Space requirements vary depending upon type and drive.

Carbon Dioxide Compressors



A series of convenient types and sizes for every requirement is available.

A6—7

Vilter
Air Conditioning • Refrigeration

The Vilter Manufacturing Co.

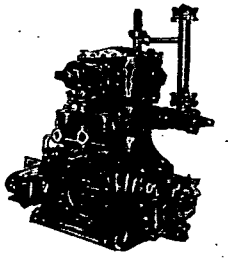
Dept. N-23

Milwaukee 7, Wisconsin

AIR CONDITIONING EQUIPMENT FOR INDUSTRIAL OR COMFORT COOLING

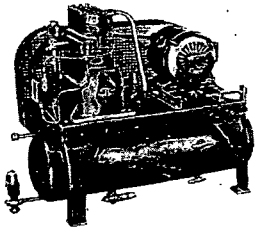
Branch Offices and Distributors in Principal Cities

Ammonia and Freon Compressors



Shell and Tube Equipment

Ammonia, Freon, Methyl Chloride Self-Contained Condensing Units



Ammonia Compressors—The result of over seventy years of research, development and experience gained through thousands of installations of all types, in all industries. Famous for high tonnage capacity at low HP and low operating costs. Built in a wide range of capacities from 2 to 100 tons standard *A.S.R.E.* rating in Vertical Types; up to 750 tons in Horizontal Types.

Freon Compressors—Embody many outstanding features that prevent leakage and minimize friction—resulting in extremely low relative HP per ton. Made in capacities up to 150 tons. Capacitrols are available, providing flexibility of operation.

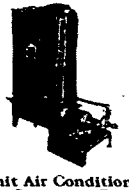
Self-Contained Condensing Units—Vilter self-contained condensing units for Ammonia, Freon or methyl chloride are made in sizes from ¼ to 30 HP. Dependable, economical operation.

Unit Air Coolers—Available in a wide range of sizes and types for any air conditioning requirement—product coolers, dry coil coolers, spray type coolers, low temperature electric defrosting coolers, and floor or ceiling central system air conditioners.

Water Coolers and Condensers—A complete line of shell and tube and double pipe water coolers, brine coolers and condensers for Ammonia or Freon.

Air Conditioning—All sizes and types of air conditioning coils, evaporative condensers and air washers—and special units for central station comfort cooling systems.

Valves, Fittings and Piping—Ammonia and Freon valves and fittings. Prefabricated piping in all sizes for refrigeration and special process work.



Unit Air Conditioners Spray and Dry Coil Types

Catalog—Valves, Fittings, Accessories Write for Free Copy

Prefabricated Process and Refrigerant Piping

Air CONTROLS INC
Div. of
The Cleveland Heater Co.

2311 Superior Ave., **Cleveland 14, Ohio**

PIONEER BLOWER MANUFACTURERS

REX BLOWERS

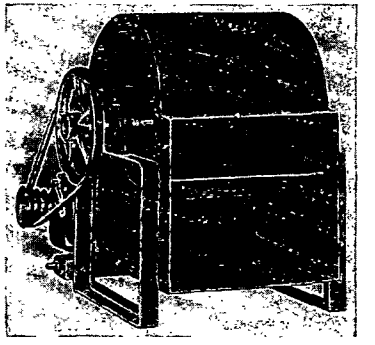
A complete line of centrifugal blowers well suited for use in the heating, cooling and ventilating field. There are two series of Rex Blowers—the A. C. and H. P. models, ranging in capacity from 400 to 12,000 cfm. Twin units double these volumes. The H. P. series is more compact for use where space requirements are a factor.

WHEELS: Double Inlet, carefully balanced, assuring true running and high efficiency. Forward curved multi-blade type for maximum air delivery at lowest speeds. Each blade is die formed and securely riveted to inlet rings to form a rigid assembly.

HOUSINGS: Made of heavy gauge steel, die formed and electrically welded for maximum air intake. Wide outlet distributes the air evenly.

BEARINGS: Self-aligning to insure smooth operation; mounted on sound and vibration absorbing rubber bumpers. Phosphor-bronze bushings are self-lubricating to supply right amount of oil.

OTHER FEATURES: Motor pulleys on smaller sizes have four steps to provide four speeds. Variable speed supplied on 16 in. size and up. V-Pulleys universally used.



Rex Blower

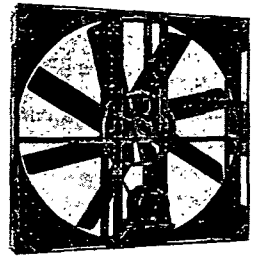
Shafts are accurately ground and polished to 0.0005 in.; of heavy weight to prevent whip which causes noise and wear.

Angles of discharge can be varied and construction changed to meet special requirements. REX Blowers are rated in accordance with the test code of the A.S.H.V.E. Performance is quiet, dependable and trouble-free.

For complete information, write for *Catalog No. 222* which contains a complete description of REX Blowers and tables of dimensions, capacities and other data.

American Coolair Corporation

3606 Mayflower Street, **Jacksonville 3, Florida**
Manufacturer of COOLAIR Ventilating and Exhaust Fans
A Pioneer Manufacturer of V-Belt Drive Exhaust Fans
Charter Member: Propeller Fan Manufacturers Association



TYPE S—6 TO 9 FEET

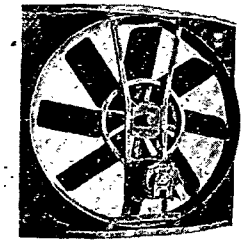
Especially designed for ventilating and cooling in industrial plants and shops, power stations, warehouses and other large buildings—the Type S fan has a heavily braced double frame, special pillow-block ball bearings on each side of fan wheel. 8 to 12 reinforced fan blades and multiple heavy duty V-belts for long industrial service. This fan is usually installed in roof bay or gable, penthouse or outside wall.

RAPID AIR CHANGE NEEDED FOR EFFECTIVE COOLING

Unprecedented production demands in recent years have highlighted the necessity for *rapid air change* in factories, shops and offices to combat heat fatigue and maintain morale among workers. Many plant managers have only recently learned that a complete air change every 30 seconds to one minute is necessary for effective cooling. Coolair pioneered this principle and developed large, efficient exhaust fans to do the job.

Coolair also has pioneered in developing a complete line of ultra-quiet home-cooling attic fans. A wide range of sizes and types are stocked by Coolair dealers and distributors in most sections of the country.

In planning a Coolair installation, determine cubic content of space to be cooled and select fan from table on opposite page adequate to provide at least a one minute air change. Write for FREE bulletins containing detailed suggestions for industrial and commercial installations —and for name of nearest Coolair agent or dealer.



TYPE O—26 TO 62 INCHES

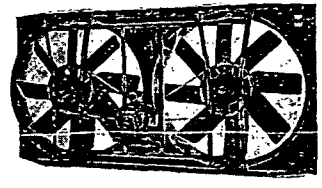
Patented built-in springs (on smaller motored sizes) and streamlined orifice combine to make this type quiet in operation. Designed for reversible operation it is equally efficient for vertical or angular discharge when equipped with ball bearing motor. Usually installed in full or section of window opening, outside wall, partition, skylight opening, penthouse—often used in a battery of units for most efficient cooling of large buildings. Covered by U. S. Patents 1992112 and 2191418.

QUALITY FEATURES OF COOLAIR EXHAUST FANS

1. Built-In Springs on Smaller Motored Units—dampen vibration and practically eliminate noise for quiet installations by insulating moving parts from frame. **2. Light, Compact Fabricated Steel Frame**—fits into many openings where bulky metal housings cannot be used. **3. Reversible**—when equipped with reversible motor, fan will blow in or exhaust as desired. **4. Ball Bearings in Fan Hub**—eliminate sleeve bearing chatter and end thrust knock—permit operation in any position (specify ball bearing motor for vertical or angle discharge). Use grease, instead of oil, requiring attention not more than once a year. **5. Eight Large, Slow-Moving Steel Blades**—instead of 4 or less as on cheaper fans—up to 12 blades on Type S models. Low tip speeds for quiet operation and steady flow of air. **6. V-Belt Drive**—for high efficiency using small motor for economical operating speed. **7. Long Hour Service Motor**—Nationally known makes of motors. **8. Certified Air Ratings**—in accordance with Standard Test Code of A.S.H.V.E. and N.A.F.M. **9. Full Streamlined Orifice**—(on Type O & OT) avoids "spill-off" at end of blades, reduces power consumption.

TYPE OT—TWIN UNIT

This unique Coolair Twin-Unit is two fans of Type O specifications mounted side by side in one frame and operated by a single motor. Widely used where limited headroom or vertical wall space will not permit the use of a single fan large enough for the job. Especially adapted for installation in partitions, outside wall and on end can be fitted into existing window and door openings. Covered by U. S. Patents 1992112, 2108738 and 2191418.



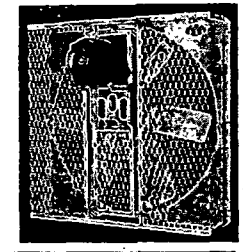
Dimensions in Inches

Fan Size	Overall Height	Overall Width	Overall Depth (Approx)
28-W	31	33	15
2-O	30⅜	30⅜	16
2½-O	36⅜	36⅜	18
3-O	42⅜	42⅜	18
3½-O	49	49	19
4-O	55⅛	55⅛	19
4½-O	61⅛	61⅛	19
5-O	67⅜	67⅜	19
6-S	75¼	75¼	30
7-S	87	87	32
8-S	99⅝*	99⅝*	49
9-S	112⅛*	112⅛*	54
2-OT	30⅜	61¼	18
2½-OT	36⅜	73¼	18
3-OT	42⅜	85¼	19
3½-OT	49	98	19
4-OT	55⅛	110¼	22

*These dimensions include five-eighths inch for bolt heads.

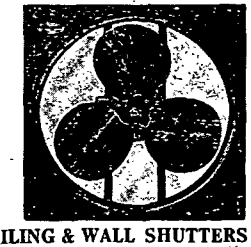
28-W FAN WITH SAFETY GUARD

Coolair's lowest priced belt-drive fan is equipped with built-in springs, adjustable diameter motor pulley and safety guard. This fan can be easily and quickly installed in upper or lower half of any standard window of work rooms or offices where proper ventilation is necessary for health, comfort and efficiency of workers. See tables for data.



DIRECT DRIVE FANS

Four sizes 16 to 24 inches in diameter. General purpose exhaust fan for most commercial and industrial uses. Data on sizes, performance and dimensions furnished on request.



COOLAIR AUTOMATIC CEILING & WALL SHUTTERS

Precision-built all-steel shutters that open and close automatically when fan is turned on or off. Ceiling shutters eliminate need of ceiling grille and trap door in attic installations. Wall shutters give weather protection for fans discharging directly to open air.

Performance Data—Coolair V-Belt Drive Fans

Fan Size		Horse Power	Fan R. P. M.	Cubic Ft. Air Per Min.
28-W	VL	¼	580	6000
2-O	VL	¼	570	6200
	D	½	630	6800
2½-O	VB	¼	411	8000
	VC	½	454	8800
	D	½	522	10100
	D	¼	600	11700
3-O	VB	¼	312	10000
	VC	½	345	11000
	C	½	398	12700
	D	¾	450	14400
	D	1	500	16000
3½-O	VB	½	261	13000
	VC	½	300	15000
	C	¾	345	17000
	D	1	380	19000
	D	1½	440	22000
4-O	VL	½	258	19000
	VC	¾	317	22000
	D	1	353	25000
	D	1½	405	28000
4½-O	VB	½	224	22800
	VC	¾	255	25000
	C	1	276	27000
	D	1½	319	32000
	D	2	355	35000
5-O	VB	½	200	27000
	VC	¾	225	30000
	C	1	245	33000
	C	1½	282	38000
	D	2	310	42000
	D	3	355	48000
6-S	B	1	155	35000
	C	1½	177	40000
	C	2	195	45000
	D	3	224	50000
	D	5	270	60000
7-S	B	2	170	58000
	C	3	195	67000
	D	5	230	76000
	D	7½	270	85000
8-S	B	3	150	80000
	C	5	178	95000
	D	7½	205	110000
	D	10	225	120000
9-S	B	5	150	110000
	C	7½	170	125000
	D	10	190	138000
	D	15	215	154000
2-OT Twin	VB	¼	455	9800
	VC	½	500	10800
	VC	½	570	12400
2½-OT Twin	VB	½	359	14000
	VC	½	411	16000
	VC	¾	470	18200
3-OT Twin	VB	½	312	20000
	VC	¾	360	23000
3½-OT Twin	VB	¾	272	27800
	VC	1	300	30700
4-OT Twin	VB	1	258	38000
	VC	1½	317	44000
	D	2	345	48000

B—Very Quiet (Homes, Theatres, Hospitals, etc.).
C—Quiet (Stores, Offices, Restaurants, Barber Shops, etc.).
V—Equipped with built-in springs.
D—Industrial (Laundries, Factories, Canneries Bakeries, Pressing Clubs, Garages, etc.).
L—Has adjustable diameter motor pulley for Very Quiet and Quiet performance.

Bayley Blower Company

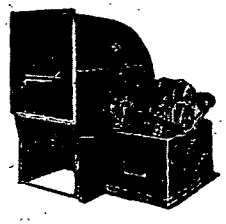
1817 S. Sixty-Sixth Street **Branches in Principal Cities** **Milwaukee 14, Wis.**

Builders of Heating, Ventilating, Cooling, Purifying, Humidifying and Air Washing Equipment; Exhaust and Drying Apparatus, Mechanical Draft and Blast, Fans and Blowers of all Types

TYPE "F" PLEXIFORM FANS AND TYPE "AP" AEROPLEX FANS

Plexiform and Aeroplex Fans are both designed for heating, ventilating and air conditioning service. The **Type "F"** fan is using a multi-blade wheel (see cut at left) while the type **"AP"** fan is built around a wheel of high speed design (see cut at right.)

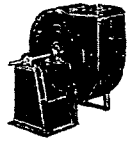
The wheels determine the character of the performance or the fan "Characteristics," thus the Type "F" is a slow speed fan having a rising power curve, while the **"AP"** is a high speed design and has self-limiting power characteristics. Both fans are highly efficient and can be built single inlet, single width, or



double inlet, double width in ten different arrangements of drive and eight directions of discharge. The standard fans range in capacity from 1200 to 300,000 Cfm and these fans are readily available in Class I or II designs for various types of duty. Class III and IV can be furnished on special application.

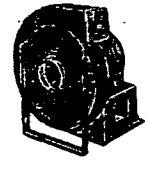
Type "EX" Fans

For all exhaust problems, pneumatic conveying, fume exhaust etc. **"EX"** is highly adaptable, feasible and serviceable fan. It is sturdy in construction, easily installed and maintained. The design is reversible and standard sizes Nos. 15 to 80 inclusive are available.



Type "H" Fans

For pressure applications from 6 in. to 42 in. for belt drive or direct connection to motor, **Type "H"** is a most practical design. Standard sizes Nos. 25 to 80 are listed but the design is suitable for modification of wheel diameters to meet various motor speeds.



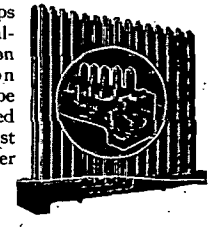
Both **"EX"** and **"H"** fans can be designed to fit specific applications. The design permits easy modification of details and the fans can be used for induced, forced draft, cupola service, primary or secondary air supply to oil burners, etc. Special details, outlet and inlet, or mounting can be designed to suit your special assembly.

Turbo Spray

The Turbo Spray for industrial and comfort conditioning, cooling and humidifying, is a practical design as the atomization is by mechanical means, thus there can be no clogging or interruption of service even in atmosphere heavily laden with fibers or other foreign material. Single or multiple bank washer, one or two stage designs can be furnished from 2500 to 100,000 Cfm capacity. To fit space requirements width and height can be modified to suit.

Chinook & Chinookfin

Both **Chinook and Chinookfin Coils** are based on the tube-within-a-tube design, originated by Bayley. The principle involved permits single header with all tube ends free expanding, thus eliminating the likelihood of freeze ups and cracking usually due to expansion and contraction strains. Both type coils are designed for usual hot blast heating and cover a wide range of capacities and applications.



Burden Company

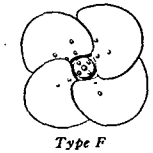
1151 South Broadway, **Los Angeles 15, Calif.**

Manufacturers of

Plastic and Aluminum Fan Blades and Blower Wheels

Burden Plastic Fan Blades offer the following unique combination of desirable properties:

LOWER NOISE LEVEL—STABLE OVER WIDE TEMPERATURE RANGE—LOW MOISTURE ABSORPTION—RESISTANT TO CORROSIVE ACTION—LIGHTWEIGHT WITH GREAT STRENGTH.

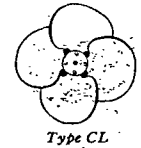


Type F

AVAILABLE IN THREE TYPES

TYPE "F". Standard with four flat, overlapping, plastic blades with steel hub and aluminum spider. Sizes available: 8, 10, 12, 14, 16, 18, 20, 22 and 24 in. diameters.

TYPE "C". This is like our Type "F" except that the blades have a curved tip. Available in same sizes as Type "F". Both plastic and aluminum.



Type CL

TYPE "CL"—CLOVER LEAF PATTERN—(Patent applied for). This blade is made without a spider. Blade attaches directly to hub. Available in 6, 8, 9, 10 and 12 in. diameters, both plastic and aluminum.

Types "F", "C" and "CL" are "Patented" Overlapping Design giving quiet performance under pressure conditions.

SPECIAL BLADES—When quantities justify die costs, will produce special types. We invite your inquiries.

Special One Piece Blade

SPECIFICATIONS

Pitches TYPE "F" and "C"—14, 16, 18, 21, 23, 25, 27, 30 and 33 deg.
TYPE "CL"—21, 23, 25, 27, 29, 30, 33 and 35 deg.

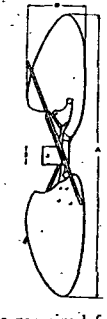
Rotation CLOCKWISE (C.W.) or COUNTER CLOCKWISE (C.C.W.) as you face air stream. Specify when ordering.

Bore Sizes STANDARD—¼, 5/16, ⅜ and ½ in. Larger sizes available.

Hubs STANDARD—For motor on suction side of blade.
REVERSED—For motor on discharge side of blade.

DIMENSIONS TYPE "F" BLADES

Fan Size		Dimension		Fan Size		Dimension	
Dia. In.	Pitch Degrees	"A"	"D"	Dia. In.	Pitch Degrees	"A"	"D"
10"	14°	10"	1 3/16"	14"	25°	14"	3 1/2"
10"	21°	10"	2 1/4"	14"	33°	14"	3 7/16"
10"	23°	10"	2 5/8"	16"	14°	16"	2 5/8"
10"	33°	10"	3 1/4"	16"	21°	16"	3 5/8"
12"	14°	12"	2 1/8"	16"	25°	16"	3 7/8"
12"	21°	12"	2 5/8"	16"	33°	16"	4 1/8"
12"	25°	12"	2 7/8"	18"	14°	18"	3"
12"	33°	12"	3 3/8"	18"	21°	18"	3 7/8"
14"	14°	14"	2 7/8"	18"	25°	18"	4 1/4"
14"	21°	14"	3 5/16"	18"	33°	18"	5 1/4"



Standard Hub—All Type "F" Blades "S" = ⅞ in. Length 1¼ or ¾ in.

Measurements are for preliminary purposes. If close measurements are required for layout purposes, we suggest working from full size blade that is being used.

For dimensions of unlisted sizes and for other types of fans, write the factory.

Burden Blades—Licensed Under General Electric Co. Patent No. 1,957,237.

Belanger Fan & Blower Co.

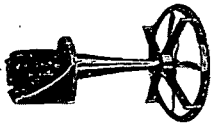
Manufacturers Wolverine Fans & Blowers

Detroit 16 1230 Eighteenth Street Michigan

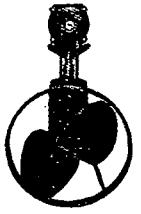
The Wolverine Silentair Exhaust Fan is the result of over 20 years experience in the design and manufacture of air handling equipment. Silentair Fans are designed for installation on walls, ceilings and partitions to provide high operating efficiency and silent performance. Fan blades are of epicycloidal type and are rated in accordance with Standard Test Code for Axial Fans adopted by N.A.F.M. and A.S.H.V.E. Made in ten sizes from 10 in. to 48 in. delivering from 650 to 18,500 cfm.

Silentair Exhaust Fan

Wolverine SKS Stack Fans V Belt Drive are ideal for spray booths, ducts or stacks for exhausting vapor, smoke, steam or fumes. Made for pipe sizes of 13 in. to 49 in. V Belt Drive and adjustable motor support. Motors are outside the air stream.



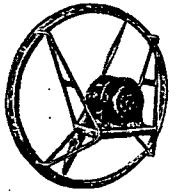
Extension Shaft Fan



SKS Stack Fan

Wolverine Silentair Belt Drive Fans designed for comparatively low speeds and provide low cost volume exhaust. Fan sizes from 15 in. to 48 in. with 1,465 to 23,500 cfm capacity.

The Wolverine Stack Exhauster transforms any roof ventilator into an effective fan exhauster. Made in a wide range of sizes and capacities. Stack Exhausters are constructed of heavy gauge steel, reinforced for motor support. The propellers are directly connected to the fan duty motors.

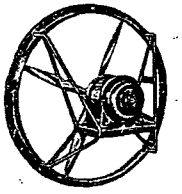


Airplane Type Single Propeller Fan

Silentair Beltdrive Fan

Wolverine Extension Shaft Fans are desirable where smoke, grease, corrosive fumes or explosive vapors must be removed. Motor is out of the air stream. Sizes from 16 in. to 48 in. with capacities from 2,280 to 25,500 cfm.

The Wolverine Single-Propeller Airplane Type Fan operates efficiently in either exhausting or blowing positions. Welded steel construction, cast aluminum propeller. Sizes range from 12 in. to 60 in.

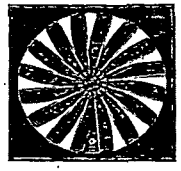


Airplane Type Two Propeller Fan

The Wolverine Two-Propeller Airplane Type Fans are made in 16 in. to 60 in. sizes.

Wolverine Stack Exhauster

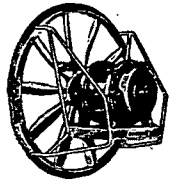
The Wolverine Multi-Blade V Belt Drive Fan is made in sizes 30 in. to 84 in. with capacities from 6,275 to 72,500 cfm. The large blade area of this fan permits comparatively slow operation and unbelievably low operating noise level. It is a quiet, powerful, economical fan of rugged construction and can be installed easily in wall openings or steel sash of industrial or commercial buildings or as attic fans in dwellings.



Multiple Blade V Belt Driven Fan

Belanger Fan & Blower Co.

Manufacturers Wolverine Fans & Blowers



Airplane Type Multi Propeller Fan

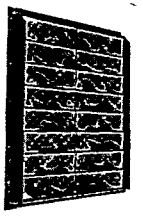
Multi-Propeller Airplane Fans are made in sizes 12 in. to 36 in., with 6, 8 and 10 propeller blades and capacities from 985 to 13,000 cfm.

High Boy

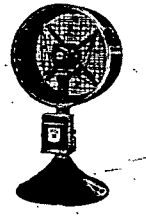
The Wolverine Roof Fan is a complete unit for easy installation on new or old buildings. The low installation and operating costs are features that appeal to engineers, architects and contractors. Adaptable to roofs with angles up to 45 degrees. Provides positive low cost ventilation. Propellers are made with 12 to 20 blades driven by ball bearing motors. Unique damper design assures automatic and positive opening and closing action. Fan sizes from 30 in. to 60 in., and capacities from 5,560 to 45,275 cfm.

Roof Fan

The Model H. D. C. Cooler Blast Fan is a heavy duty portable fan for air circulation and man cooling purposes. Affords an economical and positive means of cooling where workmen are subjected to intense heat or fumes. Available with standard Airplane type or Silentair propeller. Made in fan sizes from 28 in. to 48 in.



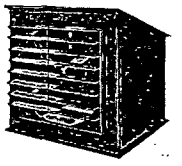
Automatic Shutter



Model HDC Cooler Blast Fan

The Model S. C. A. Cooler Blast Fan is a powerful utility fan, designed to operate on floor, wall or ceiling, adjustable to direct air flow at various angles. Useful in foundries and factories. Cfm free air delivery 6,400 to 16,750.

The Highboy Cooler Blast Fan is portable and adjustable; minimum height 6 ft 1 in., maximum height 9 ft 4 in., with free air delivery of 4,600 cfm. Useful in hot spots in plants. Also for offices, stores and homes.



Pent House

Model SCA Cooler Blast Fan

Air-Flo Shutters and Dampers are designed to work with any fan or blower, and prevent entrance of wind, rain, snow or insects. Among the several types are: Automatic, Hand and Motor-operated; types of dampers are Automatic backdraft, Automatic Stack, Balanced Ceiling; also Friction, Bypass, Volume, Mixing and Louvre Dampers.

Air-Flo Pent Houses with automatic shutters are made for fan sizes from 15 in. to 60 in.

Complete catalog with detailed dimensions and capacity tables may be obtained by writing for data on Wolverine products, and helpful assistance is available in planning a Wolverine Fan Installation.

Buffalo Forge Company

450 Broadway, Buffalo, N. Y.

Sales Representatives

ALBANY 7, N. Y.—Mr. R. B. Taylor....1303 Standard Bldg.
ATLANTA, GA.—Mr J. J. O'Shea.....305 Techwood Drive
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BOSTON 76, MASS.—Mr. E. D. Johnson, 507 Main St., Melrose Station
CHICAGO 6, ILL.—Emmert & Trumbo.....20 N. Wacker Drive
CINCINNATI 2, OHIO—Mr. F. W. Twombly.....626 Broadway
CLEVELAND 13, OHIO—Mr. T. A. Weager, 418 Rockefeller Bldg.
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DAVENPORT, IOWA—D. C. Murphy Company, Inc., 305 Security Bldg.
DES MOINES 9, IOWA—D. C. Murphy Company, 214 Old Colony Bldg.
DENVER 17, COLO.—Hendrie & Bolthoff Mfg. & Supply Co., Box 5110 Terminal Annex D
DETROIT 16, MICH.—Coon-DeVisser Co., 2051 W. Lafayette Blvd.
GREENVILLE, S. C.—Mr. Roy A. Stipp....1 Longview Terrace
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KNOXVILLE 21, TENN.—Mr. C. F. Sexton, P. O. Box 2224, 702 Empire Bldg
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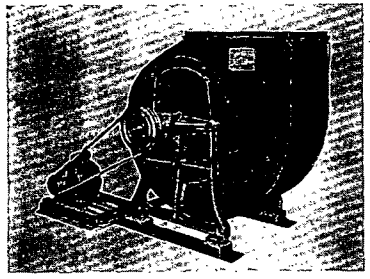
NEW YORK 7, N. Y.—Koithan & Johnson....39 Cortlandt St.
NEWARK 2, N. J.—Mr. G. C. Norman, 27 Washington St., Room 203
OMAHA 2 NEB.—Wain Engineering Co., 300 Brandeis Theatre Bldg.
PHILADELPHIA 2, PA.—Davison & Hunger, 1200 Cunard Bldg.
PITTSBURGH 22, PA.—Mr. H. Lee Moore.....431 Fulton Bldg.
PORTLAND 5, ORE.—Arthur Forsyth Company, c/o Alex J. Papulski, 921 S. W. Oak St.
ST. LOUIS 3, MO.—Mr. J. W. Cooper.....2726 Locust St.
SALT LAKE CITY 1, UTAH—Lee Pace & Turpin, 142 S. Fifth West St.
SAN ANTONIO 6, TEXAS—Langhammer Rummel Co., 436 Main St.
SAN FRANCISCO, CALIF.—Buffalo Forge Company, Charles W. Lockhart, 1200 Central Tower Bldg., Third and Market Sts.
SEATTLE 4, WASH.—A. T. Forsyth Company, A. T. Forsyth. 500 First Ave., South
TOLEDO 2, OHIO—C. M. Eyster Company..1118 Madison Ave.
TUCSON, ARIZ.—Tidmarsh Engrg. Co.....23 N. Main St.
WASHINGTON 5. D. C.—Buffalo Forge Company, Mr. G. S. Frankel, 512 Woodward Bldg.
WILKES-BARRE, PA.—Power Engineering Co., 517 Brooks Bldg.
KITCHENER, ONT., CANADA—Canadian Blower & Forge Co.

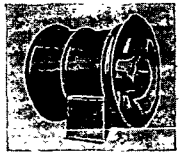
PRODUCTS: Heating and Ventilating Equipment including: Unit Heaters, Multiblade Fans, Pipe Coil Heaters, Buffalo Air Washers, Buffalo Unit Air Washers, Buffalo Unit Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Exhaust Fans, Blowers, Dust Collectors, Disc Fans, Spray Nozzles.

Buffalo Limit-Load Fans



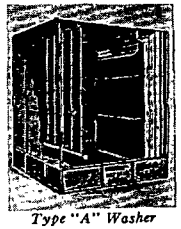
Buffalo Limit-Load Fans for ventilating embody several improvements to deliver stepped-up efficiency under practical conditions. Durably built for years of service. Dynamically balanced. Quiet, economical to operate. Non-overloading characteristic prevents motor from overloading and burning out, regardless of fan load.

Axial Flow Fans



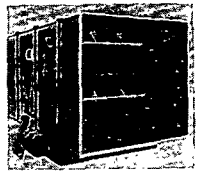
This non-overloading high pressure fan—with directional guide vanes—propels the air stream in a true axial direction. Energy losses are reduced to a minimum with a marked power saving.

Buffalo Air Washers



Type "A" Washer

Buffalo Air Washers and Humidifiers are built for the most efficient washing, purifying and tempering of air under varying atmospheric conditions. Designed and constructed for low cost installation and maintenance. Careful attention to structural details insures long service-ability with a minimum of attention. Several types and models to meet specific plant requirements.



Wet Filter Washer

Fans for Every Ventilating Need

Buffalo Fans represent over 60 years of specialization in the design and construction of fans for practically every ventilating and air-handling application from small kitchen fans to rugged fans for boiler draft. Complete information on request.

Buffalo Unit Heaters

"Highboy" and "Lowboy" and "Breezofin" Units for floor, wall or ceiling installation. Sizes for all buildings of all types.

Champion Blower & Forge Co.

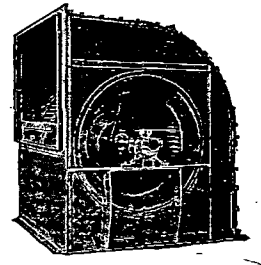
Manufacturers and Engineers

Plant and Offices: Lancaster, Pa.

Address Correspondence to Div. 9

Manufacturers of Blowers, Ventilating Fans and Exhaust Fans for Air and Material; and Blast Gates

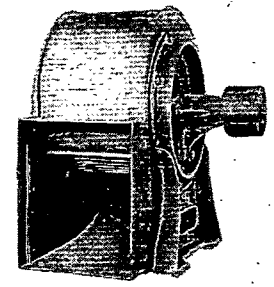
Representatives in Principal Cities



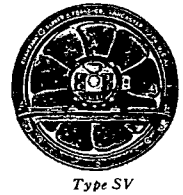
Type S

Type S Forward curve ventilating fans, single and double width, as well as direct motor drive.
Sizes 30 in. to 60 in. wheel dia.

Type S Forward curve ventilating fans, single and double width, and electric drive up to 36 in. wheel diameter.



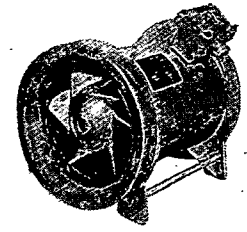
Type S



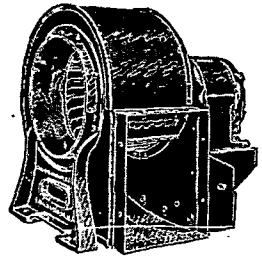
Type SV

Type SV Super Ventilating fans, direct motor drive up to 36 in. diameter. Motor belt drive up to 48 in. size.

Type M Axial Flow non-overloading fans for high and low pressures. Exhausting and Ventilating.



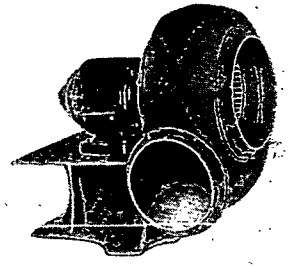
Type M



Type BC

Type BC Backward curve ventilating and exhaust fans, single and double width; belt driven and direct connected electric.
Sizes 12 in. to 60 in. wheel dia.

Type CE Electric cast iron exhaust and forced draft fans. Also volume control dampers.



Type CE

Chelsea Products, Inc.

1206 Grove St., **Irvington 11, N. J.**

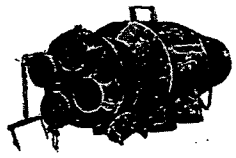
Charter Members Propeller Fan Manufacturers Association

FANS

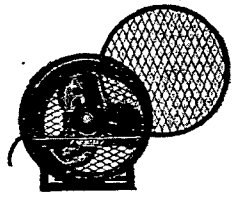
Trade Mark Registered

EXPORTED TO ALL COUNTRIES OF THE WORLD BY

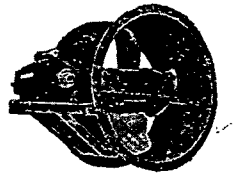
Western Electric Export Corporation



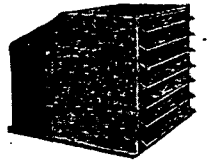
TYPE OPJ—OCTOPUS JR.
Portable Exhauster and Blower



TYPE DUB—SHIPHOLD EXHAUST
Blower or Exhauster



TYPE DXB—BOOSTER
For Spray Booths, Fumes, Excessive Heat



TYPE PH—PENTHOUSE
Roof Ventilator

FEATURES

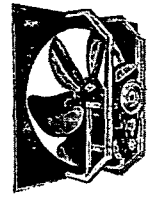
(On Comfort Coolers)

1. **New Streamline Venturi Orifice**—greater volume.
2. **Certified Ratings**—as per the A.S.H.V.E. method.
3. **Efficient Fan Duty Motors**—low current consumption.
4. **Die Stamped Blades**—precision balance and Alignment.
5. **Moving Parts Rubber Cushioned**—quietness.
6. **Ball Bearing on Fan Shaft**—long life and little attention.
7. **Faultless All Steel Construction**—stability.

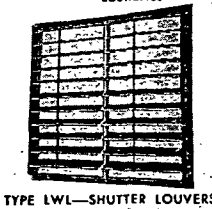
Chelsea Fans are Known All Over the World

TYPE CP—PRESSURE BLOWER
General Purpose Pressure Type

TYPE AA—UTILITY
For All Purposes; Answers Any Fan Requirement



TYPE IND—INDUSTRIAL
For Factories, Foundries, Laundries



TYPE LWL—SHUTTER LOUVERS
Automatic, Regular and Ceiling Models

Bulletins and Engineering Data furnished on request. Personnel includes many skilled operators—employees who have been identified with the air conditioning industry for over 30 years.

DeBothezat Fans Division

American Machine and Metals, Inc.

***Main Office and Factory*—East Moline, Illinois**

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Kansas City 6, Mo.	Midland Bldg.	Washington 5, D.C.	1333 "G" St., N.W.

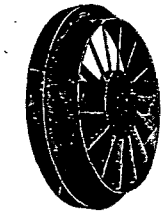
SALES ENGINEERING OFFICES IN ALL PRINCIPAL CITIES

Non-Overloading Power Characteristics

AXIAL FLOW PRESSURE FANS

Certified Ratings • Guaranteed Performance

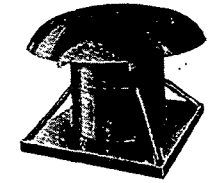
HIGH STATIC PRESSURES



Axial-Flow Fan

DeBothezat Volume and Pressure Type Axial-Flow Disc Fans have been developed in separate series to cover the extensive range of static pressures and capacities. To meet varying conditions of pressure, load and speed with equally high efficiency over the entire range, pitch angles and motor speeds are engineered to fit each particular application. Available in sizes from 16 in. through 144 in. (Complete "Ventilating Sets" and "Giant Fans" Catalogs on request).

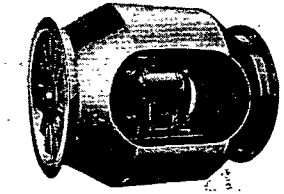
CONTROLLED VENTILATION



Power-Flow Roof Ventilator

DeBothezat Power-Flow Roof Ventilators of guaranteed performance ratings, dynamically correct in design, are rigidly constructed. Streamlined and of low height, they present a trim appearance. Positive controlled ventilation is assured at all times independent of variable weather factors or wind velocities. The hinged hood provides ready access to motor and fan. Available in 16 in., 21 in., 27 in., and 36 in. sizes. (Complete "Roof Ventilators" Catalog on request).

CORROSIVE FUMES REMOVAL



Bifurcator

DeBothezat Bifurcator is an especially designed housing ingeniously arranged and proportioned in combination with a direct driven axial-flow pressure fan of non-overloading characteristics. The air handled by the fan is by-passed around the drive motor which is placed entirely outside of the air duct in a through ventilated chamber. Available in sizes 16 in. through 48 in. (Complete "Bifurcators" Catalog on request).

SPOT COOLING

"Hy-V" Air Jet

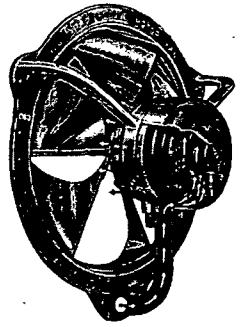
DeBothezat "Hy-V" Air Jets are extraordinarily efficient man coolers, particularly for workmen who are exposed to radiant heat from ovens, furnaces, forges and molten metals. Arranged for either column or floor mounting, they are adjustable to blow a concentrated blast of cooling air in any direction to remote spaces without the aid of ducts. Available in 18 in. through 30 in. sizes. (Complete "Hy-V Air Jets" Catalog on request).

ILG Electric Ventilating Co.

2832 North Crawford Ave., **Chicago, Ill.**

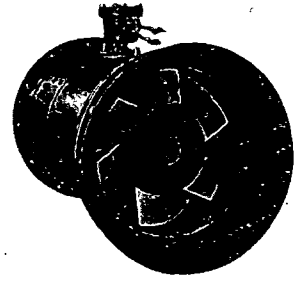
Offices in 40 Principal Cities

Propeller Fans, Vaneaxial Fans, Centrifugal Fans, Unit Heaters, Unit Coolers



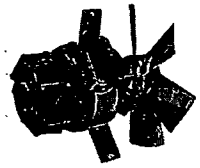
ILG Self-Cooled Motor Propeller Fans

Used for exhaust of stale air, fumes, heat, dust, odors, etc. Self-cooled motor combines protection of enclosed motor with low operating cost of open motor—constantly cooled by fresh, clean air, circulated internally—never "gums-up" from contact with foul air—saves 5 to 10 per cent on power costs. Rugged, heavy-duty framework. Dynamically balanced fan wheel, direct-connected to motor. Smooth, quiet, effortless operation—low cost, long lived. "One-Name-Plate" Guarantee. Certified ratings. Sizes, 8 in. to 72 in. Get Catalog No. 145.



ILG Vaneaxial Fans

Heavy-duty, extremely compact design, as developed by ILG for use aboard ship by the U. S. Navy and Maritime Commission. Suitable for duct and trunk mounting for vertical and horizontal operation. All steel housing flanged at both ends. Removable streamlined inlet cone. Guide vanes straighten air flow, reduce turbulence. In ILG design, vital working parts are quickly, easily accessible for installation, lubrication, or servicing. Proven efficiency and dependability. "One-Name-Plate" Guarantee 10 in. to 46 in. sizes. Get new data sheets.

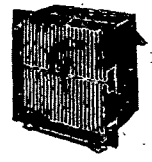


"UN" TYPE FANS

For vertical or horizontal mounting anywhere except in ducts. Several models with various mounting arrangements. Get data sheet.

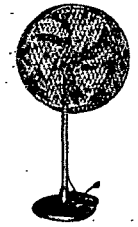
"CQ" TYPE FANS

For mounting outside of ducts. Wide range of sizes and models for various mounting arrangements. Get data sheet.



KITCHEN VENTILATORS

Built-in becomes permanent part of kitchen wall. Quiet, high capacity, efficient, equipped with ILG Self-Cooled Motor. 3 sizes. Also models for window installation. Get Catalog No. 142.

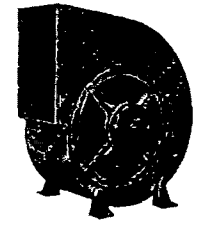


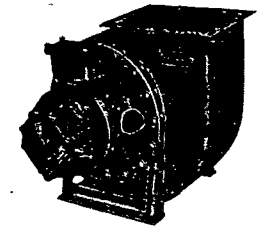
NIGHT COOLING FANS

For permanent installation in attic, use ILG Self-Cooled Motor Propeller Fans (top of page). Portable models for use at downstairs windows.

FOR ILG UNIT HEATERS AND COOLERS, SEE PAGE 982

DIRECT-CONNECTED CENTRIFUGAL FANS

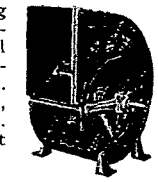




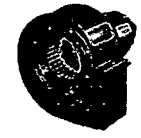
"BC" Type—Load-limiting type with backward curved blades. Motor load remains constant over wide range of air volume and change in static pressure. Wheel mounted directly on motor shaft with motor partially recessed in side of casing. No motor base required. Unobstructed inlet. Also available for belt-drive. 11 sizes. Get Catalog No. 245.

"CC" Type—All steel, heavy gauge, hot dipped galvanized, welded housing—no angle iron braces. Flanged inlet and outlet connections. Non-overloading type wheel mounted directly on motor shaft, with motor partially recessed in housing to save space—no motor sub-base required. Unobstructed inlet. 11 sizes. Get Data Sheets.

"B" Type—Motor recessed in side of casing, eliminating separate base. Multiblade, forward-curved wheel direct-connected to motor shaft. Also available for belt drive. 12 sizes. Get Catalog No. 245.



"BW" Type—Housing similar to ILG Direct-Connected Centrifugal Fans. Wheel with forward curved blades. Full universal discharge, right or left hand drive. Wide range of sizes. Get Catalog No. 245.



Type "B" Volume Blowers—Small volume, low pressure, quiet running. Multiblade wheel direct-connected to motor shaft. Cast-iron base. Universal discharge. 12 capacities. Get Catalog No. 145.

Type "P" Volume Blowers—For exhausting dust, fumes, removal of steam, vapors. Four discharge positions to avoid friction in short bends. 7 capacities. Get Catalog No. 145.

Type "6S" Utility Blowers—Particularly suitable for building into apparatus which requires ventilation or air movement. Extremely flexible in arrangement, furnished with or without inlet flange, outlet flange, stand, etc., providing a practically custom-built unit on basis of quantity manufacture. Wide pressure range. Low power input. Get specifications, including dimensions and performance graph.

LA-DEL CONVEYOR & MFG. COMPANY

ENGINEERS OF LA-DEL AIR MOVEMENT

NEW PHILADELPHIA, OHIO

MANUFACTURER OF AXIAL FLOW FANS

AXIAL FLOW FANS FOR EVERY VENTILATION PROBLEM

Designed to meet the specific requirements of every type of ventilation—industrial, commercial, aviation marine or domestic—La-Del Axial Flow Fans provide quiet, efficient operation, free from excessive maintenance and noise. La-Del fans are available in a wide range of types and sizes to meet every need; each installation is custom-engineered to provide new standards of quiet, efficient, low-cost and trouble-free operation.

La-Del has the experience and technical knowledge necessary to provide the correct answer to every ventilation problem. La-Del experienced ventilation engineers study each application; La-Del's modern production planning and manufacturing methods insure the utmost in space-saving ventilating fans, designed to operate at low power cost.

One of the foremost suppliers of our Armed Forces during the war years, La-Del's expanded production facilities allow the manufacture of Axial Flow Fans in a liberal capacity range, permitting the selection of the fan that will provide the ventilation you require in the most efficient manner.

What Is True Axial Flow?

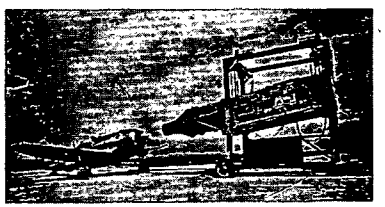
The Axial Flow Fan consists of one or several rotating wheels, carrying twisted, airfoil-shaped blades on a round hub and counter-rotating devices that suppress the twist in the air caused by the rotating wheel. Its aerodynamic design eliminates all turbulence behind the fan propeller, permitting the building up of high pressure and the consequent conveyance of air straight through the duct.

What Does This Mean To You?

Simply that a La-Del Axial Flow Fan will provide ventilation, where it is needed, in an economical manner. The flow of air you require to eliminate smoke, noxious odors or other atmospheric conditions, goes directly to the point where it will be immediately effective.

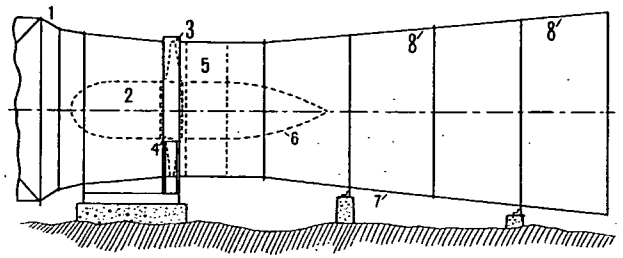
Here is a truly economical and efficient means of ventilation—air, passing directly through the duct, is moved at low cost, and it provides efficient ventilation, without excessive power consumption. Further, La-Del can also furnish *exclusive* adjustable pitch blades which broaden the useful operating range of the fan by fully 75 per cent. Such fans have already been tested and fully proved on the most difficult ventilating applications.

Special Fan For *N.A.C.A.*



La-Del ability to handle the unusual and difficult jobs is evidenced by this huge blower furnished the N.A.C.A. Aircraft Engine Research Laboratory. It is a life saver to test pilots and capable of producing an air speed of 250 miles per hour. This La-Del fan permits test engines to be operated at maximum take off power and eliminates the danger of take-offs with untried engines. Its output is 60,000 cfm at 45 in. P.S; weighs several tons and yet portable as shown. Two stage, adjustable pitch, equipped with 500 hp motor and auxiliary blowers that ventilate the main motor through hollow vanes.

Check These Outstanding LA-DEL Features



1—Flared intake and streamlined nosing assure uniform air flow distribution at propeller blades.

2—Streamlined belt fairings fully enclose drive belts, prevent leakage, protect belts from air stream and avoid turbulence ahead of blades.

3—Adjustable pitch propeller blades for peak efficiency with balanced characteristics throughout adjustment and fan operating range.

4—Straightening blades scientifically designed to correct helical motion imparted by propeller blades to a true axial flow.

5—Propeller blade from tip to hub is a fully effective working surface, with blades terminating inside the hubs.

6—Streamlined tail maintains uniform axial flow without turbulence.

7—Fan efficiency is directly affected by stack efficiency. Carefully calculated stack design results in uniform air flow and low exit velocity.

8—Fan and stack diameters (at small end) correspond to keep frictional losses at lowest point.

Look to LA-DEL for (EFFICIENT / ECONOMICAL) *VENTILATION*

From the smallest to the largest, La-Del is prepared to furnish an axial flow fan to meet every requirement. At the left, an aviation type fan, only 6 inches long and 6½ inches in diameter, delivering 400 cfm at 6 in. pressure. At the right, a La-Del Fan installation with exhaust stack and hood. This fan, 11 ft in diameter, delivers 500,000 cfm at 6½ in. static pressure.

For space-saving efficiency, reduced operating cost, unexcelled performance, quiet operation and freedom from maintenance troubles—remember La-Del when planning ventilation. La-Del Ventilation Engineers will gladly show you how and why—send your inquiries, outlining your requirements, direct to The La-Del Fan Company, Division of La-Del Conveyor & Manufacturing Co., New Philadelphia, Ohio.

Hartzell Propeller Fan Co.

DIV. OF CASTLE HILLS CORP.

Piqua, Ohio

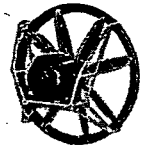
Sales-Engineering Offices in Principal Cities

Manufacturers of Propeller-Type Fans and Blowers For Every Industrial Use

SINGLE PROPELLER FANS

Sizes, 12 to 144 in.; free air deliveries 590 to 270,000 cfm. NEW curved orifice overlaps propeller tips, forming Hartzell Airseal (patented), giving extra air delivery, minimum turbulence. Send for *Bulletin 1101.*

MULTIBLADE FANS



Sizes, 12 to 144 in.; free deliveries 1,200 to 280,000 cfm. (Up to 200,000 cfm against 1.00 in. static pressure) Long life, low upkeep. NEW curved orifice and patented Hartzell Airseal. Send for *Bulletin 1101.*

BELT-DRIVE FANS

Sizes, 12 to 144 in.; air deliveries 590 to 280,000 cfm. Available in one-propeller, two-propeller and multiblade styles. For use in fire-hazardous locations such as paint spray booths. Grease-sealed, dust free ball bearings. Send for *Bulletin 1401.*

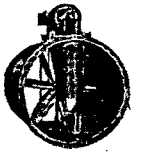
EXTENSION SHAFT FANS

Sizes, 12 to 55 in.; types, one-propeller, two-propeller and multiblade. Designed for use in ducts through which corrosive elements, excessive heat or abrasive dust passes. Send for *Bulletin 1401.*

TWO-PROPELLER FANS

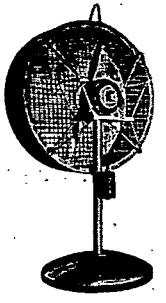
Sizes, 16 to 144 in.; free air deliveries 2,400 to 275,000 cfm. (Up to 200,000 cfm against 0.34 in. static pressure) NEW curved orifice and patented Hartzell Airseal mean maximum air. Send for *Bulletin 1101.*

BLOWERS



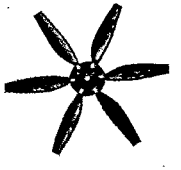
Sizes, 18 to 48 in.; free air deliveries 3,300 to 50,500 cfm. (Up to 34,000 cfm against 8 in. pressure). Built in both belt drive (shown) and direct drive models. Easy installation. Send for *Bulletin 1601.*

COOL BLAST FANS



Models: Portable, Stationary and Heavy Duty Cool Blast Fans (stationary shown), sizes 24 to 55 in., for large-volume air movement; Industrial Stand Model, up to 36 in. size, with substantial but light-weight high stand; Utility, up to 36 in. size, extremely versatile, with simple, close-to-the-ground mounting. Send for *Bulletin 1201.*

COOLING TOWER FANS



Sizes, 3 to 16 ft; four and six blades, adjustable pitch. 10 to 16 ft sizes are special Hartzite plastic, steel hub. Smaller sizes are aluminum alloy. Send for *Bulletin 1501.*

SPECIAL EQUIPMENT

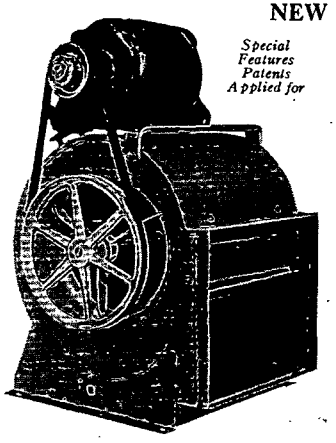
There is a Hartzell fan or blower to meet your needs or we will design and build one for you. Write. All Hartzell air deliveries are in accordance with the Standard Test Code for Centrifugal and Axial Fans.

The Lau Blower Company

2007 Home Avenue, **Dayton 7, Ohio**

Engineers and fabricators of general Air Handling Equipment
Single Inlet and Double Inlet Blowers • Propeller Fans • Accessories

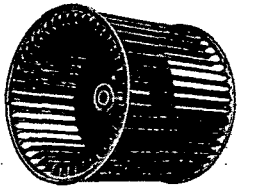
NEW Series "A" Blower Assemblies



Special Features Patents Applied for

The Lau Series "A" Blower Assembly—result of years of exhaustive tests of all types of blowers—years of research and design evolution—is the all-time, outstanding achievement in the blower field. Greater mechanical strength. Greater efficiency. A more compact unit (overall size considerably smaller than formerly). Will fit more jobs. Embodies many new and revolutionary features exclusive with Lau. Includes new 3-point suspension type bearing bracket—an integral part of the shroud—identical for various angles of discharge. New frictionless, self-aligning bearing—completely encased in Neoprene. New center suspension wheel (see below). New discharge outlet design and construction. Cut-off cannot set crooked on outlet. No wavy edges. Faster installations. New 1-piece motor mounting easily convertible, rear to top or vice versa, by simple use of two sheet metal screws. Many other features. Complete range of sizes. Every size tested and rated for performance in accordance with A.S.H.V.E. and *N.A.F.M.* Codes.

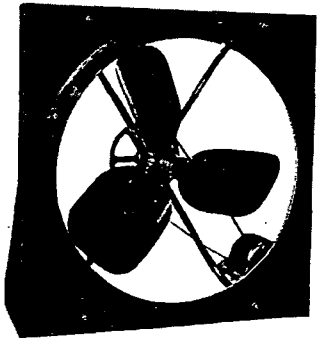
NEW Series "A" Blower Wheels

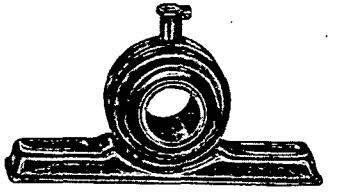


New center-suspension wheel *tested and proved* by us to have greater mechanical strength, truer concentricity, and far more efficient performance than ordinary types of wheel. Complete details supplied to interested parties upon request.

Propeller-type "Niteair" Fans

For a wide variety of applications where it is necessary or advantageous to exhaust undesirable air and provide fresh air from the outside. Equally applicable for industrial, commercial, residential and farm building installations. Efficient and economical method for correcting innumerable air-control problems—removing dust-laden, foul, contaminated, or excessively hot air, fumes, gases, smoke. Circulating cool night air through living and sleeping rooms for greater summer comfort. Venturi-type entrance housing reduces air "drag" and turbulence—eliminates most common cause of "air noise." 5 sizes—22 in. to 48 in.





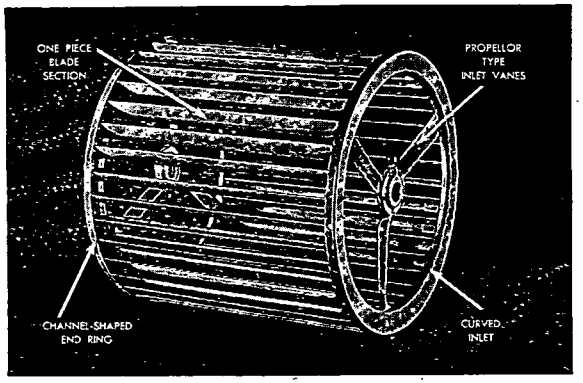
Self-aligning Pillow Blocks

Catalogs, performance data, specifications, and prices available on request on above and other air handling equipment. Inquiries solicited for any application. Our engineers will gladly assist you. *Write us regarding your requirements.*

MORRISON PRODUCTS, INC.

East 168th St.
& Waterloo Rd.

Cleveland 10,
Ohio



Blower Wheels

BLOWER WHEELS *for* MANUFACTURERS OF WARM AIR HEATING AND AIR CONDITIONING

Every manufacturer whose products employ blowers will welcome the MORRISON AIRSTREAM Blower Wheel—an outstanding advancement in the art of fan design and manufacture. The MORRISON AIRSTREAM Blower Wheel is a definite stride forward in aerodynamic design.

• ONE-PIECE BLADE CONSTRUCTION

All blades in every Blower Wheel are made from one strip of soft steel on progressive die equipment, eliminating any possibility of loose blades.

• THREE-PIECE BALANCED ASSEMBLY

Two pressed end rings with integral hubs and a one-piece blade assembly constitute the complete blower wheel. When spotwelded together, these three pieces form a strong, rigid, true running blower wheel.

• EQUALIZED WEIGHT DISTRIBUTION

End-mounted Blower Wheels distribute their weight closer to the bearings. Smaller shafts and bearings are therefore permissible. Shaft whip is eliminated and shaft deflection reduced.

• LIGHT IN WEIGHT

MORRISON AIRSTREAM'S unusual design reduces weight to a minimum. Integral pressed hubs eliminate heavy cast or screw machine parts. Lightness affects motor loads—makes for quieter operation and reduces starting torque.

• **MORRISON AIRSTREAM BLOWER WHEELS are furnished in standard diameters** of 10⅝ in., 12 in., 14½ in. and 16 in. and in various widths. Catalogue showing equipment required for manufacturing blower housings and drawings showing suggested method of manufacturing complete blowers furnished upon request together with performance data on various sizes.

The NEW YORK BLOWER COMPANY

GENERAL OFFICES
3145 SOUTH SHIELDS AVENUE • CHICAGO 16
FACTORIES at LA PORTE, IND. and CHICAGO, ILL.

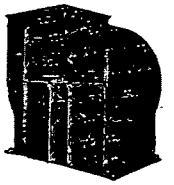
Representatives in Principal Cities

FANS • BLOWERS • UNIT HEATERS • MAKE-UP AIR UNITS
AIR WASHERS • HEAVY DUTY HEAT SURFACE

M15—15,000 cfm

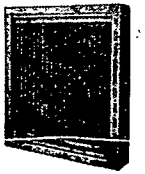
A unit that delivers, warmed, filtered, outside air to industrial spaces to replace exhausted air and balance minus pressure. Corrects drafty conditions and uncontrolled infiltration. Made in 4 sizes from 5,000 cfm to 20,000 cfm. Described in *Bulletin 453.*

Type ME Centrifugal Fans



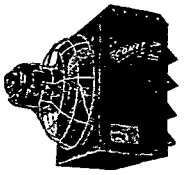
Quiet operating slow speed wheels for heating, ventilating and air conditioning systems, or high speed wheels with non-overloading horsepower characteristics for industrial applications. Wheel diameters from 15 in. to 80 in., with any speed or discharge required. Class I, II or III construction. Capacities up to 108,000 cfm.

Steelfin Hot Blast Heating Surface



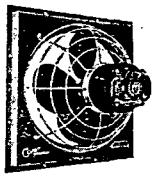
Extra heavy duty, fin-and-oval tube, all-steel, welded construction. A hot dip metallic coating over all, including headers, affords perfect bonding and conductivity. Suitable for continuous heating service on steam pressures up to 150 lb.

Comet Unit Heaters



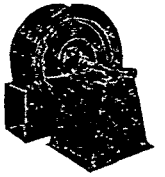
Heavy duty, welded steel, fin-and-tube heating element. Suitable for continuous heating service on steam pressures up to 150 lb or more. 10 sizes with capacities from 31 Mbh to from 31 Mbh to 300 Mbh. *Bulletin 454.*

Comet Exhaustair



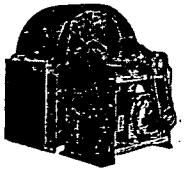
Delivers large volumes of air at low resistance and low current consumption. All wheels are machine balanced for smooth, vibrationless operation. Made in three types and ten basic sizes. Wheel diameters from 12 in. to 60 in. Direct or belted drive. Capacities from 400 cfm to 33,500 cfm. *Ask for Bulletin 442.*

Type SSP Industrial and Heat Fans



For dust and gas removal, conveying of materials and handling hot gases. Housings, drives, and discharge arrangements to meet any requirement. Wheel diameters from 14 in. to 66 in. Capacities from 450 cfm to 60,500 cfm.
Details and engineering data in Bulletin 432.

General Purpose (Jeep) Fans

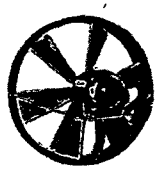


Portable, self-contained units for Class I industrial and ventilating applications. Recommended for ease of installation, low maintenance and space saving features. Made in three types and eight basic sizes. Capacities 400 cfm to 12,000 cfm. *Bulletin 443.*

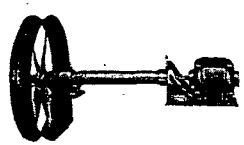
PROPELLAIR Inc.

1944 CLARK BLVD.
SPRINGFIELD, OHIO

VENTILATING SPECIALISTS IN ALL PRINCIPAL CITIES



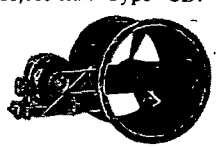
For Ducts, Walls, Windows, Hoods, Roof Ventilators



PROPELLAIR DIRECT-CONNECTED FANS

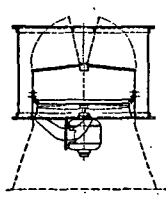
For use wherever motors may operate within the air stream, from free air to medium and relatively high resistance. A compact, durable design with fan having two to six blades. Sizes: 12 in. to 60 in. Capacities: 800-85,000 cfm. Type "CD."

For Heat, Acids, Alkalies—Fumes, Gases, Dust



BELT-DRIVEN PROPELLAIR DRUM-TYPE

A complete fan unit in short duct section ready for installation in lines from 20 in. to 48 in. diameter. Type "CS" may be used for severe acid or alkaline conditions, explosive fumes and gases. Type "CSV," for excessive temperatures, circulates outside air through belt and fan shaft tubes to keep drive and bearings cool. Also furnished without duct section. Capacities: 4100 to 43,000 cfm.



For Roof Ventilation

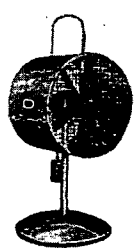
PROPELLAIR VERTI-STACK

Dependable and economical power roof ventilators. Butterfly dampers open wide the instant fan is started, close automatically as fan coasts to a stop, offer virtually zero resistance as heat, fumes, moisture, dust shoot high into air. Rain is prevented from entering by fan when operating. Drainage gutter prevents leakage when dampers are closed. Sizes: 20 in. to 60 in. Capacities: 3700 to 77,000 cfm.

For Heat, Moisture, Fumes, Dust, and Gases

PROPELLAIR EXTENDED-SHAFT FANS

This design locates motor outside air stream when fan is installed in duct at right angle turn, elbow, "Y," or offset. Simple installation usually can be supported by duct without auxiliary bracing. Drive shaft is enclosed and sealed within steel tube. Sizes: 12 in. to 60 in. Capacities: 2000 to 68,000 cfm. Type "CE."



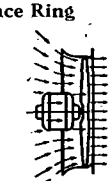
For Cooling Men and Materials

PROPELLAIR PEDESTAL & CRADLE-MOUNTED FANS

Pedestal-type Man Coolers have sturdy steel drum protected front and rear, formed steel base, rigid support, and crane hook. Also made with adjustable cradle mounting (Type "CU"). Capacities to 20,650 cfm.

AIRFOIL-SECTION Blades

Airfoil Principle Entrance Ring



Propellair fans have airfoil-section blades with variations of pitch, curvature, and thickness to compensate for different lineal speeds of points at various radii. Air movement is uniform over whole fan area. The Propellair curved entrance ring eliminates eddy currents; helps efficient Propellair blades deliver highest pressure and volume.

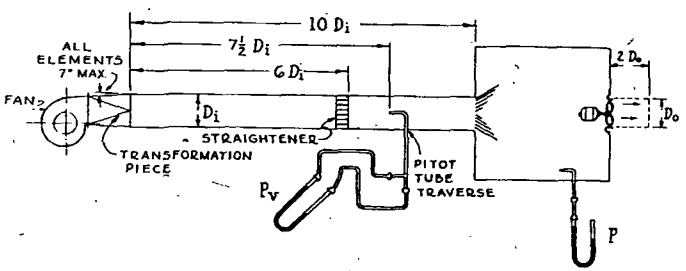
PFMA

Propeller Fan Manufacturers Association

2-255 General Motors Bldg., **Detroit 2, Mich.**

When you see the **PFMA** Certified Rating Insignia, you know that the company displaying it certifies that the performance of its products is in strict accordance with the Standard Test Code.

Air tunnel design for Standard Test Code measurements is illustrated below, the propeller fan under test being mounted at the discharge end.



Exhausters without Duct

Supply Duct
Chamber
Fan Outlet

Supply Duct: Velocity pressure, p_v; Differential reading between impact tube and static tube. **Chamber:** p by impact tube. **Fan Outlet:** P_v; Correct p_v to P_v, the velocity pressure D_o corresponding to the area A of the minimum ring diameter. P_t; add p to P_v. P_s; Subtract P_v from P_t. Correct all pressures to Standard Air Conditions. All additions and subtractions mentioned above are arithmetical.

Testing equipment and methods are prescribed in detail by the Standard Test Code, from which the accompanying air tunnel dimension diagram is reproduced to indicate the exactitude which is required in PFMA testing.

Copies of the complete Standard Test Code are available for 25 cents through the Association.

The **PFMA** insignia shown at the right, is your assurance of a dependable basis to judge fan performance. Guesswork is eliminated and data listed are reliable and based on sound engineering technique, permitting you to evaluate comparative performances.

Where fans are rated by Standard Test Code procedure, volume, pressure, power input, mechanical efficiency and all related factors, are measured under precisely controlled conditions.

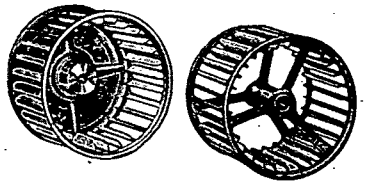
Look for the PFMA insignia
Insist upon Certified Fan Ratings

The Torrington Mfg. Co.

50 Franklin Street, **Torrington, Conn.**

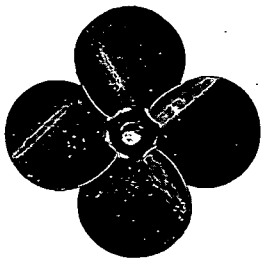
Manufacturers of Blower Wheels and Propellor Type Fan Blades.

AIRISTOCRAT Quiet Propeller Fan Blades AIROTOR Blower Wheels

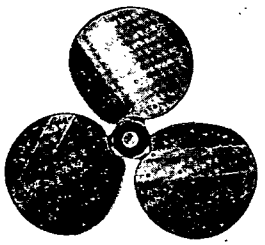


Left—Airotor Blower Wheel—Single Width—Single Inlet Patents 2,251,062; 2,251,063; Des. 126,045; 2,272,695

Right—Airotor Blower Wheel—Double Width—Double Inlet—Spider End Plates



Airistocrat "Standard" Series Pats. 2,072,322 and 2,021,707



3-Blade Airistocrat "Y" Series

Torrington Airotor Blower Wheels are light, sturdy and inexpensive—incorporate new principles of design and construction, which insure rigidity and concentricity. **Single Width—Single Inlet** wheel is of simple four-piece construction. **No rivets or welds are used;** concentric rib serving as backing for blade strip is formed at same time as hub socket, insuring trueness of wheel. Rigid radial ribs prevent deflection by thrust. Three thicknesses of metal in rims make for maximum strength. Excellent for many heating and ventilating uses. Manufactured in both aluminum and steel in 1½ in., 2, 3, 3⅝, 4½, 5, 6, 7½, 9 and 10½ in. diameters. Clockwise or counterclockwise rotation. Same sizes available in DA type double width, double inlet wheels.

Torrington Airotor Blower Wheel—Double Width—Double Inlet—Spider End Plates. Has blades punched and formed in a single strip, rigidly held by flanged single piece end rings. Hubs are rigidly mounted by peening. Wheels of 3⅝ in., and 10½ in. diameter are available at present. Additional sizes now being developed.

AIRISTOCRAT Quiet Propeller Fan Blades are widely recognized for their all-around excellent performance. The unique, patented construction embodies entirely new principles in the art of fan design—produces a blade unsurpassed for quiet operation, rugged construction and attractive appearance. Every **Airistocrat** unit is carefully built and the blades are hand gauged for correct contour and alignment. Statically balanced, these blades deliver full air volume with a minimum of noise. Aluminum alloy blades and steel spiders are standard except where otherwise noted.

Available in a variety of finishes. Catalog gives detailed dimensions and guaranteed performance curves recorded under **NEMA** and **NAFM** code tests at various speeds for each of the **Airistocrat** models described on page 1071.

New Blower Wheel Catalog charts capacities, gives detailed dimensions, suggests housing dimensions for all above wheels. Valuable "Tips" for designing air impelling units are also included.

"Standard" Series—Has blades mounted on a steel spider. Sturdy, attractive steel or aluminum blades which have withstood extreme laboratory breakdown tests. Sizes 8 in., 10, 12, 14, 16, 18 and 20 in. diameters in a variety of pitches to meet every need.

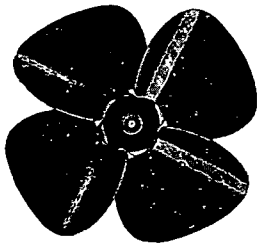
Three Blade "Y" Series—The design of this blade is the result of two years of laboratory experiment to produce a better air circulator blade. At recommended speed these blades produce a high velocity air stream effecting deep penetration with unusual quietness. Sizes 10 in., 12, 14, 16, 18, 20, 24 and 30 in. diameters, steel or aluminum blades.

Pressure "P" Series—Similar in construction to "Standard" Series but with blades especially designed for higher pressures. Sizes 10 in., 12, 14, 16 and 18 in. diameters.

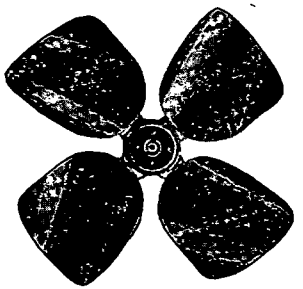
Pressure "U" Series—Four blade models of steel designed for pressure operation. Sizes 20 in., 22, 24, 26, 28 and 30 in. diameters.

AIRISTOCRAT "B" Series Attic Fan Blades represent a complete redesign of the popular "A" Series. Efficiency has been still further improved. The same proportions, proved aerodynamically correct, have been carefully adhered to in all diameters. New larger center disc and heavier spider arms increase strength and a new blade shape adds to the appearance of this carefully designed product. Flexibility of parts allows assembly with either 3, 4 or 5 blades in standard diameters 24, 30, 36, 42 and 48 in. All steel construction. Available in the following finishes: 1. Plain. 2. Colored lacquered blades, black lacquered spider and hub. 3. All one color lacquer.

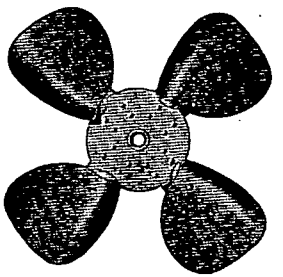
"One-Piece" Series Propellor Fan Blades—Exceptionally rigid models blanked from one piece of metal. Made in both steel and aluminum. Sizes 3 in., 4, 4½, 5, 5½, 6, 6½, 8, 9, 10, 12 and 16 in. diameters, all four blades; also 5½, 7, 8, 9, 10 in. 5-blade. Available in the following finishes: 1. Plain. 2. Lacquered. 3. Nickel or cadmium plated (steel only).



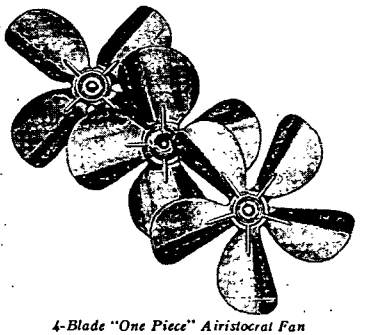
4-Blade Airistocrat Pressure Fan "P" Series



4-Blade Airistocrat Pressure Fan "U" Series



4-Blade Airistocrat Attic Fan "B" Series (also made in 3 and 5 blade models)



4-Blade "One Piece" Airistocrat Fan

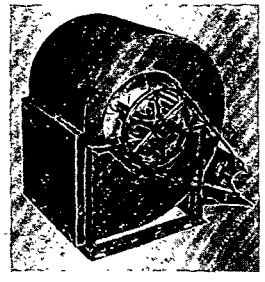
Schwitzer-Cummins Company

1125 Massachusetts Avenue

Indianapolis 7, U.S.A.

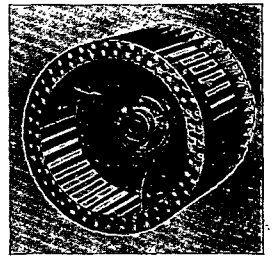
Engineers and Manufacturers of Centrifugal Blowers and Blower Wheels—Attic Fans—Ventilating Fans of All Types

PRODUCTS OF THE VENTILATING DIVISION



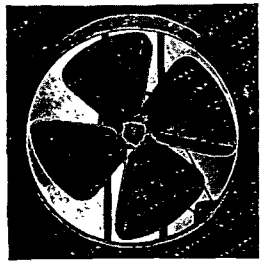
HY-DUTY BLOWERS SINGLE INLET AND DOUBLE INLET

Ruggedly built and especially designed for fine performance and quiet operation. Four different outlet positions for double inlet models, eight positions for single inlet. Wheel diameters 9¾ in. to 25 in. Air deliveries 200 cfm to 15,000 cfm. *N.A.F.M.* code tested.



HY-DUTY BLOWER WHEELS

We are tooled for a large range of sizes, single inlet and double inlet, diameters 4½ in. to 50 in. in many widths. Can furnish our standard wheels or build to your specifications. HY-DUTY wheels in Schwitzer-Cummins blowers are the center disc, wide blade type with heavily reinforced driving discs and are accurately balanced.

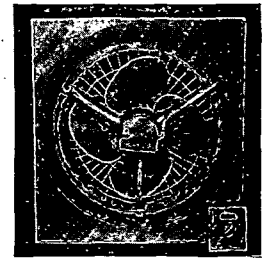


FRESH-AIR MAKER HOME AND INDUSTRIAL VENTILATORS

A rugged, all welded, light weight, low cost, high delivery, ball bearing ventilator with unusual flexibility for installations in homes, apartments, factories, business establishments.

FIVE FAN DIAMETERS

24 in., 30 in., 36 in., 42 in., and 48 in. 5000 cfm to 20,500 cfm.



A beautiful **Fresh-Air Maker** window ventilator designed to harmonize with the most luxurious apartment furnishings. Unique features for quick installation without alterations to surroundings so tenant may possess his own ventilator.

Catalogues and Engineering Information on Request

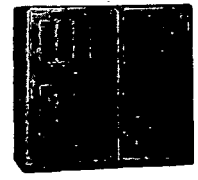
Utility Appliance Corporation

FORMERLY UTILITY FAN CORPORATION

4851 S. Alameda St., Los Angeles 11, Calif.

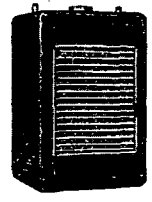
Utility Gas-Fired Heating Equipment, Evaporative Coolers, Blowers and Fans

FORCED AIR FURNACES



Basement and closet types . . . Compact design . . . Multiple-fin element with air-cooled hollow baffle . . . Element guaranteed against burnout forever . . . Automatic controls . . . Filters . . . Built-in motor overload protection.

UNIT HEATERS

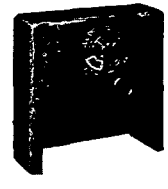


Compact heat exchanger with no inside baffles—Individual burners for each element section—Silent, disc-type fan—All-welded cabinet—Directional grilles—Built-in draft diverter—Temperature limit control. Four sizes.

FLOOR FURNACES

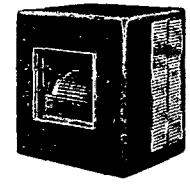
All-welded construction . . . Die-stamped grilles . . . heavy cast-iron burner . . . removable inside jacket . . . interlocking gas valve. Floor or dual registers. 25,000, 37,000 and 50,000 Btu input.

CIRCULATING HEATERS



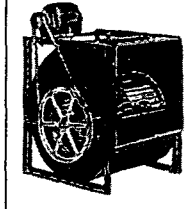
Fan sends stream of air through nozzle-shaped outlet to hold warm air in living zone. Built of heavy furniture steel . . . all die-formed and electric welded. Vented and unvented models.

EVAPORATIVE COOLERS



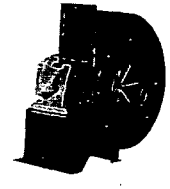
Comfort cooling—residential, commercial, industrial. Exclusive feature—Uni-flowmeter (Pat. Pend.) for uniform water distribution. Patented No-Sag cooling pads. 14 Models.

STANDARD BLOWERS



Dynamically balanced, multiple-vane centrifugal blowers. Four-side angle iron frame increases rigidity, eliminates vibration—permits installation with any of four discharge positions.

EXHAUSTERS



High flow and high pressure designs in four drive arrangements, for exhausting many materials. High efficiency . . . decreased sound. Available with acid-proof plastic surfacing.

CLOVERLEAF TYPE FANS

Propeller Fans are available with Cloverleaf (over-lapping) or Streamline type blades, with direct or belt drive for mounting into windows, openings, or on steel or wood panels.

Utility Blowers are tested in accordance with the A.S.H.V.E. Code.

Write for complete information, catalogs and prices.

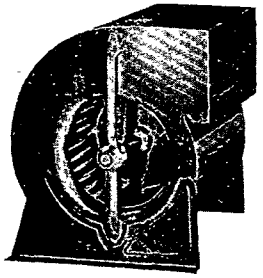
The Viking Air Conditioning Corporation

5600 Walworth Avenue, **Cleveland, Ohio**

Viking products, which include blowers, fans, and humidifiers, offer top economy and efficiency to furnace and air conditioning manufacturers because Viking

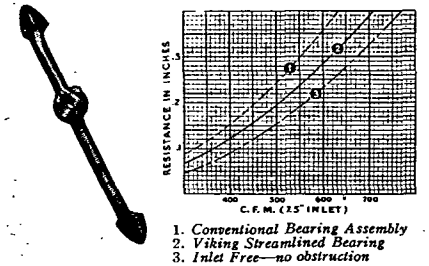
specializes **in equipment for air conditioning. The economies of mass production due to specialization are thus passed on to all Viking customers.**

BLOWERS AND BLOWER PARTS FOR *Air Conditioning Manufacturers*



COMPLETE ASSEMBLY

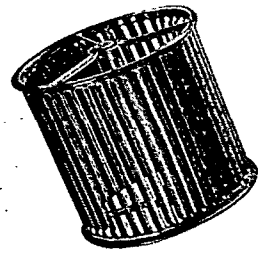
Viking's blower assembly embodies several entirely new features, among which are the streamlined bearings, the new-type blower wheel, the lower outlet, and the smaller unit size overall. Space savings alone in 10 in. blowers run 2½ in. in height and 1½ in. from front to back. Extremely specialized equipment plus high production of 9 in., 10 in., 12 in., 14 in., and 16 in. blowers result in completely assembled blowers at practically the cost of parts only.



1. *Conventional Bearing Assembly*
2. *Viking Streamlined Bearing*
3. *Inlet Free—no obstruction*

STREAMLINED BEARINGS

Important innovation in bearing design is this unit Viking Streamlined Bearing. Note the graph at left for comparative record of Free Areas. You see that whereas a typical bearing allowed 77 per cent Free Area for air intake, the Viking Streamlined Bearing permits an 85 per cent Free Area, resulting in improved overall performance of the entire blower assembly.



BLOWER WHEEL

Viking blower wheels are produced by a continuous stamping process and consist basically of only 6 parts—2 blade sections, 2 spoke ends and 2 hubs. Larger inlets help provide increased air delivery. The wheels are designed for the higher pressures of modern air conditioning requirements. Costs are lower than for any pre-war wheel.

Send For More Information

Though parts alone are available if desired, the majority of standard requirements can be met most profitably with a Viking assembly. As specialists in making air conditioning blowers, we invite your inquiries. Write for our Special Manufacturers Brochure, which explains Viking facilities and products. Address Desk A.

Established in 1891

Wagner Electric Corporation

Sales Offices in 29 Principal Cities

6464 Plymouth Avenue, **Saint Louis 14, Mo., U. S. A.**

Wagner Motors Give Maximum Efficiency on Heating, Ventilating, and Air Conditioning Equipment

Wagner motors embody the latest developments in design. They are simple, rugged, and dependable and have long life in addition to good electrical performance.

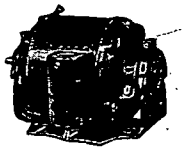
Split-Phase Motors

All standard frequencies and voltages; sleeve and ball bearings; open, dripproof, and totally-enclosed; rigid, resilient and flange mounted. (1/20 to 1/3 hp).

Permanent Split-Capacitor Motors

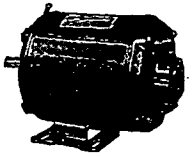
Available as constant-speed, two-speed, or adjustable-speed. All standard frequencies and voltages; sleeve bearings; totally-enclosed; round frame with rubber mounting rings; (1/20 to ¾ hp).

Shaded-Pole Fan Motors



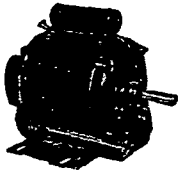
50 or 60 cycles, 115 or 230 volts; sleeve bearings; totally-enclosed; rigid, round frame, or resilient mountings; with or without 3-speed regulator.

Repulsion-Start Induction Motors



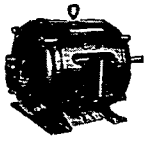
All standard frequencies and voltages; sleeve and ball bearings; open, totally-enclosed, and dripproof; horizontal and vertical; rigid and flange mounted (small sizes resilient mounted). (1/8 to 15 hp).

Capacitor-Start Motors



All standard frequencies and voltages; sleeve and ball bearings; open, totally-enclosed, and dripproof; horizontal and vertical; rigid, resilient, and flange mounted. (1/20 to ¾ hp).

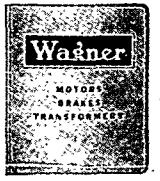
Squirrel-Cage Induction Motors



All standard frequencies and voltages, sleeve and ball bearings; open and totally-enclosed; horizontal and vertical. Made in several electrical types varied as to torque and current characteristics. (2- and 3-phase, 1/6 to 400 hp).

Wagner Direct-Current Motors

Available in dripproof frames, rigid mounted, in all standard voltages, with sleeve or ball bearings. (1/20 to 3 hp).



Write for Complete Information

Bulletin MU-185 will give you a comprehensive description of single-phase and polyphase motors.

M46-1

APPARATUS DEPARTMENT

GENERAL ELECTRIC

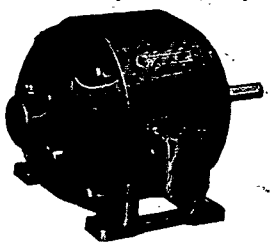
SCHENECTADY, N. Y.

Sales Offices, Warehouses, Service Shops, and Distributors in Principal Cities

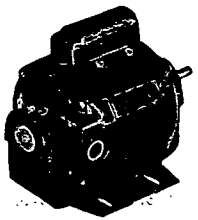
MOTORS FOR HEATING, VENTILATING, AND AIR CONDITIONING

General Electric offers a complete line of motors for compressors, fans, and pumps, from which you can select easily the motors with electrical and mechanical characteristics best adapted to your equipment. Many of the most common applications are listed below. Information on other motors—vertical, enclosed, etc., with various electrical and mechanical modifications—can be obtained at a G-E office near you.

For additional information, ask for Motor Catalog GEA-4281.



Tri-Clad induction motor, Type K, polyphase*



Fractional-horse-power capacitator-motor, Type KC

SOME G-E MOTORS AND THEIR USES

Application	Speed	Type Winding	Type	Horsepower Range	Power-supply Classification
Fans and Centrifugal Pumps	Constant or Adjustable	Shunt	B & CD	1/8—200	Direct-current
		Compound	B & CD	1/8—200	
Reciprocating Pumps and Compressors	Constant	Capacitor, Normal torque	KC	1/4—3	Single-phase, Alternating-current
		Capacitor, High-torque	KCJ	1—3	
Small Direct-connected Fans	Constant	Resistance, Split-phase	KH	1/40—1/3	
		Shaded-pole	KSP		
	Constant or 3-speed	Capacitor, Low-torque	KCP	1/50—5	
Belted Fans, Centrifugal Pumps	Constant	Capacitor, Normal-torque	KC	1/4—3	
		Repulsion-induction	SCR	5—10	
Reciprocating Pumps and Compressors	Constant or Multispeed	Squirrel-cage, Normal-torque	K	1/4—1000	Polyphase, Alternating-current
		Squirrel-cage, High-torque	KG	5—200	
Pumps, Compressors, Fans	Constant or Adjustable-varying-speed	Wound-rotor	M	1/2—1000	
	Constant	Synchronous	TS	25—2000	

Types of Enclosures: Open (dripproof)—protected from falling objects or dripping liquids. Splashproof — where wetness is a factor. Totally enclosed — for complete protection. Explosion-proof—for inflammable gases. Dust-explosion-proof—for combustible dusts.

*Trade-mark reg. U. S. Pat. Off.

APPARATUS DEPARTMENT

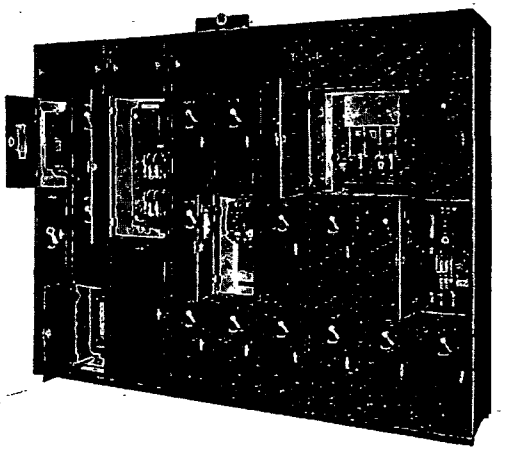
GENERAL ELECTRIC

SCHENECTADY, N. Y.

Sales Offices, Warehouses, Service Shops, and Distributors in Principal Cities

CONTROL FOR HEATING, VENTILATING, AND AIR-CONDITIONING MOTORS

General Electric offers a complete line of standard manual and automatic controls for all types of motors driving compressors, fans, pumps, etc. Publications describing these items, as well as such control accessories as pressure governors, pressure switches, float switches, electrically operated valves, and indicating selsyns, are available on request. For special applications, G-E control to meet your exact requirements can be designed. Most frequently, however, the needs of the air-conditioning industry are best served by one of the many possible G-E "packaged" control combinations designated as Cabinetrol* equipments.



Typical Cabinetrol unit, showing open motor-starter panels

The Cabinetrol system of motor control is based upon the use of standardized enclosures equipped with standard control devices. General Electric can build quickly most air-conditioning control systems—simple or complex—by properly combining standard control units and accessories into the required number of basic Cabinetrol sections. If future expansion should require further control equipment, additional Cabinetrol units can easily be added to the basic system.

The standard enclosures used in the Cabinetrol system will mount standard motor-starting devices up to and including NEMA Size 4. Ample space is provided for incoming-line, feeder, and metering equipment. Bulletin GEA-3856 details more fully the advantages of this new system of centralized low-voltage control.

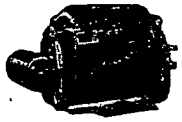
The General Electric Company will gladly assist in the solution of any electrical problem related to heating and ventilation.

*Trade-mark reg. U. S. Pat. Off.

Westinghouse Electric Corporation

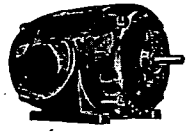
Plants in 25 Cities—Offices Everywhere

MOTORS FOR HEATING, VENTILATING AND AIR CONDITIONING APPLICATIONS



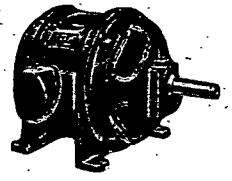
Capacitor Type FL

Induction Type FS

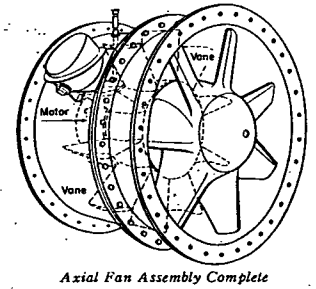


Repulsion-Start Induction-Run Type CRP

Open, Drip-Proof Squirrel-Cage Induction Type CSP



Open Squirrel-Cage Induction Type CS



Axial Fan Assembly Complete

Westinghouse manufactures a complete line of motors for all types of compressors, fans and pumps. For full details contact your nearest Westinghouse office or write for Catalog 2000 (Motors) and 7000 (Control).

FRACTIONAL HP MOTORS

For direct-connected fans on unit heaters and ventilators; propeller fans for roof ventilation; and furnace and centrifugal blowers, etc. Vertical types are recommended for "Down Blast" heaters and vertical coolers.

SINGLE PHASE A-C—Type FH—Split phase. 1/20 to 1/3 hp, 115, 230 volts.

Type FL—Permanently split capacitor design. 1/20 to 3/4 hp, 115, 230 volts.

Type FJ — Capacitor-start induction run. 1/6 to 3/4 hp, 115, 230 volts.

POLYPHASE A-C—Type FS—Induction squirrel cage design for 2 and 3 phase circuits. 1/6 to 3/4 hp, 220, 440 and 550 volts.

DIRECT CURRENT — Type FK—Single speed and multi-speed motors. 1/20 to 1 hp, 32, 115 and 230 volts.

INTEGRAL HP MOTORS

SINGLE PHASE—For fans and centrifugal pumps requiring constant speed operation. Available in standard voltages and frequencies.

Type CRP—Repulsion-start induction-Run—3/4 to 5 hp.

Type CU—Repulsion-induction—2 to 7 1/2 hp.

Type CJP—Capacitor, 3/4 to 10 hp.

POLYPHASE A-C—Type CS and CSP—Squirrel cage, constant speed, induction motors. Available in three torque classes for all standard voltages and frequencies 1/2 to 200 hp.

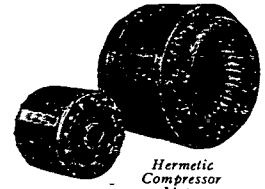
Standard enclosures include: Open, totally enclosed (fan cooled or non-ventilated) and splash-proof. Types can be supplied with Underwriters' label for Class I, Group D, and Class II, Groups E, F and G Hazardous Locations. Motors with special mounting or modifications can also be supplied.

AXIAL FAN MOTORS

Type CS—A modified design of a-c fan-cooled motor is used for axial flow fans in ventilation installations. The drawing at the left shows a typical unit, integral with the duct work. Motors from 1 to 25 hp in a variety of speeds can be supplied.

HERMETIC COMPRESSOR MOTORS

Types CS and CT feature a wound stator with machined steel shell of oil-tight construction, wrapped around the stator punchings. Insulation is suitable for operation in freon and similar refrigerbore. Rated 1 to 3 hp (single phase), 115 and 230 volts, 1 hp and larger (polyphase), 60, 50, 25 cycles, 208, 220, 440 and 550 volts.



Hermetic Compressor Motor

MOTOR CONTROL

The Westinghouse line of control includes manual or magnetic, reversing or non-reversing, across-the-line and reduced voltage starters for single or multi-speed a-c and d-c motors. Magnetic types are furnished with built-in or separate pilot devices for local or remote control. All standard NEMA enclosures are available. Representative types are described and shown below.

MANUAL CONTROL

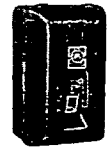
Motor Snap Switch—(Class 10-015) for starting and stopping small a-c and d-c motors where overload protection is not required. Rated up to 2 hp, 110 to 600 volts.

Sentinel Breakers—Type H—(Class 10-023) provide control and burnout protection for fractional hp motors—up to 1 hp, 110 to 220 volts. Protects motors against locked rotor or overload conditions.

"De-ion" Motor Watchman—Type DnW—(Class 10-100) designed for starting, stopping and protecting small single phase and polyphase a-c motors, from 1/6 to 7 1/2 hp, 110 to 600 volts.

Motor Snap Switch Class 10-015

Sentinel Breaker Type H Class 10-023



"De-ion" Motor Watchman Type DnW Class 10-100

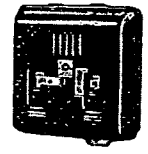
"De-ion" Linestarter Class 11-200

MANUAL MULTI-SPEED CONTROLS

Controllers for Type FL Motors—2 speed, double throw, double pole toggle action switch for operation on 115 or 230 volts.

Controllers for Type FK Motors—A variable resistor connected in armature circuit. Selector switch has six speed operations.

"De-ion" Motor Watchman—Type DnW—(Class 10-130) Starter suited to industrial fans—for use with 2 speed, 2 winding motors. 1/6 to 7 1/2 hp, 160 to 600 volts.



"De-ion" Motor Watchman Type DnW Class 10-130

MAGNETIC CONTROL

For starting squirrel-cage induction motors directly across the line. Provides remote control and complete protection when used with pushbuttons, selector switches, etc.

"De-ion" Linestarters—(Class 11-200)—Rated 1 to 25 hp, single phase, 1/2 to 750 hp polyphase, 110 to 600 volts.

"De-ion" Combination Linestarters—(Classes 11-203—fusible, 11-204—non-fusible; and 11-206—with AB nofuze circuit breaker) 1 to 200 hp, 110 to 600 volts.

Multi-Speed Controller for Type FL Motor

"De-ion" Combination Linestarter Class 11-204

PILOT CONTROL

Standard Duty Pushbuttons—Type SD-2 (Class 15-010) and **Heavy Duty Pushbuttons**—Type HD (Class 15-020)—For use with magnetic starters. Units and stations are designed for pilot control circuits 110-600 volts a-c and 115-600 volts d-c in sheet steel, water and dust-tight or oil immersed enclosures.

Dust and Water-Tight Standard Duty Station — Type SD-2 Class 15-020

Air Devices, Inc.

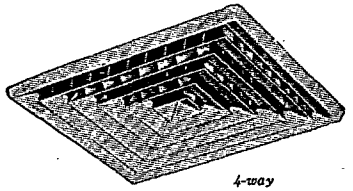
Air Diffusers ● Exhausters ● Air Filters
Filter Holding Frames ● Hot Water Generators

17 East 42nd St.
New York 17, N. Y.

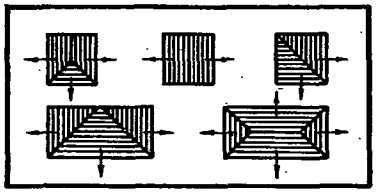
AGITAIR PRODUCTS

Agents in All Principal Cities

AGITAIR DIFFUSERS



4-way

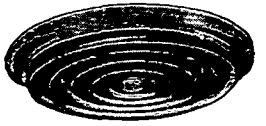


Circular AGITAIRS combine beautiful design with finest operating features to give rapid temperature equalization and draftless diffusion of air. Model CSF is quiet, easy to install. Pressure losses at minimum. Type A diffusers are smaller, weigh less, are easy to install. Type CM is for marine use. In all sizes for all types of mounting and with lighting combinations. Dampers are provided where needed.

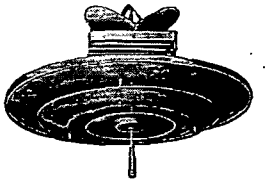
The AGITAIR Diffuser Data Book, available to architects and engineers, will help you design and install air distribution systems. Consult our engineers.

Type R AGITAIR is the only diffuser that offers the higher efficiency of the square or rectangular shape in diffusing air quietly, draftlessly and with rapid temperature equalization throughout the average square or rectangular room. Its patented construction can be assembled into patterns which discharge the required amount of air in one to four directions.

Each side delivers a quantity of air proportional to the areas served. Thus the engineer or architect can select an attractive diffuser that fits his design—rather than make his design fit the diffuser.



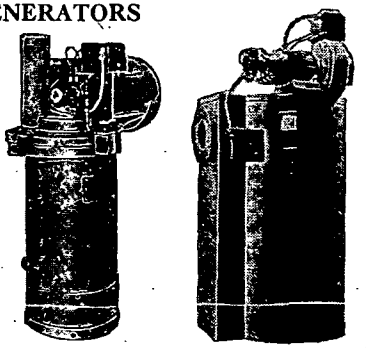
Type A 125



Model CSF with Type B Damper

OIL BURNING HOT WATER GENERATORS

Ideal for hot water or heating service in domestic, apartment house, industrial and marine applications because they are light in weight and can be installed without special foundations or floor coverings. The AGITAIR is down-fired to conserve floor space and facilitate servicing of the burner. Burner is fully automatic, with simplified design to reduce weight, cost of installation and possibility of failure. Special stacks are not necessary. *Type A*, weight 300 lb, 400 to 1000 sq ft radiation. *Type B* in larger capacities from 1000 to 4500 EDR. *Ask for bulletin.*



AGITAIR AIR FILTERS

Permanent, cleanable AGITAIR Air Filters give more effective filtering with greater dust holding capacity and exceptionally low resistance.

The new, improved filter medium divides the air stream into myriads of streams which impinge upon and deposit dirt and dust on the viscous coated surfaces. Large interstices provide exceptional dust-holding capacity—AGITAIR serves longer without cleaning.

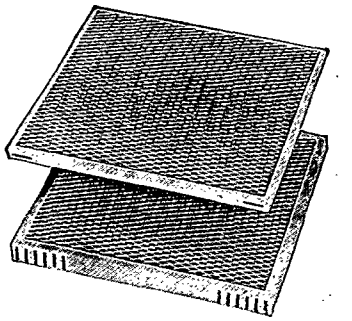
Ideal for applications where filters are in service for many hours without a "rest" period, because AGITAIR'S viscous coating has a vigorous "wick action" that keeps the surface of the dust deposit always wet, no matter how fast the dust cake builds up.

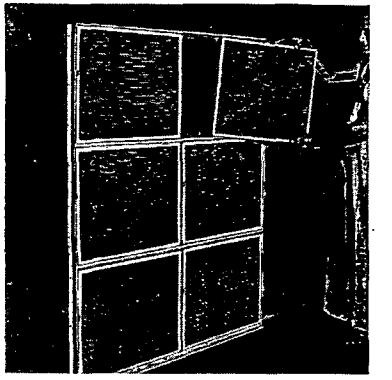
Construction is sturdy arc-welded steel, with protected faces to withstand severe cleaning abuse. They are easier to clean, and can be returned to top efficiency with little trouble.

Heating, ventilating, and air conditioning: AGITAIR Air Filters in thicknesses of 1 in. and 2 in. Sizes up to 1200 sq in.

Industrial: Heavy-duty AGITAIR filters 2 in. and 4 in. thick. Sizes up to 2400 sq in. AGITAIR engineers have solved many industrial air filtering problems, such as protection for heavy blowers and other equipment. Call on our experience.

Grease Filters: For kitchens, etc. Capacity 1½ cfm. per sq in. with fan system; 1 cfm. per sq in. on gravity exhaust. Standard sizes up to 25 x 30 in.; others to 2400 sq in. Complete assemblies available.





Typical Filter Holding Frame

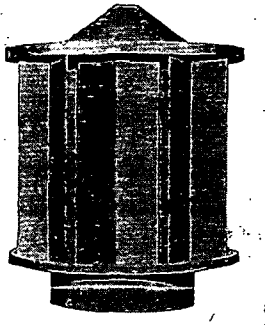
Made of heavy steel, assembly is sturdy, neat. Available for all standard and special sizes of AGITAIR filters.

V-Type assemblies of AGITAIR filter holding frames are available to increase filter area where space is limited.

FILTER HOLDING FRAMES

Adaptable to all types of installations. Saves time and labor, eliminates bolts and rivets in frame, and minimizes air leakage.

AGITAIR WIND-ACTUATED EXHAUSTERS



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 flow of air or gases when there is no movement of outdoor air across the head.

Available in sizes from 4 in. to 48 in. neck diameter, with capacities shown at right. For calculation of natural ventilation requirements, and for proper design principles, refer to Chapter 9 of this GUIDE. Fan-equipped units also for higher ratings. *Bulletin EX-101.*

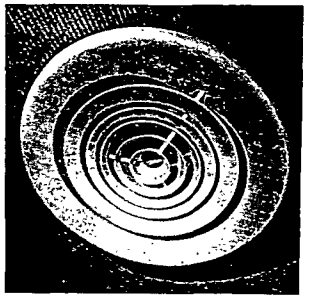
Capacity—Wind-Actuated

Size of Vent O. D. Neck	Neck Area Sq Ft	Nominal Capacity cfm
S 4"	0.0873	38
S 6	0.1964	86
S 8	0.3491	154
S 10	0.5454	240
S 12	0.7854	346
S 14	1.069	470
S 16	1.396	614
S 18	1.767	777
S 20	2.182	960
S 22	2.640	1162
S 24	3.142	1382
S 28	4.276	1881
S 32	5.585	2457
S 36	7.069	3110
40	8.727	3840
44	10.56	4626
48	12.51	5500

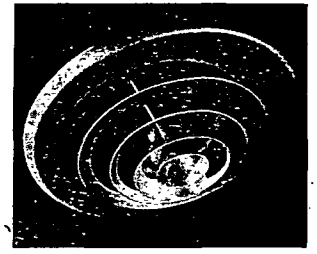
Anemostat Corporation of America

10 East 39th Street, **New York City 17, N. Y.**

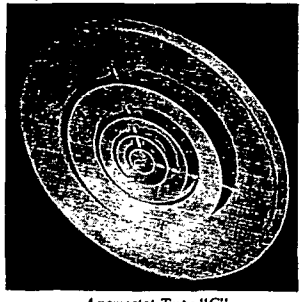
The Anemostat High Velocity Air Diffuser



Anemostat Type "AC"
For supplying, extracting or returning air.



Anemostat Type "B"
For diffusion of supply air.



Anemostat Type "C"
For diffusion of supply air only.

THE ANEMOSTAT PRINCIPLE

ANEMOSTATS produce unparalleled results because of their unique design which operate on the following interdependent principles:

1. Air expansion within the device, which reduces velocity instantly.
2. True Aspiration, which causes room air equal to 30 to 35 per cent of the supply air to be drawn into the device where it is mixed with the supply air. The percentage of aspiration depends on the type of Anemostat used.
3. Creation of a multiplicity of air currents and counter-currents at low velocities, which causes slow but adequate secondary air motion.

Type "AC" Anemostat is a combination device for supplying air and either extracting it or returning it to the conditioner. Designed to extract or return 75 cfm of room air for every 100 cfm of supply air. This percentage of extract or return may be increased or decreased by varying the extract velocities. It furthermore has an aspiration effect of 30 per cent. May be used with velocities up to 2500 fpm, and wherever both supply and return, or extract are required through the same unit. Should not be used with ceiling heights exceeding 14 ft.

Type "AR" Anemostat is a diffusion device for supply air only, for use in applications where a relatively high rate of air change (more than 12 per hour) must be maintained. It has 35 per cent aspiration. It is particularly suitable where obstructions such as beams or columns are close to the diffusion unit and restrict the normal radius of diffusion.

Type "B" Anemostat is a diffusion device for supply air only. It has 35 per cent aspiration. May be used with velocities up to 4000 fpm and is suitable for industrial and commercial installations. Can be used on either exposed or concealed duct work.

Type "C" Anemostat is a diffusion device for supply air only. It has 35 per cent aspiration. May be used with velocities up to 2500 fpm. Must be installed flush with ceiling and cannot be used on exposed duct work.

Type "HU-3" and "HU-4" Anemostats have been developed to obtain draftless, economical heat distribution from vertical discharge unit heaters, and uniform heat coverage of floor areas. Type "HU-3" and "HU-4" Anemostats may be combined with practically all sizes and types of Vertical Discharge Heaters on the market. A number of unit heater manufacturers now supply "HU" Anemostats as a part of their equipment. "HU" Anemostats may be used on duct work for general heating installations.

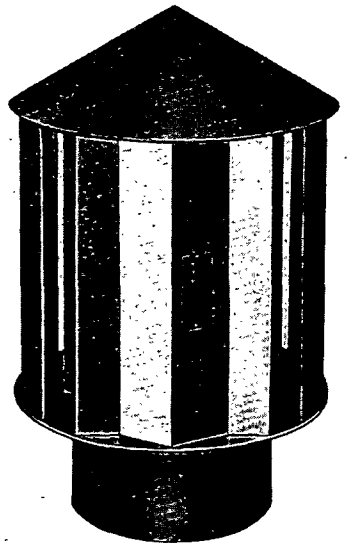
"No Air Conditioning System is better than its Air Distribution"

G. C. Breidert Co.

634 South Spring St., **Los Angeles 14, Calif.**

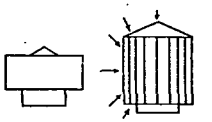
Representatives Located in Principal Cities of the U. S.

BREIDERT AIR-X-HAUSTERS FOR ROOF VENTILATING, VENT FLUES & CHIMNEY TOPS



Type B

The Breidert Air-X-Hauster introduces a new principle in ventilator design. Because of the revolutionary, aerodynamically-correct design of the Breidert Air-X-Hauster, wind currents striking it from *any* angle are converted into a powerful suction force that rapidly exhausts stale air from the interior of the house, kitchen or building. The Breidert remains stationary, has no moving parts. *Back-drafts are eliminated where there is no interior negative pressure!*



Old Method — *Breidert Method*

The Breidert ventilator offers *certified* capacity ratings based on tests made with wind blowing at *all* angles (as shown). These high capacities were proved and certified by Smith, Emery Co., Pacific Coast branch of Pittsburgh Testing Laboratories. *Insist* on certified ratings based on directional wind tests at various vertical angles as shown in considering *any* ventilator. Breidert Air-X-Hausters were used extensively during the war on many types of combat and cargo ships, war housing, military barracks, and other government buildings. They also are widely used on all types of factories, commercial buildings, and residences.

For Kitchen Ventilation . . . The Breidert system provides a continuous, silent, effective circulation of air that exhausts heat and odors *at their source*, with no operating or maintenance expense. There are no "hang-over" cooking odors because the exhaust action of the Breidert is continuous. The neat, compact appearance of the "Type A" Breidert especially recommends it for residences.

For Vent Flue Caps . . . The Breidert does not have the defects of conventional types of caps and accessories. It eliminates the necessity for down-draft diverters, with accompanying dangers of explosion in case unburned gas accumulates or is blown into the room.

For Chimney Tops . . . By stopping all down-drafts (interior negative pressure excepted), a Breidert Air-X-Hauster on the chimney absolutely prevents the fireplace from smoking and damaging furnishings. *It provides positive "draw" regardless of wind direction!*

Write for Free Engineering Data Book . . . contains specifications and installation data, certified capacity ratings, etc. Address Dept. HV.

Barber-Colman Company

Rockford, Illinois

ENGINEERED AIR DISTRIBUTION OUTLETS

VENTURI-FLO
Overhead Air Diffusers

Venturi-Flo overhead air diffusers have flow characteristics similar to those of the well known fluid-flow measuring device—the Venturi Meter. The relationship between the neck area of the unit proper and the Venturi-Flo throat area is so proportioned as to create a slight back pressure in the neck at all times, thereby automatically insuring uniform distribution around the entire periphery of the unit.

Three types of overhead diffusers are available, the recessed, the surface, and the thermostatically controlled types. A wide range of sizes permits handling air volumes up to 15,000 cfm per unit. Fittings for attaching any standard light fixtures to the outlets may be obtained for all designs. The surface and recessed types can also be furnished as combination supply and exhaust units and with adjustable dampers.

Uni-Flo Grilles and Registers are especially designed for air conditioning applications. They are engineered and prefabricated with directional flow fins for each individual application. Proper air distribution is assured and the necessity for adjustment after installation obviated.

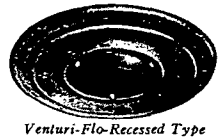
Uni-Flo Grilles and Registers can be furnished in a variety of shapes and sizes for plain and curved surfaces.

Registers are similar in construction to grilles but with the addition of spring loaded positive closing fan or key-operated dampers.

Electroplated Finishes: Gunmetal, brushed bronze, plain zinc, and satin copper; also available in gray prime coat and satin aluminum.

Uni-Flo Air Distribution Accessories: Include the Volocitrol and the Airturn. The Airturn is a scientifically designed and highly efficient air turning member. The aero-dynamically correct vanes reduce losses caused by eddies, reverse air flow and low pressure areas at the turn to a minimum. Airturns are prefabricated and are available in units of one piece construction in sizes up to and including 48 in. x 48 in.

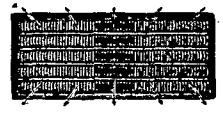
The Volocitrol has been designed to noiselessly provide positive and adjustable control of air volume, pressure and distribution across a supply outlet. They are available in two models, one for use with side wall and the other for overhead systems of air distribution.



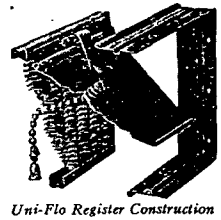
Venturi-Flo-Recessed Type

Venturi-Flo-Surface Type

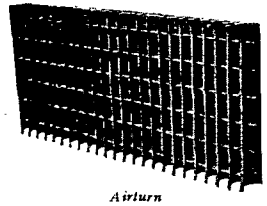
Venturi-Flo-Thermostatically Controlled Type



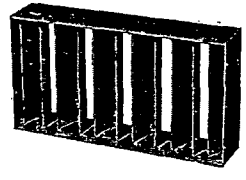
Uni-Flo Grille Model "M"



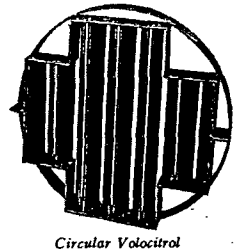
Uni-Flo Register Construction



Airturn



Volocitrol



Circular Volocitrol

SEE OUR COMPLETE CATALOG IN SWEETS

W. B. CONNOR ENGINEERING CORP.

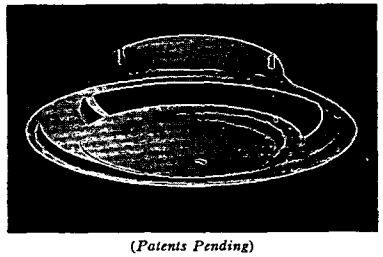
114 East 32nd Street,
New York 16, N. Y.

Offices
in All Principal Cities

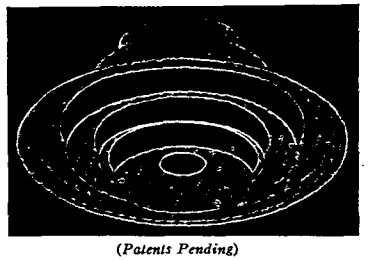
Canadian Representative: D & D Air Conditioning Co., Montreal, Canada

Manufacturers of KNO-DRAFT Adjustable Ceiling Air Diffusers

KNO-DRAFT Adjustable Air Diffusers insure efficient air distribution, maximum secondary air induction, noiseless and draftless diffusion, and uniform temperature throughout the occupied zone, *regardless* of season or ventilation requirements. Every KNO-DRAFT unit is easily and quickly adjustable for individual or seasonal requirements. The same KNO-DRAFT Diffuser is equally effective to expel chilled air parallel to the ceiling or eject heated air downward to prevent stratification.



(Patents Pending)



(Patents Pending)

MODEL K KNO-DRAFT Diffuser—For Supply Air—attractive—light, yet sturdy—for high or low ceilings or attachment to exposed duct work. Anti-smudge rim prevents streaked ceilings. Sizes 4 in. to 40 in. neck diameter for capacities from 50 cfm to 20,000 cfm per unit.

MODEL SR KNO-DRAFT Diffuser–For Combination Supply and Return air to simplify duct work. Sizes 6 in. to 24 in. supply-air neck diameter for supply capacities from 50 cfm to 9,000 cfm per unit, with central, return neck area 75 per cent of supply neck area.

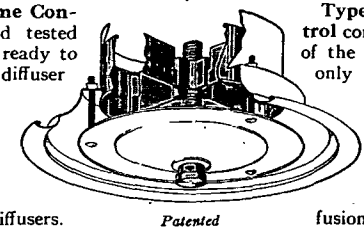
The KNO-DRAFT Diffuser will effectively distribute large volumes of air—premixing room and supply air. It permits the use of higher duct velocities—resulting in smaller ducts and lower costs. Duct designs are simplified and fewer outlets are required. KNO-DRAFT Diffusers blend well with any architectural treatment. They are simple in construction, light in weight, and easily installed.

KNO-DRAFT Diffusers are geometrically proportional, size for size, insuring like resistance at like neck velocity for any size—and balanced static pressure throughout the system.

THE TYPE D AIR VOLUME CONTROL is designed for application exclusively to KNO-DRAFT ADJUSTABLE CEILING AIR DIFFUSERS. It is furnished already assembled within the diffuser and requires neither field assembly nor attachment to ducts, angle rings or other external appurtenances.

Type D Air Volume Control is adjusted and tested before shipment and ready to function when the diffuser is installed.

Its operation is entirely independent of the air direction adjustment which is part of all standard KNO-DRAFT Air Diffusers.



Patented

Type D Air Volume Control complements the function of the air diffuser. It varies only the *quantity*, not the *characteristic* of the air distribution. With it, a series of diffusers may be balanced without affecting the air diffusion efficiency.

With KNO-DRAFT ADJUSTABLE CEILING AIR DIFFUSERS
equipped with TYPE D AIR VOLUME CONTROL,
the desired air pattern for any room or zone is AT YOUR FINGER TIP.

Charles Demuth & Sons

Mineola, N. Y.

Air Distributors—Oil-Fuelled Conditioners

THE Demuth air distributor is known in the air conditioning industry as the distributor with the curved, vertical vanes. This special design feature gives the air supply a slightly rotative motion while being released over a 360-deg rim throw.

This provides air-release turbulence to prevent drafts yet aids in corner-wiping and unusually thorough mixture of fresh air supply. There is also created a central vortex in the unit that recirculates a substantial volume of room-air. Results are uniform temperature in areas treated, absence of decoration smudge and no draft complaints.

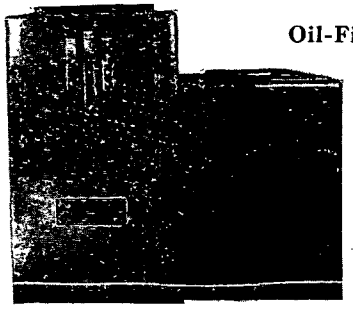
Type S-2-L, Left; Type S-2, Center; Type S-L, Right

Many Finishes

To make the Demuth a handsome fixture as well as a practical piece of apparatus we offer a wide variety of finishes. There are: aluminum and stainless steel, polished and satin (machine produced;) ordinary steel in colored enamel finish; also galvanized steel, unpainted or enamelled, for marine and industrial applications.

All pieces spun in manufacturer's own shop. Sizes from 4-in. to 42-in. duct diameters.

The Demuth is also available in combined lighting styles. Type S-2-L our semi-recessed model with standard globe. Type S-2 (above) our current popular semi-recessed design and Type S-L, also seen, shows how practically any design of lighting fixture can be combined with our distributors for special purchaser preferences.



Oil-Fired Winter Air Conditioner

Heats, humidifies, filters and circulates the air. Five models rated from 80,000 Btu to 300,000 Btu. Insulated casing, combustion chamber refractory lined.

All-steel, all-welded construction of Demuth design. Oil burner (gun type). Heat interchanger provides long gas travel before release.

Fuel economy a special point, together with quick heating and troublefree performance. Enamelled finish, with polished aluminum trim. Unusually compact casing for heating capacity offered per unit.

Hendrick Manufacturing Company

Hendrick Perforated Metal Grilles

48 Dundaff Street, **Carbondale, Pa.**

SALES OFFICES IN PRINCIPAL CITIES—CONSULT TELEPHONE DIRECTORY

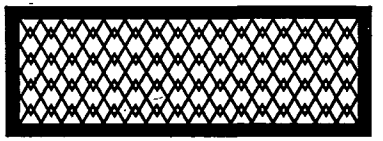
PRODUCTS—Hendrick Perforated Metal Grilles; Mitco Open Steel Flooring; Mitco Armorgrids; Mitco Shur-Site Treads.

HENDRICK PERFORATED METAL GRILLES

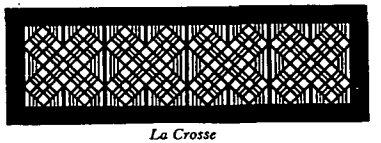
To architects, engineers, contractors and others who buy or specify grilles, Hendrick offers literally hundreds of designs from which to select the pattern or patterns best suited to the specific application.

Among their many decorative applications, Hendrick Perforated Metal Grilles are used in front of multiple louvre control for air direction. The wide range of patterns permits perfect harmony with every style of architectural design or period construction.

Musak



M No. 9



La Crosse

In addition to those popular designs which have been specified so consistently that they are today regarded as standard patterns, Hendrick offers a number of exclusive designs, many of them covered by design patents.

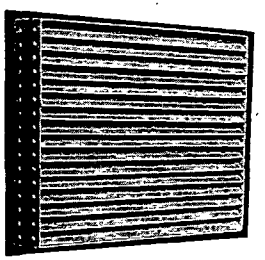
Originally designed to meet specific requirements, these Hendrick designs are available, without premium, to those who seek something that is distinctive as well as different.

All Hendrick Grilles are characterized by clean-cut perforations and fine finish. In addition, Hendrick Grilles are put through a special flattening operation which insures easy installation.

HENDRICK FIXED LOUVRE GRILLE

One of the most popular grilles in the Hendrick line is a door grille, developed originally for hotels and hospitals but equally ideal for bathroom doors in residences.

Hendrick Fixed Louvre Grille is built up of a series of strips bent to a fixed angle and rigidly fastened into a band frame, a construction permitting free circulation of air—but preventing vision through the grille from any angle. Easily installed in any door.



Fixed Louvre Grille

Regularly furnished in No. 18 U. S. Gauge Steel; also obtainable in other desired commercially rolled metals; thickness 1⅛ in. to fit standard door requirements.

Write on your letterhead for a copy of 194-page handbook, "Hendrick Grilles."

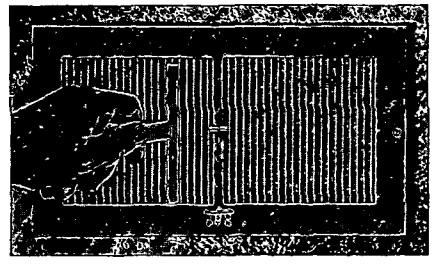
Hart & Cooley Manufacturing Co.

Established 1901

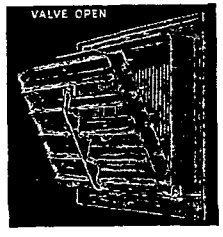
Air Conditioning Registers and Grilles - Warm Air Registers
Damper Regulators - Furnace Regulators - Pulleys - Chain

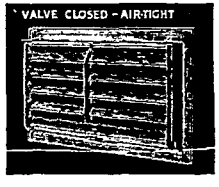
Holland, Mich.

NO. 75 DESIGN—FLEXIBLE FIN TYPE with TURNING BLADE VALVE to provide DOUBLE DEFLECTION. Also without Valve as Grille or Intake



CONTROL OF AIR FLOW IN TWO PLANES





Instant Adjustment of Air Flow (Up, Straight or Down)

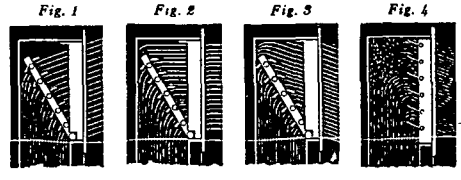
Is obtained by turning the regulator on the register face to the proper setting with a key furnished with each register. When the valve is opened, as shown at the left, the individual valve louvres automatically stop in position to provide the proper air flow—Up (Fig. 1) for cooling systems to avoid drafts; Straight (Fig. 2) for ventilating systems; Down (Fig. 3) for heating systems to prevent stratification. When the valve is closed, as shown at the left below, it completely stops the flow of air.

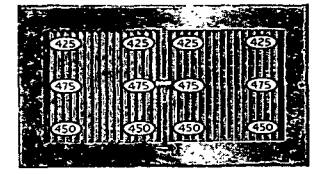
Air Flow Can be Quickly Adjusted Sideways

No. 75 Design has a flexible fin-type face. Each fin may be twisted individually with a wrench furnished with each register or grille to provide any desired sideway deflection of the air flow.

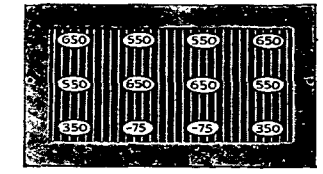
Greatly Reduced Turbulence and Resistance

Figs. 1, 2, and 3 show the air flow with No. 75 Design; Fig. 4, with the conventional register. Compare the turbulence in the stackhead of the latter with the smooth flow obtained with No. 75 Design. So efficient is No. 75 Design that there is actually less resistance with this register, using a standard stackhead, than if no register at all were used.





Velocities with No. 75 Design



Velocities with Conventional Register

EVEN DISTRIBUTION OF AIR OVER ENTIRE FACE

The turning blade valve distributes the air evenly with a uniform velocity over the entire face, as shown in Figs. 1, 2, and 3 on the preceding page. Note how the air rushes through the upper part of the face with a conventional register, as shown in Fig. 4. Since the entire face of No. 75 Design register is utilized for discharge of air, smaller and in some cases fewer registers can be used without causing excessive velocities.

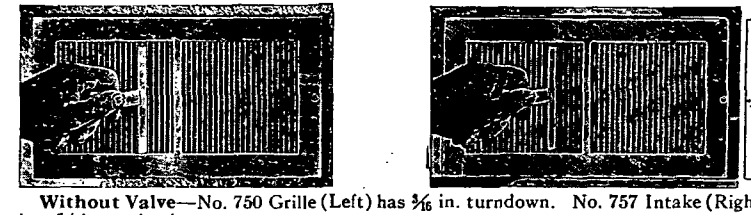
Prevention of Streaked Ceilings—With either UP, STRAIGHT, OR DOWN deflections the air does not strike the ceiling immediately in front of the register; streaked ceilings are thus avoided.

Excellent Concealment of Duct—The depth and close spacing of the vertical bars, combined with the valve, provide almost complete concealment of the duct, adding considerably to the pleasing appearance of the register face.

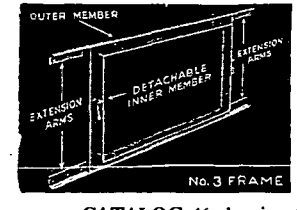
Special Settings—No. 75 Design functions equally well when located at the end of a horizontal duct or, by installing it upside down, when the air is delivered to it from above.

AVAILABLE IN FOUR TYPES

With Turning Blade Valve—No. 751 Register (Left) has Sponge Rubber Gasket and 3/16 in. turndown. No. 754 Register (Right) is similar except has 7/8 in. projection.



Without Valve—No. 750 Grille (Left) has 3/16 in. turndown. No. 757 Intake (Right) has 7/8 in. projection.



FOUR TYPES OF INSTALLATION FRAMES AVAILABLE

No. 75 Design items can be used with or without installation frames. No. 3 Sidewall Stud Frame (illustrated), fastens directly to stud, forming a solid, streak-proof foundation for register. No. 8 Frame is similar for baseboard use. No. 5 Baseboard Stack Frame provides inexpensive, streak-proof installation. No. 2 Band Iron Frame provides for connecting register to stackhead.

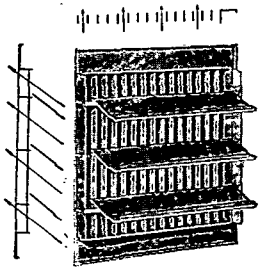
CATALOG 46 showing the complete H & C line, available upon request.

The Independent Register Co.

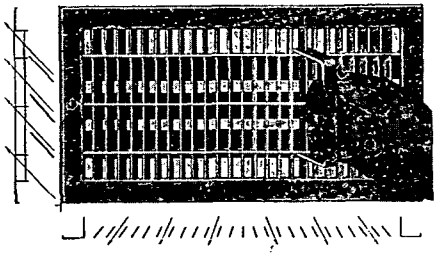
ESTABLISHED 1898

3747 East 93rd Street, **Cleveland, Ohio**

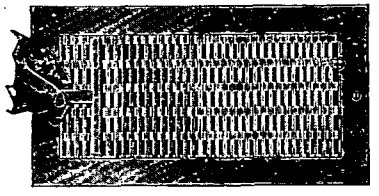
AIR CONDITIONING REGISTERS AND GRILLES



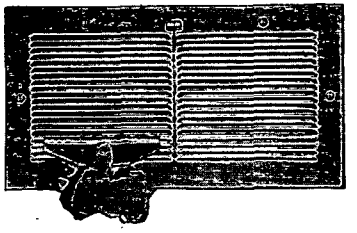
Rear View
Showing Adjustable Deflecting Vanes



No. 321A Grille with Deflecting Vanes—With vertical grille bars and horizontal deflecting vanes. The grille bars may be individually adjusted to direct air flows to right or left; and the vanes are made individually adjustable to deflect air flows up or down.



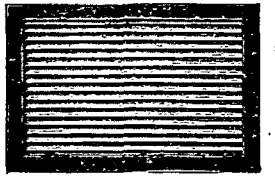
No. 238 Wrought Steel—4-way adjustable direction of air flow. Flexible vertical grille bars, multiple valves.

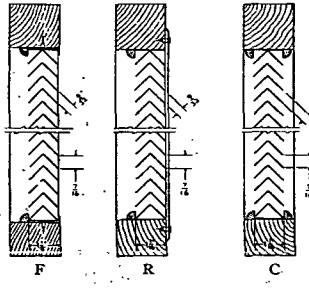


No. 139 Wrought Steel—Flexible horizontal grille bars, bendable for up, down or straight air flow. Single valves.

Independent No-Vision Grilles—No. 1312 for Doors, Walls and Partitions

The grille bars are "V" shaped; it is impossible to see through the grille from any viewpoint.





F R C

No. 1312F—One outer frame edge turned inward. See "F".

No. 1312R—With overlapping rim 5/8 in. wide, on all four sides.

No. 1312C—With grille core only, installed with moulding.

The Pyle-National Company

1334-58 North Kostner Avenue, **Chicago 51, Illinois**

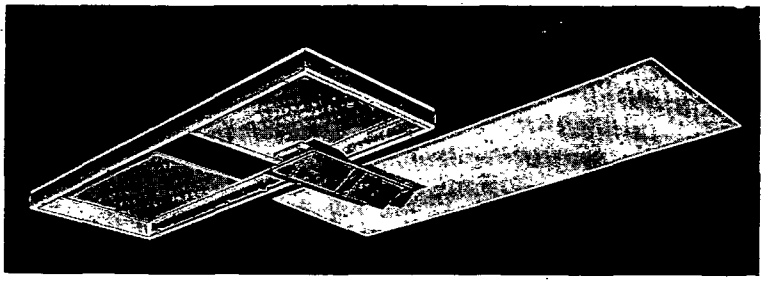
Representatives in Principal Cities

Manufacturers of

MULTI-VENT

Panels

for Heating • Ventilating • Air Conditioning



Control Plate *Valve* *Distribution Plate*

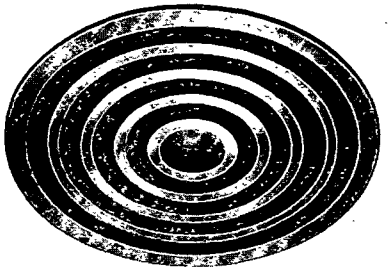
MULTI-VENT Panels are used for the even distribution of warm, cool or fresh air within room spaces. The control plate frame is inserted in the overhead duct at the ceiling. The orificed adjustable air valve provides absolute displacement of static head, and may be set for varying quantities of air to enter the conditioned space, and at varying velocities or speed of entry, producing the comfort zone rate of air motion and degree of temperature desired by the occupant (provided the system includes the required auxiliaries). The perforated distribution plate, of large area, located in the ceiling, accomplishes a wide, gentle, uniform spread and dilution of the conditioned air and simultaneously provides panel heating or cooling. When used with the other correct auxiliaries Multi-Vent results in uniform temperatures and comfort.

Application data will be furnished on request

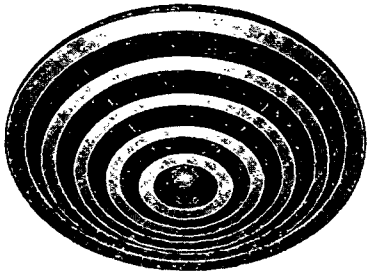
Tuttle & Bailey, Inc.

New Britain, Connecticut

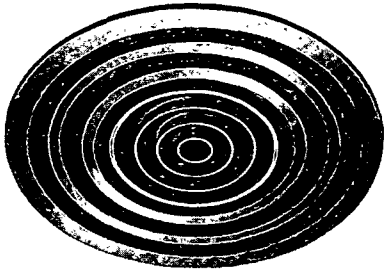
PRODUCTS: Ceiling Diffusers—Grilles—Registers—Intakes—Air Control Devices—Ornamental Grilles, Cast or Wrought Metals—Copper Convector Heaters



Type S1



Type E2



Type R1

AEROFUSE OUTLET

FOR HEATING

VENTILATING AND

AIR CONDITIONING

TYPE S

Flush-type diffuser for installation on ceiling. Ideal combination of real beauty and functional superiority. Provides (1) Complete Air Mixture (2) Rapid Temperature Equalization (3) Uniform Air Distribution (4) Total Elimination of Drafts.

TYPE E

Type E Outlets are designed for installation on exposed ductwork and provide the same efficient performance as the S Type. The rings of Types E2 and E3 are stepped down, which greatly increases the capacity of a given size of outlet, resulting in an appreciable saving in the cost of the outlet and the ductwork.

TYPE R

Combination supply and return (or exhaust) unit. Designed particularly for use on installations where simplification of the duct layout is of primary importance since the return (exhaust) duct can be run to the same location as the supply duct.

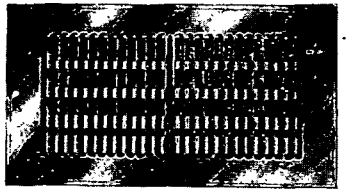
Send for complete engineering data and descriptive catalog.

Tuttle & Bailey, Inc.

New Britain, Connecticut

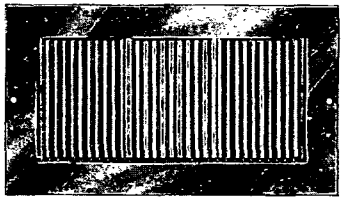
PLIAVANE ADJUSTIBLADE REGISTER

An inexpensive register suitable for low cost housing installations. The air flow may be directed sideways by the individually adjustable face vanes and up or down by the back blades which can be "adjusted" from the face of the register itself.



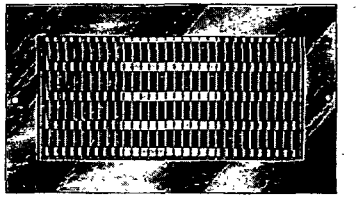
AIR CONDITIONING GRILLES

Furnished in both the fixed deflection (Airline Design) and the sectionally adjustable deflection (Flexair Design) types with bars running either horizontally or vertically. The Tuttle & Bailey Air Conditioning Grille is scientifically and sturdily constructed to perform efficiently under all operating conditions.



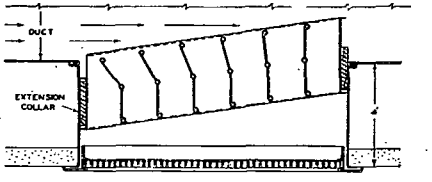
DOUBLE DEFLECTION GRILLES

Furnished with face bars of fixed (Airline) or adjustable (Flexair) deflection types and equipped with individually adjustable back blades. Also available with horizontal face bars and vertical back blades.



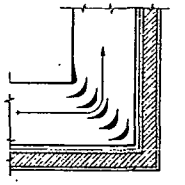
SANTROLS

An individually adjustable series of deflecting blades for positive control of air volume and to insure uniform distribution over the entire supply outlet.



DUCTURNS

Scientifically designed turning blades which eliminate the necessity of long radius turns and allow the use of right angle elbows in a duct system.



REGISTER & GRILLE MFG. CO.

Incorporated

70 Berry Street, **Brooklyn 11, N. Y.**

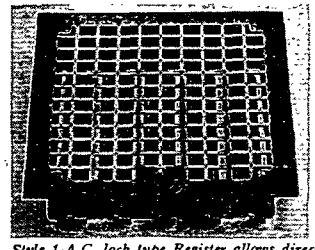
Headquarters for all types of Registers and Grilles

RESIDENTIAL AND COMMERCIAL

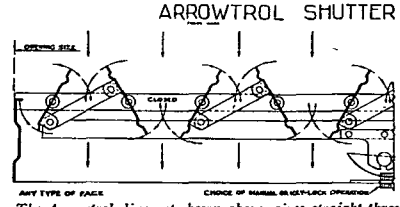
Register shutters of different types can be furnished with all types of Register Faces or Grilles.

All Register Shutters have our exclusive feature of brass collars inserted in the ends of the shutter to minimize rusting.

REGISTERS FOR VENTILATION

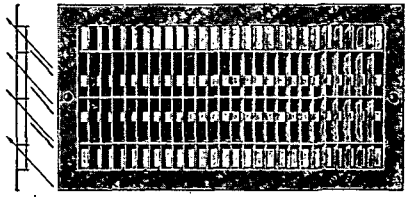


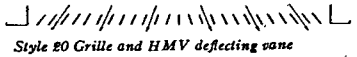
Style 1-A.C. lock type Register allows directional flow of 135 deg either right and left or up and down. Will open 45 deg beyond 90 deg



The Arrowtrol, line cut shown above, gives straight throw in connection with volume control

FOUR-WAY DEFLECTION TO AIR FLOWS

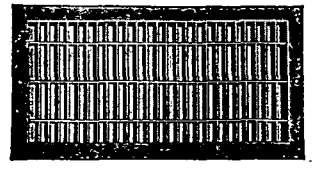




Style 20 Grille and HMV deflecting vane

Front bars vertically adjustable, rear vanes horizontally adjustable; *or* Front bars horizontally adjustable, rear vanes vertically adjustable.

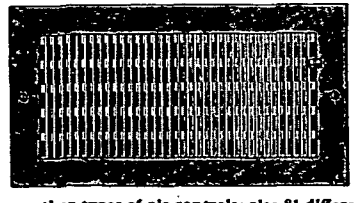
R & G ADJUSTABLE DIRECTED AIR FLOW TWO-WAY DEFLECTION



Use No. 20 Grille for adjustable right and left deflection. Style 10 has horizontal adjustable bars for up and down deflection.

THE "THIN MAN" REGISTER FOR RESIDENTIAL USE

Style 8 T.M., shown, allows right and left deflection and up or down control at the back.



Other designs of faces are available.

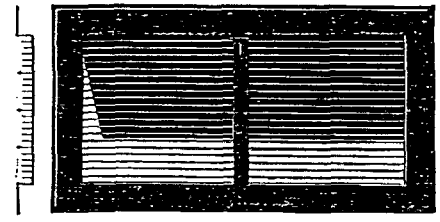
Ask for our catalog which shows other types of air controls; also 81 different Stamped Metal designs and over 100 designs in Cast Metals—Iron, Brass or Bronze.

United States Register Company

General Offices: **Battle Creek, Mich., U.S.A.**

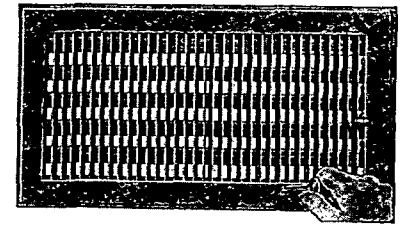
Branches: MINNEAPOLIS, MINN., KANSAS CITY, MO., ALBANY, N. Y., NEW YORK, N. Y., SAN FRANCISCO, CALIF.

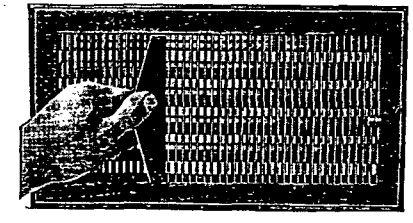
Air Conditioning Registers, Vents and Grilles



Style 153LF—Louver-Type Air Conditioning Register-Bars ¼ in. deep—Spaced 4 openings to the inch affords Non-Vision. Can be supplied in Directional Flow in either Horizontal or Vertical Bar Styles. Can be furnished with all styles of Setting Frames.

Style 249LF—Duo-Deflection Air Conditioning Register. Gives complete Air Control. Vertical Front bars—Key-pin adjusted to provide 45 deg Right and Left or Two-way Side Flow. Lever operated Horizontal Back-valves give from Full Closed to any degree of Up-flow and to 45 deg Down-flow. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads of Standard Size Dimensions.



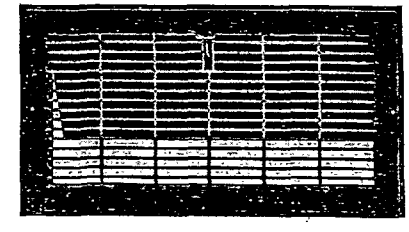


Style 256LF—Flex-bar Air Conditioning Register. Vertical Front Bars set 22 deg Right and Left. Side Flow Deflection attained by setting of Grille Bars with bending wrench to accommodate room condition. Back-valves give same Up and Down control of air flow as 249LF above. FULL FACE COVERAGE. Can be supplied with any style of Setting Frame. Fits all Stack Heads of Standard Size Dimensions.

All of above Styles can be supplied with either Lever or Individually adjusted Multiple Valves or Louvers. *i.e.* 153VVI—Vertical Valves Individually adjusted. 145VVL—Lever operated Vertical Valves.

Grilles and Vents in Matching designs are available.

For Complete Information Write for Latest Catalogs with Engineering Data.



Style 103LF—Horizontal Lattice Perforated Register for Forced Air Systems. Not directional flow.

Complete Gravity and Air Conditioning Register, as well as Fitting Catalogs furnished on request.

Young Regulator Company

5209 Euclid Avenue, **Cleveland 3, Ohio**

DAMPER REGULATORS; REMOTE CONTROL REGULATORS; DAMPERS

Manually and Automatically Controlled

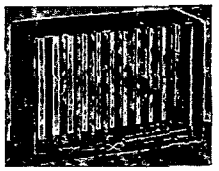
Sales Representatives in Principal Cities

No. 700A

Ten designs of Young Damper Regulators, which meet almost every condition where regulators are required for controlling air volume through ducts. They are positive in action, tamper-proof, withstand vibration and shock. Are operated and locked by key.

Young Remote Control System No. 700A and No. 805

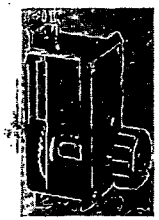
Individually controls volume of air to each room. Operates dampers at a distance of 250 feet or more. Depth of box ¾ in. Corner pulleys No. 704 with connections for tubing used at bends.



No. 805

Young Remote Control Regulator No. 703A

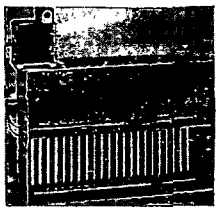
Cutaway view with box 2 in. deep.



No. 703A

Young Remote Control No. 702

Has reverse dial, used when cable leads off bottom of Control.



No. 815

Young Relief Damper No. 815

Automatically controlled, with modulating low voltage motor and remote bulb thermostat.

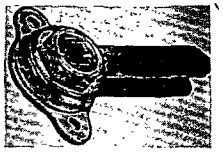
Young Fresh Air Damper No. 820

Is for outside air. Is similar to No. 815 except that it is a single instead of double damper.

No. 401A

Young Surface Damper Regulator No. 1A—⅜ in.

Young Valcalox Damper Regulators No. 400A—5/16 in. and No. 401A—⅜ in.

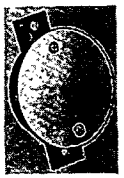


No. 402A

Young Air-Split Damper Regulator No. 900A

Young Air-Split Damper Regulator No. 905

Is similar to No. 900A except the rotating head is placed on inside of Grille.

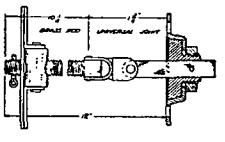


No. 301A

Young Convector Regulator No. 910

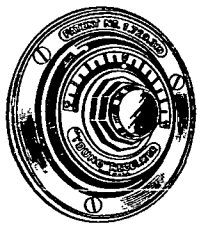
Young Volume Control Grille No. 915-L

This assembly is for installation on the side of a duct having a series of outlets.

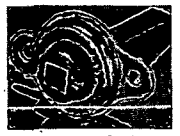


No. 900A

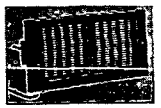
Young Concealed Regulators No. 300A—5/16 in. and No. 301A—⅜ in.



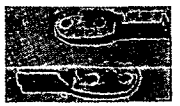
No. 1A



No. 655—5/16 in.
No. 656—⅜ in.



No. 915L



No. 602—5/16 in.

The American Rolling Mill Company

Executive Offices, **Middletown, Ohio**

ATLANTA 3, GEORGIA
1437 Citizens & Southern National Bank Bldg.
BERKELEY 2, CALIFORNIA.....Seventh and Parker Streets
BOSTON 16, MASSACHUSETTS.....439 Park Square Bldg.
BUFFALO 2, NEW YORK.....501 Seventeen Court St. Bldg.
CHATTANOOGA 2, TENN.....602 Chattanooga Bank Bldg.
CHICAGO 4, ILLINOIS.....310 S. Michigan Bldg.
CINCINNATI 8, OHIO.....24 Cooper Bldg., Hyde Park
CLEVELAND 15, OHIO.....1216 B. F. Keith Bldg.
COLUMBUS 15, OHIO.....Room 431, Atlas Bldg.
DALLAS 1, TEXAS.....2112 Tower Petroleum Bldg.
DAYTON 2, OHIO.....506 Mutual Home Bldg.
DETROIT 2, MICHIGAN.....5-261 General Motors Bldg.
HOUSTON 2, TEXAS.....1640 Commerce Bldg.
INDIANAPOLIS 4, INDIANA.....1106 Fletcher Trust Bldg.
KANSAS CITY 3, MISSOURI.....7100 Roberts St.
LOS ANGELES 15, CALIFORNIA.....329 Petroleum Bldg.
MILWAUKEE 2, WISCONSIN,
627 First Wisconsin National Bank Bldg.
MINNEAPOLIS 14, MINN.....171-27th Avenue, S.E.
NEW YORK 5, NEW YORK.....120 Broadway
PHILADELPHIA 7, PA.....1808 Lincoln-Liberty Bldg.
PITTSBURGH 22, PA.....1627 Henry W. Oliver Bldg.
RICHMOND 19, VA.....1308 State Planter's Bank Bldg.
ST. LOUIS 1, MISSOURI.....1725 Ambassador Bldg.
SOUTH BEND 24, INDIANA,
503 City Natl. Bank & Trust Bldg.

Choose the Correct ARMCO Grade

There is a grade of ARMCO sheet metal particularly suited to every air conditioning application. For detailed information get in touch with the nearest district office or write direct to The American Rolling Mill Company, 2871 Curtis St., Middletown, Ohio.

ARMCO Ingot Iron
(Galvanized)

Ducts
Washer Chambers
Plenum Chambers
Steam Line Casings
Spray Towers
Drip Pans
Housings
Machine Guards
Unit Conditioners
Roof Ventilators
Eliminator Blades

ARMCO PAINTGRIP
(Galvanized)

A special mill-Bonderized galvanized sheet that can be painted without pretreatment. Preserves life and beauty of paint and enamel.

ARMCO Cold Rolled PAINTGRIP: A zinc-flashed sheet with a special mill-Bonderized surface that takes and holds paint.

Hot Rolled
(Sheets and Strip)

Fan Blades
Blower Casings
Fuel Oil Tanks
Unit Conditioners
Stoker Hoppers

ARMCO ZINCGRIP

A special zinc-coated sheet that can be severely formed without peeling or flaking of the tightly adherent zinc coating. Also available with a PAINTGRIP finish.

Cold Rolled
(Sheets and Strip)

Furnace Casings
Room Unit Casings

Plates
(ARMCO Ingot Iron)

Smoke Stacks
Coal Hoppers
Breeching
Unfired Pressure Vessels
Low-fired Boilers
Tanks

High Strength Steels

Low alloy, high tensile steels possessing great strength. Used with proper design ARMCO 50Y and 55Y assure lighter framework, lighter tanks and similar items. Under atmospheric service conditions it has corrosion resistance superior to regular steel

Stainless Steel
(Sheets, Strip and Plate)

Combustion Chambers
Heat Flues and Tubes
Humidifier Pans
Pre-heaters
Furnace Parts and Supports
Fan and Blower Blades

Special grades have excellent resistance to destructive heat-scaling up to 2000° F

ARMCO ALUMINIZED Steel

An aluminum-coated sheet steel with exceptional resistance to heat and corrosion. Resists heat discoloration up to approximately 900° F., and will withstand oxidation at even higher temperatures. Also offers exceptional heat reflectivity where this is desirable.

BETHLEHEM STEEL COMPANY

GENERAL OFFICES: BETHLEHEM, PA.

District Offices

Akron	Cleveland	New Haven	Savannah
Albany	Columbus	New Orleans	Springfield, Mass.
Atlanta	Dallas	New York	Syracuse
Baltimore	Detroit	Philadelphia	Toledo
Boston	Houston	Pittsburgh	Tulsa
Buffalo	Indianapolis	St. Louis	Washington
Chattanooga	Johnstown, Pa.	St. Paul	Wilkes-Barre
Chicago	Kansas City, Mo.	San Antonio	York
Cincinnati	Milwaukee		

BETHLEHEM PACIFIC COAST STEEL CORPORATION

General Offices: San Francisco

Seattle Portland Los Angeles Salt Lake City Honolulu

Export Distributors

BETHLEHEM STEEL EXPORT CORPORATION, NEW YORK

Bethlehem Products for Heating, Ventilating and Air Conditioning

STEEL SHEETS

Bethlehem manufactures a full line of sheet steel to handle all types of heating, ventilating and air-conditioning jobs. Sheets are produced in the general classifications of galvanized, hot-rolled (black); and cold-rolled. Galvanized sheets are made in gages No. 8 to 31 inclusive in widths of 24, 30, 36, 42 and 48 in. Hot-rolled sheets are made in thicknesses from No. 2 gage to No. 18 gage, in widths varying with the thicknesses. Cold-rolled sheets are made in gages No. 11 to 30 inclusive.

Bethlehem also produces galvanized steel roofing, siding and accessories. The designs include 2½ in. and 1¼ in. corrugated; 2-, 3- and 5-V crimp; Stormproof (non-siphoning drain-type); roll roofing and "plain brick" or "rock-faced stone siding". For extra rust-resistance, use Beth-Cu-Loy Sheets, available in black, galvanized and formed sheets. Containing 0.20 to 0.30 per cent copper, they have two to three times as much rust resistance as mild carbon sheets.

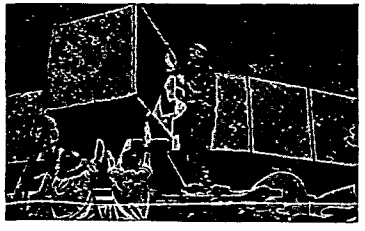
STEEL PIPE

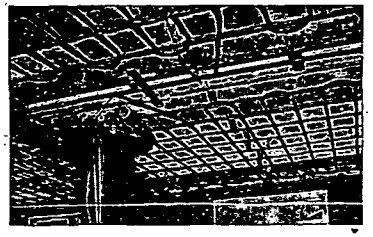
Ammonoduct—This pipe is ideal for radiant heating or refrigeration uses. It is made by Bethlehem, in sizes from ½ in. to 3 in., inclusive, and in uniform 21 ft lengths, plus or minus 1 in.

Made by the continuous-weld process from a low-carbon open-hearth steel, Ammonoduct is well suited for radiant-heating or refrigeration installations because of its bending qualities and ease of welding. It is so ductile that it can be cold-bent and cold-formed without fracture of the metal wall. Annealing costs are eliminated.

Beth-Co-Weld—A quality steel pipe manufactured by Bethlehem for use in general piping installations and in conventional heating systems. Like Ammonoduct, it is made by the continuous-weld process, and is furnished in sizes from ½ in. to 3 in., inclusive, and in uniform 21 ft lengths, plus or minus 1 in.

Bethlehem also produces lap-weld pipe in sizes 2 in. to 16 in., inclusive.





Jones & Laughlin Steel Corporation

Jones & Laughlin Building, **Pittsburgh 30, Pa.**

WELDED and SEAMLESS STEEL TUBULAR PRODUCTS HOT AND COLD ROLLED SHEETS

J & L Welded Pipe

Jones & Laughlin manufactures Standard Weight, Extra Strong, and Double Extra Strong Welded Pipe, Black and Galvanized, for steam, gas, air, water, refrigeration and sprinkler work. Sizes: ⅛ in. to 10 in. inclusive.

Jones & Laughlin Steel Pipe is made of soft, weldable steel rolled from solid ingots made to a special analysis. The steel pipe produced is soft and ductile, free cutting, strong at the welds, and free from excess scale. J & L Pipe is commercially straight and free from blisters, cracks or other injurious defects.

Careful attention is given the threading of the pipe with good clean-cut threads fitted with sound couplings correctly tapped to give a tight joint. Soft, ductile steel of free cutting quality enables the contractor to cut clean, sound threads on the job.

The Jones & Laughlin process of galvanizing assures a thorough coating and insures against pipe being clogged with spelter. The galvanized coating adheres strongly and does not tend to flake off.

J & L Seamless Pipe

J & L Seamless Pipe is made in three weights; standard and extra strong in sizes 2⅜ in. O. D. to 14 in. O. D. inclusive, and double extra strong in sizes 2 in. to 8 in. inclusive.

J & L Seamless Steel Pipe is pierced from a solid billet—there are no welds. The result is dependable and uniform wall strength. The method of manufacture, and the use of only specially selected steel, assure exceptional ductility, a quality that is essential to successful coiling and bending, and flanging for Van Stone joints.

J & L Seamless Pipe can be used with full satisfaction in either threaded joint or completely welded installations.

Ductility, strength and safety—make this product especially adaptable for air, steam, gas and gasoline lines, boilers, refineries, dry kilns, refrigerating systems and other exacting applications.

J & L Electricweld Tubes

Mechanical Tubes ⅝ in. O.D. to 4 in. O.D. inclusive.

A product of superior quality resulting from a combination of the first grade steel strip, hot and cold rolled, together with skilled production crews and supervision by men of many years' experience in producing high grade electricweld tubes.

Other J & L Tubular Products

J & L also manufactures Reamed and Drifted Pipe in sizes 1 in. to 12 in. inclusive, Dry Kiln Pipe, Pipe for Refrigeration Service, Line Pipe and a complete line of Oil Country Tubular Products in welded and seamless.

Hot Rolled Sheets

J & L Hot Rolled Sheets are made in a full range of plain carbon steel grades to a maximum of about .25 per cent carbon and including copper-bearing steel. Furnished in cut sheets in full lists of gages and widths or coils in 10 gage and lighter in over 24 in. to 90 in. widths. Also available pickled and oiled, annealed, patent or stretcher-leveled and resquared on application.

Cold Rolled Sheets

J & L Bright Cold Rolled Sheets in a full range of plain carbon steel grades including copper-bearing steel furnished in cut lengths or coils in the following gages and maximum widths—11 to 13 gage over 24 in. to 84 in. width; 14 to 20 gage over 24 in. to 90 in. widths; 21 and 22 gage over 24 in. to 72 in. widths; 23 to 24 gage over 24 in. to 60 in. widths.

United States Steel Corporation Subsidiaries

Carnegie-Illinois Steel Corporation, Pittsburgh and Chicago
Columbia Steel Company, San Francisco
Tennessee Coal, Iron & Railroad Company, Birmingham
United States Steel Export Company, New York
District Offices in all Principal Cities

U·S·S COPPER STEEL

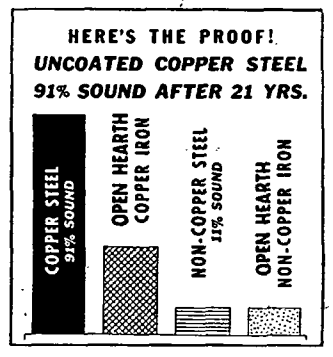
For Superior Rust Resistance at Low Cost

Corrosion resistance and cost are two determining factors of the type of metal to be used for various air conditioning jobs.

Copper Steel has 2 to 3 times the atmospheric corrosion resistance of plain steel or pure iron as shown in the results of unbiased tests made at Pittsburgh, Ft. Sheridan and Annapolis by the American Society for Testing Materials.

The cost of U·S·S Copper Steel is less than that of pure iron or copper-bearing pure iron and only slightly more than plain steel. Thus there often is a dividend of 200 per cent to 300 per cent longer life and a saving in the first cost as well.

When galvanized, U·S·S Copper Steel produces a sheet that is rust resistant all the way through—not just on the surface. It should be used for all ducts carrying humidified air or placed in damp locations such as basements, shower rooms, etc.



Corrosion test of A.S.T.M. on 22 gage black sheets exposed at Annapolis, Md., October, 1916. The copper steel sheets outlasted all others in the test.

U·S·S PAINTBOND

U·S·S PaintBond should be used whenever galvanized steel is to be painted. This special Bonderized sheet can be painted immediately, offers a much better surface for painting, lessens danger of the paint flaking and retards corrosion. It is used for ductwork, furnace housings and outdoor metal work.

Send for our PaintBond booklet.

U·S·S DUL-KOTE

U·S·S Dul-Kote is a specially treated non-spangled galvanized sheet which also can be painted immediately without aging or otherwise preparing the surface. It is available in the South and in the West.

Send for our Dul-Kote booklet.

OTHER U·S·S PRODUCTS INCLUDE:

Black Sheets—All grades, hot rolled, cold rolled in a number of different finishes.

Stainless—Heat resisting steel for various uses where temperatures are high and corrosion severe.

Cor-Ten—High strength steel—greater strength, greater atmospheric corrosion resistance for smokestacks, hoods, etc.

For more information on U·S·S Galvanized, Copper, Black and Stainless Sheets, send for our Guide for Sheet Metal Workers.

Atlantic Metal Hose Co., Inc.

112 West 64th St. New York 23, N. Y.

ATLANTIC SEAMLESS FLEXIBLE METAL HOSE

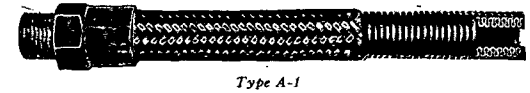
ABSORBS VIBRATION

on REFRIGERATING and AIR CONDITIONING EQUIPMENT

For standard pressures in a variety of metals

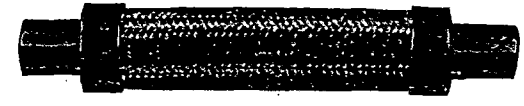
Type A

For high pressures and severe service



Type A-1

For refrigeration and air conditioning Freon Lines



Vibration Absorber

Throughout the Heating, Ventilating and Air Conditioning Industry—where the ability to control vibration and compensate for expansion must go hand in hand with the ability to take punishment and withstand pressure—Atlantic hose continues to break performance and economy records year after year.

Durable—the specially selected alloy and method of construction assures long life even under severest service.

Simplifies Installations—Both slip-on as well as pipe thread fittings can be provided in all commercial sizes for quick assembly.

We also manufacture hose for the conveyance of dust, air, exhaust, steam and all liquids.

Engineering Service—Our engineering department will cooperate with prospective users in the selection of suitable standard flexible hose, or in the design of hose to meet special requirements. Advise us of the details of the service and we will be pleased to offer our recommendation.

Chicago Metal Hose Corporation

Maywood, Illinois

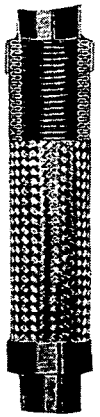
PLANTS

MAYWOOD and ELGIN, ILLINOIS

Distributors

HOUSTON, TEXAS.....Peden Iron & Steel, Inc.
LOS ANGELES, CALIFORNIA.....Los Angeles Rubber & Asbestos Works
NEW ORLEANS, LOUISIANA.....Standard Supply & Hardware Company
PORTLAND, OREGON.....Marshall-Wells Company
SAN FRANCISCO, CALIFORNIA.....C. W. Marwedel
SEATTLE, WASHINGTON.....Strehlow Supply Company

Rex Vibra-Sorbers Control Vibration and Noise



Rex Vibra-Sorbers are used to control vibration and reduce noise in refrigeration and air conditioning machinery. They are also available with male pipe thread fittings for use in connecting lines of pumps, turbines, and allied equipment.

Rex Vibra-Sorbers are pressure-tested to insure refrigerant-tightness, thus giving greater protection against leakage of valuable refrigerants. They are available in copper bearing alloy for Freon and Menthol, and steel for ammonia. Thus effective corrosion resistance is provided for all types of installations. C.M.H. engineers can help you solve your vibration problems. Write for information.

Specifications and List Prices

COPPER TUBE SIZE		I.D. Flexible Hose	Length Flexible Hose	Bursting Pressures Lbs. per Sq. in.	OVERALL LENGTHS		*STOCK NUMBERS		UNIT LIST PRICES
Actual O.D.	Nominal I.D.				Standard Units with Female Sweat Fittings	Standard Units with Male Sweat Fittings	With Female Sweat Couplings	With Male Sweat Couplings	
¼"	⅛"	3/16"	5¼"	3,000+	7"	7½"	F-1	M-2	$ 2.90
¼"	⅛"	¼"	5¾"	3,000+	7½"	8⅛"	F-3	M-4	3.00
⅜"	¼"	5/16"	6¼"	3,000+	8⅛"	9"	F-5	M-6	3.30
½"	⅜"	⅜"	6⅝"	2,500+	9"	9⅝"	F-7	M-8	3.60
⅝"	½"	½"	7"	2,500+	9⅝"	10⅜"	F-9	M-10	4.10
⅝"	⅜"	½"	7"	2,500+	10"	11"	F-11	M-12	4.10
⅝"	⅜"	⅝"	7¾"	2,000+	11⅛"	12⅛"	F-13	M-14	5.60
⅝"	⅜"	⅝"	7¾"	2,000+	11½"	12½"	F-15	M-16	5.60
1⅛"	1"	1"	8½"	1,500+	13"	14"	F-17	M-18	7.00
1⅜"	1¼"	1⅜"	9¾"	1,500+	14¾"	15¾"	F-19	M-20	10.00
1⅜"	1¼"	1⅜"	11"	1,500+	17"	17½"	F-21	M-22	13.00
2⅛"	2"	2"	13"	1,000+	20"	20"	F-23	M-24	18.00
2⅛"	2⅛"	2⅛"	15¾"	1,000+	24"	24"	F-25	M-26	36.00
3⅛"	3"	3"	17¾"	1,000+	27"	27"	F-27	M-28	46.00
3⅛"	3⅛"	3⅛"	21½"	1,000+	31½"	31½"	F-29	M-30	90.00
4⅛"	4"	4"	21½"	1,000+	32"	32"	F-31	M-32	120.00

* For convenience you need specify these numbers only when ordering.

NOTE: Flexible hose lengths listed above are for straight line installations only. Information on lengths for angle installations requiring bends and special units with elbows or threaded fittings is available on request.

All information on Rex Vibra-Sorbers is based on actual measurements and tests. However, due to manufacturing limitations and structural changes, this data must be regarded only as a general guide.

Seamlex Company, Inc.

Main Office and Factory

27-31 Jackson Ave., Long Island City 1, N. Y.

Since 1928—Specialists in the Manufacture of Flexible SEAMLESS All-Metal Hose

You are invited to submit your flexible tube problems to our engineers—without obligation on your part.

You May Obtain Catalog No. 45 Free of Charge

VIBRATION ELIMINATOR

Fig. 700
Seamlex with Sweat Fitting

FOR DOMESTIC OIL BURNERS

Fig. 32
Seamlex with Type "M" Fitting

FOR INDUSTRIAL OIL BURNERS

Fig. 57
Seamlex with Type "SBP" Fitting

FOR UNIVERSAL APPLICATIONS

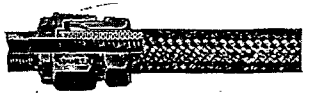


Fig. 60
Seamlex with "DT" Fitting

FOR DIESEL EXHAUST from 2 " I.D. to 24 " I.D.

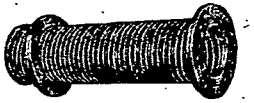


Fig. 90
Seamlex Welded Steel Hose

SEAMLEX

Bronze—Single Braided

STANDARD SIZES

I.D.	O.D.	Minimum Bending Radii for Flexing	Bursting Pressures Lbs./Sq. In.
1/16"	¼"	4"	10,000
⅛"	11/32"	5"	7,500
3/16"	13/32"	6"	5,000
¼"	11/16"	6"	4,000
5/16"	⅝"	6"	3,500
⅜"	¾"	7"	3,000
½"	13/16"	8"	2,500
¾"	1⅛"	9"	2,000
1"	1¼"	10"	1,750
1¼"	2⅛"	11"	1,500
1½"	2⅝"	12"	1,250
2"	3⅛"	13"	1,000
2½"	3⅞"	18"	800
3"	4⅜"	18"	675
3½"	4⅞"	22"	600
4"	5⅞"	24"	500
4½"	6"	26"	480
5"	6⅝"	28"	440
6"	8"	30"	400

SEAMLEX Products Are Quality Products

The American Brass Company

General Offices: Waterbury 88, Conn.

Offices and Agencies in Principal Cities

ANACONDA
from mine to consumer

IN CANADA: ANACONDA AMERICAN BRASS LIMITED, New Toronto, Ontario

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red-Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment

ANACONDA COPPER TUBES AND FITTINGS

For Heating, Plumbing and Air Conditioning

Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Fittings offer an unusual combination of advantages in hot water heating systems at a cost only slightly higher than black iron and approximately the same as wrought iron pipe. These advantages may briefly be summarized as follows:

Low Friction Loss—Because the inside surfaces of copper tubes are inherently smoother than those of pipe and tubes made of ferrous materials and also because they do not become roughened by the formation of rust, these tubes offer a lower resistance to flow. In addition, the long radius turns of Anaconda Elbows and the smooth inside surface of Anaconda Wrought Copper Fittings further reduce friction losses.

These factors naturally increase the efficiency of the system, particularly when it includes a forced pressure circulator.

Ease of Installation—In many places the flexibility of copper tubes simplifies connections that ordinarily would be awkward and expensive to make with rigid pipe and threaded fittings. Anaconda Solder Fittings are compact. They can be installed in restricted space where the use of a wrench would be impossible.

Architects and builders naturally object to large holes and notches cut in the framing members of a building for the passage of piping. Anaconda Copper Tubes can be installed with a minimum of cutting in the structure—although holes should be large enough to permit movement of tubes due to expansion and contraction.

Appearance—Anaconda Deoxidized Copper Water Tubes assembled with Anaconda Solder Fittings present an attractive appearance. It is a frequent practice to clean the tubes after they are installed and apply a coat of clear lacquer or similar substance. This keeps the tubes bright and makes an installation of which both plumber and owner can be proud.

Temper and Thicknesses—Anaconda Copper Tubes are made in both hard and soft temper and in standard wall thicknesses.

They meet the requirements for these types of tubes in Federal Specification WW-T-799a and *A.S.T.M.* Specification B88. Type K, the heaviest, is recommended for heating lines and general piping.

Accuracy of Dimensions—Anaconda Deoxidized Copper Water Tubes are all finished to the close size tolerances required by the *A.S.T.M.* and Federal Specifications, which have been found essential for efficient assembly with solder fittings.

Permanent Identification—For permanent identification, the name "Anaconda" and the letter designating the type of tube is stamped in the metal at intervals of approximately 18 in., throughout every coil or straight length of tube.

The American Brass Company

Anaconda Copper Tubes, in all standard sizes, up to and including 1¼ in. are furnished soft in 30, 45 and 60-ft coils; also hard and soft in 20-ft straight lengths. Sizes over 1¼ in. are furnished, hard or soft, in straight lengths only.

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 recommended for steam return lines.

Anaconda "85" Red Brass Pipe contains 85 per cent copper and conforms to Government specifications for Grade "A" water pipe. The mark "Anaconda 85" is stamped in the metal at one-foot intervals throughout each length.

EVERDUR*

Everdur Metal is the original copper-silicon alloy. It is manufactured by The American Brass Company in five standard compositions and in practically all commercial forms.

This high strength engineering metal is resistant to a wide range of corroding agents. Because of a versatile combination of useful properties, Everdur has become standard as a material for equipment in many fields of engineering and industry.

In addition to their non-rusting 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 oxy-acetylene or carbon arc welding.

CORROSION RESISTANCE

The corrosion resistance of Everdur is equal to that of pure copper and in some cases, slightly superior.

*"Everdur" is a trademark of The American Brass Company registered at the U. S. Patent Office.

However, like copper and all copper alloys, Everdur is not equally resistant to all corroding agents, nor to the same corroding agents under all conditions. As with copper, the resistance to corrosion may be substantially reduced in some instances by the presence of oxidizing agents. Nevertheless, Everdur does offer excellent resistance to the corrosive action of many solutions and atmospheres.

Everdur Tanks—Everdur copper-silicon alloy is an ideal material for durable, rustless 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 *A.S.T.M.* Specification B96.

Minimum specification requirements for hot rolled-and-annealed tank plates are: Tensile Strength, 50,000 psi.; Yield Strength (at 0.5 per cent elongation under load) 18,000 psi.; Elongation, 40 per cent in 2 inches.

Welds made with annealed Everdur tank plates meet the requirements for Class B and C vessels in the *A.S.M.E.* Code for Unfired Pressure Vessels.

For additional data and names of fabricators address our nearest office or agency.

EVERDUR FOR AIR CONDITIONING EQUIPMENT

Because of its strength and welding properties, Everdur may be substituted for steel and fabricated by substantially the same methods and with much the same equipment as steel.

Everdur metal has been used with marked success for fans and 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 upon request.

Mueller Brass Co.

Port Huron, Mich.

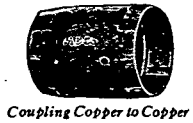
Branch Offices and Representatives in Principal Cities

Albany, N. Y.	Cleveland, Ohio	Harrisburg, Pa.	Philadelphia, Pa.
Atlanta, Ga.	St. Louis, Mo.	Indianapolis, Ind.	N. S. Pittsburgh, Pa.
Boston, Mass.	Dallas, Texas	Los Angeles, Calif.	San Francisco, Calif.
Chicago, Ill.	Detroit, Mich.	Minneapolis, Minn.	Seattle, Wash.
Cincinnati, Ohio	Flint, Mich.	Newark, N. J.	Washington, D. C.

Canadian Sales and Manufacturer

Canada Wire and Cable Co., Ltd., Toronto, Canada

PRODUCTS—STREAMLINE Copper Pipe and Seamless Tubes; STREAMLINE Hard Copper Pipe and Solder Fittings; Valves, Flared and STREAMLINE Solder Fittings for Mechanical Refrigeration; Forgings of Brass, Bronze and Copper; Castings of Brass and Bronze; Rod; Screw Machine Products; Fabricated Parts and Special Nickel and Chromium Plated Parts; Machined Formed Tubes.



Coupling Copper to Copper

Copper to Outside I.P.S.

45 Deg Elbow

Streamline Copper Pipe and Fittings for heating, plumbing, air conditioning and industrial use are made by the Streamline Pipe and Fittings Division, Mueller Brass Co., Port Huron, Mich.

The **Streamline** Solder Fitting is the original solder type fitting, introduced and manufactured by the Mueller Brass Co. of Port Huron, Mich. It incorporates many advantageous features and has proved to be the revolutionary advance of the age in the development of piping systems for plumbing and heating and for many industrial uses.

The **Streamline** Solder Fitting is not connected either by threading or flaring, but by soldering. The outside surface of the copper pipe and the inner surface of the **Streamline** fitting are cleaned with sandcloth, and solder flux is then applied to the cleaned surfaces to eliminate oxidation when the assembled joint is heated. The joint is then sufficiently heated with a blow or acetylene torch and the soldering operation is performed by feeding wire or stick solder through the feed hole in the fitting.

The **Streamline** Solder Fitting alone has the solder feed hole, groove and taper. The solder feed hole, through which the solder is introduced, enters directly into an internal feed channel. The feed channel is located equidistantly between the internal shoulder against which the pipe rests and the outer edge of the fitting. When solder is introduced it is distributed by capillarity from the feed channel and distributed evenly and thoroughly between the bonding surfaces, traveling inward to the shoulder and outward to the edge of the fitting where it appears as a continuous solder ring around the full circumference of the pipe. This ring, and feed hole completely filled with solder, constitute positive proof to the operator that the joint is permanently leak-proof. An actual pressure test is not necessary.

Streamline Pipe and Fittings Division

MUELLER BRASS CO.

Port Huron, Mich.

Patents 1770852; 1776502

Tee

Tee
Copper to Inside I.P.S.

90 Deg Elbow

Crosses

The solder may be fed from any position, whether the feed hole is located at the top, side or bottom. Owing to the never failing phenomena of capillarity, the solder will flow up, down or laterally with equal facility.

Streamline Copper Pipe is a seamless cold drawn copper tubing conforming to *A.S.T.M.* B88. It is made in sizes ¼ in. to 12 in. and Types K, L, and M, of which Type K is the heaviest. The intermediate weight, Type L, is the preferred weight for plumbing, heating and refrigeration.

For most purposes hard drawn pipe is used, though Types K and L can be furnished annealed when bending is required. Annealed Type "K" in sizes up to 2 in. is widely used for underground water "services".

An additional lighter weight in sizes 3 to 12 in. inclusive is used by paper mills for water and stock lines at pressure up to 100 lb.

Streamline Solder Fittings are furnished in all sizes to 6 in. inclusive. They are of the same thickness as Navy and MSS Fittings for 125 steam or 175 lb. non-shock water pressure.

All fittings over 6 in. are flanged and may be had with either *A.S.A.* or riveted pipe standard flanges.

Mating flanges are soldered to the pipe—*A.S.A.* 125 standard flanges are available from 1 in. up.

During the last fifteen years architects and engineers have used **Streamline** Copper Pipe and Fittings successfully in every type of building construction and in thousands of installations throughout the United States and Canada.

In addition to its rust and vibration-proof qualities and long life, **Streamline** has many other advantages such as the reduction in size of pipe lines and radiator connections from those nominally used, a neat, compact installation requiring a minimum of space and important advantages in industrial and drainage applications. There is a **Streamline** product for every piping requirement.

Write Mueller Brass Co., Port Huron, Michigan, for complete information and catalog.

Revere Copper and Brass Incorporated

Executive Office: 230 Park Avenue, **New York 17, N. Y.**

MILLS—Baltimore, Md., New Bedford, Mass., Rome, N. Y., Detroit, Mich., Chicago, Ill.

SALES OFFICES—Boston, Mass., Providence, R. I., Philadelphia, Pa., Atlanta, Ga., New York, N. Y., Pittsburgh, Pa., Cleveland, Ohio, Cincinnati, Ohio, Grand Rapids, Mich., Milwaukee, Wis., St. Louis, Mo., Indianapolis, Ind., Minneapolis, Minn., Dallas, Texas, Seattle, Wash., San Francisco, Calif., Los Angeles, Calif., Hartford, Conn., Dayton, Ohio, Houston, Texas

REVERE COPPER TUBE

Revere Copper Water Tube is produced under close inspection and meets all Federal and *A.S.T.M.* specifications. It is cold drawn, seamless, deoxidized, and possesses a gunbarrel finish on the inside as well as the outside. A wide range of standard sizes made in hard and soft tempers is normally stocked by distributors.

Types K, L, and **M** are made in the sizes, tempers, and lengths shown below. **Type K** is a heavy wall tube commonly used for underground water service lines, for plumbing and heating, oil lines, and some air conditioning applications. **Type L** is a medium wall tube furnished in hard or soft temper, and **Type M** is a light wall tube furnished in hard temper only. The latter is not recommended for underground service or for compression fittings. The actual outside diameter of these three types of tube is ⅛ in. greater than the nominal size in each case.

Revere Dryseal Copper Tube is a dependable dehydrated utility tube furnished only in soft temper coils having both ends sealed. There are nine sizes available from ⅛ in. to ¾ in. inclusive. In each case the nominal size and the actual outside diameter are the same and the wall thickness is 0.035 in. throughout.

Revere Dryseal Copper Tube is commonly used for air conditioning and heat control lines, refrigeration and oil burner installations, and compressed air lines. It can be bent by hand and easily flared for compression fittings.

Fittings—Standard solder type fittings in cast or wrought patterns are available for use with **Types K, L,** and **M** copper tube. Various types of compression fittings are made for use with **Types K** and **L** tube and **Revere Dryseal Tube.** Connections made by the use of these fittings are proof against leaks so often caused by stresses or vibration. Joints of this kind are quickly made without any thread cutting and are generally stronger than the tube itself.

Advantages of Revere Copper Water Tube—The many advantages of **Revere Copper Water Tube** of particular interest to design engineers can be summarized briefly as follows:

1. A piping system comprising **Revere Copper Water Tube** and soldered fittings can be installed in a minimum of space as no allowance need be made for swinging wrenches.

2. Soft temper copper tube can be bent easily for offsets and changes in direction.

3. There need be no fear of rust accumulation on the inside of **Revere Copper Water Tube.**

4. It is particularly suitable for forced circulation hot water heating systems. The very smooth interior of **Revere Copper Water Tube** insures high velocities with a minimum of frictional resistance.

Revere Copper Water Tube for Radiant Panel Heating—A practical manual outlining a graphical design procedure for Radiant Panel Heating installations is now available to Engineers and Architects on request.

To Engineers and Manufacturers of Heating and Air Conditioning Equipment—Revere representatives will gladly assist and cooperate with engineers and suppliers of heating and air conditioning equipment at any time regarding matters involving applications of non-ferrous products.

Revere Copper Water Tube

STANDARD DIMENSIONS AND WEIGHTS

Size In In.	Type K			Type L		Type M	
	O.D. in In.	Wall Thickness In.	Wt. Lb per Ft	Wall Thickness In.	Wt. Lb per Ft	Wall Thickness In.	Wt. Lb per Ft
¼	.375	.032	.134	.030	.126	.025	.107
⅜	.500	.049	.269	.035	.198	.025	.145
½	.625	.049	.344	.040	.285	.028	.204
⅝	.750	.049	.418	.042	.362	.030	.263
¾	.875	.065	.641	.045	.455	.032	.328
1	1.125	.065	.839	.050	.655	.035	.465
1¼	1.375	.065	1.04	.055	.884	.042	.682
1½	1.625	.072	1.36	.060	1.14	.049	.940
2	2.125	.083	2.06	.070	1.75	0.58	1.46
2½	2.625	.095	2.93	.080	2.48	0.65	2.03
3	3.125	.109	4.00	.090	3.33	.072	2.68
3½	3.625	.120	5.12	.100	4.29	.083	3.58
4	4.125	.134	6.51	.110	5.38	.095	4.66
5	5.125	.160	9.67	.125	7.61	.109	6.66
6	6.125	.192	13.9	.140	10.2	.122	8.92

Recommended Operating Pressures

Type K—Hard Temper.....up to 400 lb
Type K—Soft Temper.....up to 250 lb
Type L—Hard Temper.....up to 250 lb
Type L—Soft Temper.....up to 150 lb
Type M—Hard Temper.....up to 250 lb

Tempers and Lengths

Type K } Hard and Soft Temper in straight 20 ft lengths.
Type L } Soft Temper 1 in. and under in 60 ft coils.
Type M—Hard Temper in straight 20 ft lengths.

Wolverine Tube Division

Calumet & Hecla Consolidated Copper Company

1411 Central Avenue, **Detroit 9, Michigan**

SEAMLESS TUBE

COPPER - BRASS - ALUMINUM

Sales Offices:

Atlanta, Ga.....P. O. Box 38—Northside Station
Buffalo, N. Y.....416 Jackson Bldg.
Chicago, Ill.....3348 S. Pulaski St.
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Dayton, Ohio.....P. O. Box 1041
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Commercial Trust Bldg.—Room 1208
15th and Market Sts.
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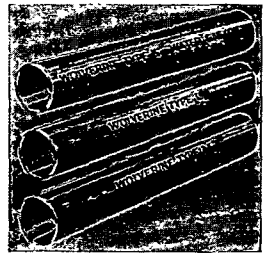
St. Louis, Mo.....4565 McRae Ave.
Wichita Falls, Texas.....P. O. Box 1030

COPPER WATER TUBE



TYPE K—Recommended for Air Conditioning, Refrigeration, Oil Burner, and Plumbing and Heating installations.

TYPE L—For Oil Burner, Air Conditioning, Refrigeration and general plumbing uses.

TYPE M—Suitable for Air Conditioning and Refrigeration installations and for interior plumbing and heating purposes.

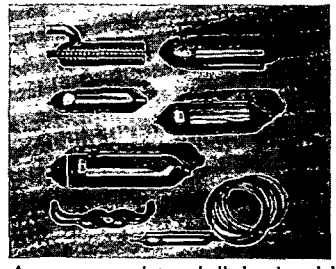
Types K and L furnished in hard or soft temper; Type M, hard only.

Wolverine Water Tube is made according to U. S. Government and A.S.T.M. specifications. For a complete list of these data, write Detroit for Form 575.

RÉFRIGERATION TUBE

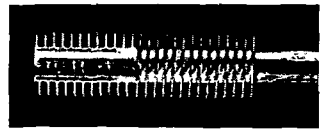
Wolverine refrigeration tube has long been the standard of the industry. Dehydrated, sealed, paper-wrapped; uniform soft temper, and moisture content well below minimum specified by A.S.R.E. Available from stock in standard coils.

ACCUMULATOR SHELLS



A new accumulator shell developed by Wolverine and produced to customers' specifications in a variety of shapes and sizes up to 3½ in. diameter.

It combines many advantages including one-piece construction and is especially adaptable to refrigeration problems. Send for Catalog E-1 describing other spun-end tubular parts.



TRUFIN, The Integral Finned Tube originated and developed by Wolverine, is one of the most impressive advances in the entire heat transfer field.

TRUFIN is suitable for nearly all kinds of heat transfer work—heaters, coolers, interchangers, condensers, and many other applications.

Send for "**TRUFIN**" Engineering Pamphlets 1, 2 and 3.

The experience of 29 years of seamless tube manufacture, the use of the latest equipment, and adherence to Government and customer specifications, are responsible for the uniform, high quality of Wolverine products. And now, backed by the 78-year experience and large resources of Calumet & Hecla, Wolverine quality is controlled from ore to finished product.

E. B. Badger & Sons Co.

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PRODUCTS AND SERVICES

BADGER "PACKLESS" CORRUGATED EXPANSION JOINTS

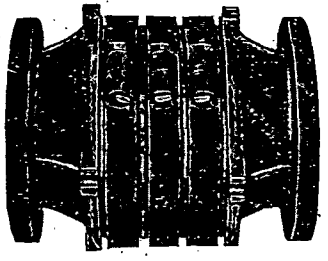
Engineers and Manufacturers of Chemical, Petro-Chemical and Petroleum Refining Equipment; Process Engineers, Designers and Constructors of Complete Plants.

BADGER "PACKLESS" CORRUGATED EXPANSION JOINTS

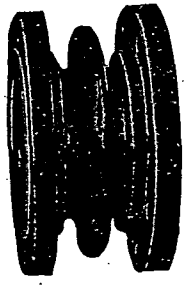
Protect equipment; eliminate costly shut-downs; prevent loss of materials flowing through pipe lines. The Badger "Packless" Corrugated type of expansion joint has been used for many years to relieve stresses, absorb vibrations between connected equipment, and compensate for changes in pipe lines due to temperature differentials.

BACKED BY FIFTY YEARS OF EXPERIENCE

Badger engineers pioneered the corrugated "packless" joint and brought it to its present high state of dependability and usefulness. To meet the increasingly severe conditions in industry, the more recent developments are included in the following brief descriptions.



Standard Flanged-End Badger Directed Flexing, Self-Equalizing "Packless" Corrugated Expansion Joint. *Also made with standard welding end. (Double units available.) Bulletin No. 100.*



Badger Non-Equalizing "Packless" Corrugated Expansion Joint, Single Corrugation. *Also made in multiple corrugation and special shapes. Bulletin No. 200.*

FLANGED AND WELDING ENDS. Badger "Packless" Corrugated Expansion Joints can be furnished with either flanged or welding ends.

TELESCOPING SLEEVES. Badger "Packless" Corrugated Expansion Joints can be furnished with telescoping sleeves if required.

INSTALLATION, OPERATING, MAINTENANCE ECONOMIES

• ***"Packless"***—Since this type of expansion joint is made from a *single tube*, no packing is required—hence no servicing. This is particularly advantageous in underground installations, because the expense of manholes and tunnels can be eliminated.

• ***Flexible***—This type of joint is readily flexed, thus reducing to a minimum the thrust on adjacent equipment or fittings.

• ***Wide Range of Traverses***—By varying the number of corrugations, the Badger "Packless" Corrugated Expansion Joint can be made to take care of traverses ranging from small fractions of an inch up to any practical limit.

• ***Wide Range of Pressures***—Standard joints are constructed for normal operating pressures; special joints for higher pressures.

• ***Compactness***—Outside diameter of Badger "Packless" Corrugated Expansion Joint is about that of a flanged fitting.

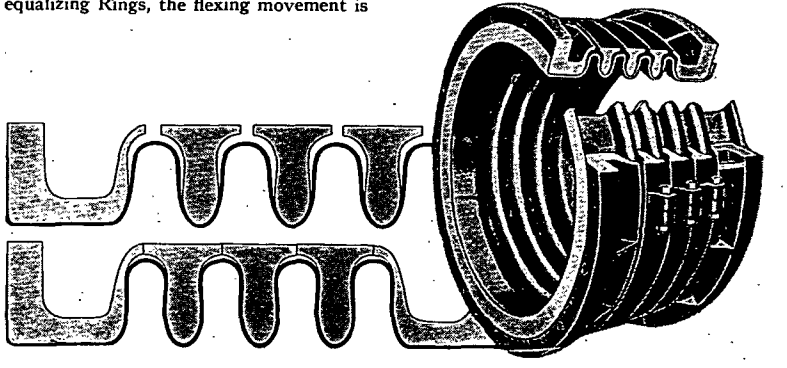
• ***Ease of Installation***—The Badger "Packless" Corrugated Expansion Joint is as easily installed as any fitting.

COPPER AND STAINLESS STEEL. Badger "Packless" Corrugated Expansion Joints are made in self-equalizing or non-equalizing types; and are made of copper, stainless steel, or other metals to meet varying conditions with respect to temperature, pressure and corrosion.

HEAT TREATMENT. Every Badger "Packless" Corrugated Expansion Joint is scientifically heat-treated during the process of manufacture to remove forming stresses. This Badger process increases resistance to flexing stresses, lengthening the life of joint.

DIRECTED FLEXING—SELF-EQUALIZING

This feature, *exclusive* with Badger "Packless" Corrugated Expansion Joints, greatly lengthens the life of the corrugated type of expansion joint. Through Self-equalizing Rings, the flexing movement is not only limited but is progressively controlled throughout the movement. Flexing stresses are kept distributed.



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Complete Service on materials to Specification on Power Plant Piping, Industrial Piping, and Industrial Heating Systems; Prefabricated Piping including Pipe Cutting and Threading, Pipe Bends, Welded Headers, Welded and Welding Fittings, Lap Joints and the Grinnell line of products for Super Power.

Grinnell Equiflo Valves for forced hot water heating systems; Grinnell Adjustable Pipe Hangers and Supports; Grinnell Cast Iron and Malleable Iron Pipe Fittings; Grinnell Flared Tube Fittings; Grinnell Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoliers (Unit Heaters); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Piping for acids and other special materials.

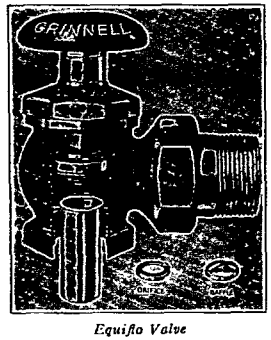
Malleable Iron, Brass, Bronze and other Castings; Brass, Cast Iron, Wrought Iron and Steel Pipe; Seamless Steel Tubing in Iron Pipe Sizes.

Valves: Check, Diaphragm, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems. This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices and baffles capable of setting up any required frictional resistance. This method of establishing any desired resistance does away with elaborate calculation of pipe sizes. Grinnell guarantees perfectly balanced circulation to each and every radiator where these valves are installed throughout the system.

Equiflo Data Book sent to interested parties.

For Data On Thermoflex Traps and Heating Specialties, see page 1211

GRINNELL ADJUSTABLE PIPE HANGERS AND SUPPORTS

One of the chief advantages of Grinnell Adjustable Hangers is that they permit adjustment of pipe lines after installation, thus obviating the necessity of turnbuckles or the removal of hangers. Their time and trouble-saving qualities during installation are equally exceptional. Below are shown a few Grinnell Hangers and Supports of particular interest to heating engineers. Send for Hanger Catalogue showing complete line.

Adjustable Swivel Rings (Patented)

Fig. No. 101. Solid Ring

These Malleable Iron Adjustable Swivel Rings can be used with Coach Screw Rod or Machine Threaded Rod in connection with practically any type of Ceiling Flange, Expansion Case, Insert, etc.

Adjustment of at least 1½ in. is secured by turning Swivel Shank. Swivel Shank automatically locks, preventing loosening due to vibration in the pipe line.

The Split Ring permits adjustment either before or after Ring is closed. A wedge type pin is loosely but inseparably cast into the hinged section for fastening this section after pipe is in place.

Fig. No. 104 Split Ring

Adjustable Swivel Pipe Rolls (Patented)

Fig. No. 174 Swivel Pipe Roll

An adjustable type of pipe roll using a single hanger rod. Swivel Shank allows vertical adjustment and automatically locks, preventing loosening from vibration.

CB-Universal Concrete Inserts (Patented)

Made of malleable iron, in one body size, to take a special removable nut, tapped for ⅜ in., ½ in., ⅝ in., ¾ in. or ⅞ in. rod as required. Nuts automatically lock by means of V-type teeth on both insert and nuts.

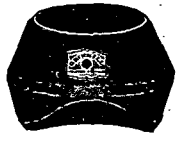
Fig. No. 282 CB-Universal Insert

GRINNELL WELDING FITTINGS

Grinnell Welding Fittings are made from Seamless Steel Pipe or tubing and possess the same physical characteristics as standard, extra strong and o.d. steel pipe or seamless steel pipe of comparable size. They can be used under the same conditions, pressures and temperatures as the pipe itself.

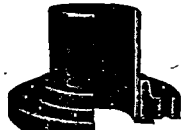
All Grinnell Welding Fittings have welding faces for all plain circumferential butt welds scarfed or beveled as follows: For wall thicknesses 3/16 to ¾ inch inclusive, 37½ deg. ± 2½ deg., straight bevel. Angles of bevel other than 37½ deg. can be furnished on special order.

90° Elbow, Long Turn



Welding Outlet

Welding Tee



Lap-Joint Stub End with Lap Flange attached

Threaded Outlet

The Dow Chemical Company

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Plastics Division

SARAN PIPE, TUBING AND FITTINGS

FOR INSTALLATIONS REQUIRING HIGH CHEMICAL RESISTANCE

These thermoplastic products made from Dow raw materials are recommended for a wide variety of applications—especially those jobs requiring high chemical resistance. (See table below.) Additional engineering data and literature are available from the following manufacturing and sales licensees—or any Dow sales office.

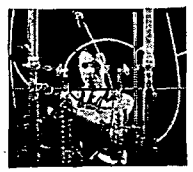
Tubing and fittings are available from: Parker Appliance Co., 17325 Euclid Ave., Cleveland, Ohio; Yardley Plastics Co., 138 Parsons Ave., Columbus, Ohio.

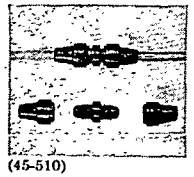
Tubing, pipe, and fittings for both are available from: American Hard Rubber Co., 11 Mercer Street, New York, New York; Commercial Plastics Co., 201 N. Wells St., Chicago, Ill.; Haveg Corporation, Newark, Del.; Elmer E. Mills Corporation, 153 W. Huron St., Chicago, Ill.; St. Louis Plastic Molding Co., 4605 Olive St., St. Louis, Mo.; Western Felt Works, 4115 W. Ogden Ave., Chicago, Ill.; Yardley Plastics Co., 138 Parsons Ave., Columbus, Ohio.

CHEMICAL RESISTANCE OF SARAN AT ROOM TEMPERATURE

Reagent	Stability Rating	Reagent	Stability Rating	Reagent	Stability Rating
98% (Conc.) H_2SO_4	Fair	50% NaOH	Good	Linseed Oil	Excellent
60% H_2SO_4	Excellent	10% NaOH	Good	Bromine Water	Not Recommended
30% H_2SO_4	Excellent	28% NH_4OH	Not Recommended	Chlorine Water	Not Recommended
10% H_2SO_4	Excellent	10% NH_4OH	Not Recommended	"Chlorax" Bleaching Solution	Excellent
35% (Conc.) HCl	Excellent	Ethyl Alcohol	Excellent	10% Duponol	Excellent
10% HCl	Excellent	Carbon Tetrachloride	Good	10% Zinc Hydrosulfite	Excellent
65% (Conc.) HNO_3	Good	Di Ethyl Ether	Good	15% Ca Cl_2	Excellent
10% HNO_3	Excellent	*Benzene	Not Recommended	15% $FESO_4$	Excellent
Glacial Acetic	Excellent	Ethyl Gasoline	Excellent	Water	Excellent
10% Acetic	Excellent	Turpentine	Excellent	Air	Excellent
5% H_2SO_3	Excellent	Triethanolamine	Excellent		
Conc. Oleic	Excellent	Lubricating Oil	Excellent		

It should be noted that chemical resistance decreases with rise in temperature.
*Shrinks slightly upon immersion in these solvents.





(45-510)

TUBING

Saran tubing accommodates both high and low pressures, withstands freezing, and is heat resistant up to 170° F. Its unusual chemical resistance gives it wide use in installations where acids and corrosive chemicals must be handled.

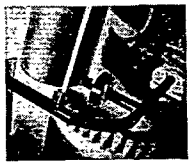
Saran tubing is readily flared with a special flaring tool, assuring a tight joint in special Saran fittings. Tees, elbows, and straight joints (illustrated) are available in sizes to match tubing (up to ¾ in. O. D.).

PIPE

Chemical resistance makes Saran pipe invaluable for many industrial installations. It is only ¼ the weight of comparable iron pipe. Available in sizes up to 4 in. O.D.

A complete weld is formed in one minute by placing on heated welding plate until molten and pressing ends together.

Easily threaded with modified standard equipment. Flanges, couplings, ells and plugs are available. Further fittings being developed.



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ENGINEERS — FABRICATORS OF NON-FERROUS METALS AND STAINLESS STEEL

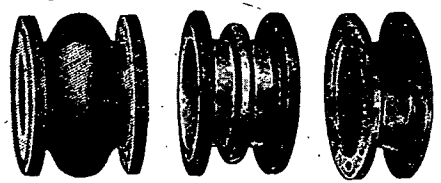
Metals Fabricated—Aluminum, Block Tin, Brass, Bronze, Copper, Everdur, Monel, Nickel, Inconel, Stainless Steel and KA2 SMO. Bulletin on request.

Metal Floats

Column *Ball* *Flat Cylindrical* *Cylindrical* *Cylindrical*

Made of copper, plain steel, stainless steel, KA2 SMO, aluminum, brass, Monel, pure nickel, Admiralty and Everdur, for open tank and all pressures.

Seamless copper ball floats carried in stock in diameters of 3 in., 4 in., 5 in., 6 in., 7 in., 8 in., 10 in., 12 in. for open tank and pressures of 25, 50, 100 and 150 lb. Floats in special sizes and pressures—made to order. Stainless steel ball floats 2½ in. to 12 in. for high pressure and corrosion carried in stock—special stainless steel floats made to order—stainless steel ball floats larger than 12 in. diameter can be made up specially. Float catalog sent on request.



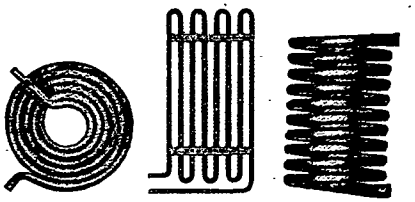
B-280 Convex *B-290 Convex* *B-281 Concave*

Copper Expansion Joints

For low pressure and vacuum. Made in two styles—convex and concave. Sizes 4 in. to 60 in. diameter. Cast iron or steel flanges. Flanges drilled to American standard unless otherwise ordered: B-290 available only in sizes 4 in. to 15 in. inclusive.

Coils

For heating, cooling and condensing. All shapes made from any size pipe or tube—standard or special connections, of copper, brass, aluminum, stainless steel, KA2 SMO, monel, inconel, nickel, block tin, and Everdur.



Bends

We make bends in every shape from all sizes of copper water tube, pipe and tubing in copper, brass, aluminum, stainless steel, monel, tin and nickel. Standard or special connections. U-bends for storage water heaters.

Also special pipe work for industrial installations, plumbing, heating and brewing. Perforated pipe, double pipe coolers, etc.

Non-Ferrous Castings—"Dairywhite" nickel silver for Process Industries Equipment. Suitable for milk and food products machinery. Castings also of 88-10-2. 80-10-10, 85-5-5-5, silicon bronze and manganese bronze, and special mixtures. Many patterns available without charge.

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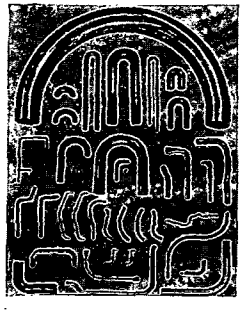
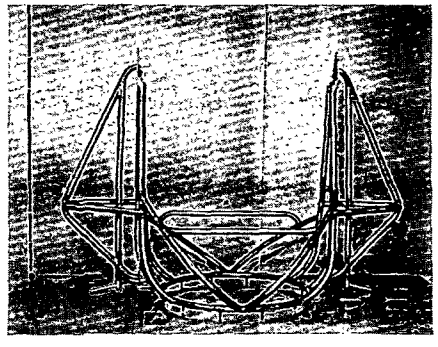
Bending Engineers—Contract Manufacturers

FABRICATORS OF FERROUS AND NON-FERROUS PIPE AND TUBING INCLUDING STAINLESS STEEL AND ALUMINUM

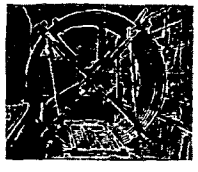
BENDS

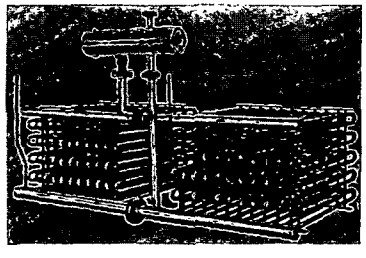


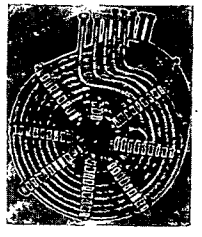
FRAMES and FRAME Assemblies



COILS







We specialize in all types of pipe and tube bending, coiling, expanding, and other types of forming and fabrication. By specializing in this line and manufacturing against order only, our service has taken us into many different fields and we have aided in the development of many new products as well as in the improved application of standard materials.

Our organization is large enough to be effective and small enough to be flexible. We solicit your quotation irrespective of size.

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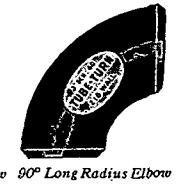
90° Short Radius Elbow



90° Long Radius Elbow

45° Long Radius Elbow

Straight Tee

Reducing Outlet Tee

TUBE TURNS is the original manufacturer of seamless welding fittings and also pioneers in their development and use. Complete details and engineering data on 4000 types, sizes and materials are included in the Tube-Turn **Catalog No. 111** sent on request.

Concentric Reducer

Eccentric Reducer

180° Long Radius Return

Lap Joint Stub End

Cap

Straight Lateral

45° Shaped Nipple

90° Shaped Nipple

Saddle

Straight Cross

Slip-on Flange

Screwed Flange

Lapped Flange

Blind Flange

Welding Neck Flange

Welding Ring

TUBE-TURN WELDING FITTINGS—RANGE OF SIZES

Type of Fitting	Description	Standard Weight	Extra Strong	Schedule 160	Double Extra Strong	Light Gauge Nominal Pipe Size	Light Gauge Iron Pipe Size
Elbows	90° Long Radius	½"–24"	¾"–24"	1"–12"	1"–8"	4"–24"	3"–12"
Elbows	90° Short Radius	1"–30"	1½"–30"				
Elbows	45° Long Radius	½"–30"	¾"–30"	1"–12"	1"–8"	4"–24"	3"–12"
Returns	180° Long Radius	½"–24"	1"–24"	1"–12"	3"–8"	4"–24"	3"–12"
Returns	180° Short Radius	1"–30"	1½"–30"				
Returns	180° Extra Long Radius	1"–2½"	1"–2½"				
Tees	Straight	¾"–24"	¾"–24"	1"–12"	1"–8"		
Tees	Reducing Outlet	¾"–24"	¾"–24"	2"–12"	1"–8"		
Reducers	Concentric & Eccentric	1x½–24x20	1x½–24x20		1x½–12x10		
Caps		1"–24"	1"–24"		1"–8"		
Stub Ends	Lap Joint	1"–24"	1"–24"				
Nipples	Shaped, 90° to Header	1¼"–12"	1¼"–12"				
Nipples	Shaped, 45° to Header	1¼"–12"	1¼"–12" 2"–24"				
Saddles							
Laterals	Straight	1¼"–24"	1¼"–24"				
Laterals	Reducing-on-Run	1¼"–24"	1¼"–24"				
Crosses	Straight	¾"–24"	¾"–24"				
Rings	Welding	¾"–12"	¾"–12"				
Sleeves	Welding	2"–24"					

SUPER Tube-Turn 45° long radius elbows, 90° long radius elbows, and 180° long radius returns available in both Standard weight and Extra Strong in sizes from 3" to 12". Tube-Turn welding fittings and flanges conform to applicable ASA and ASTM Standards. For further details please refer to Tube Turns catalog and data book No. 111.

*30 in. size short radius.

TUBE-TURN FORGED STEEL FLANGES—RANGE OF SIZES

	150 Lb.	300 Lb.	400 Lb.	600 Lb.	900 Lb.	1500 Lb.	2500 Lb.
Welding Neck	½"–24"	½"–24"	½"–24"	½"–24"	½"–24"*	½"–24"	½"–24"
Slip-On	½"–24"	½"–24"	½"–24"	½"–24"	½"–24"*	½"–24"	½"–24"
Lap Joint	½"–24"	½"–24"	½"–24"	½"–24"	½"–24"*	½"–24"	½"–24"
Threaded	½"–24"	½"–24"	½"–24"	½"–24"	½"–24"*	½"–24"	½"–24"
Blind	½"–24"	½"–24"	½"–24"	½"–24"	½"–24"*	½"–24"	½"–24"
Socket Type	½"–24"	½"–4"		½"–3½"			
Reducing-threaded or slip-on	½"–24"	½"–24"	½"–24"	½"–24"	½"–24"*	½"–24"	½"–12"

†Dimensions on sizes thru 3½ in. same as for 600 lb. flanges.
*Dimensions on sizes thru 2½ in. same as for 1500 lb. flanges.

Alco Valve Company

ENGINEERED REFRIGERANT CONTROLS

851 Kingsland Avenue, **St. Louis 5, Mo.**

NEW YORK OFFICE: 122 EAST 42nd STREET

THE COMPLETE LINE OF REFRIGERANT CONTROLS

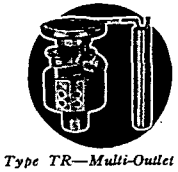
ALCO THERMO EXPANSION VALVES

THERMO EXPANSION VALVES: for automatic control of liquid refrigerant on all types of refrigeration and air conditioning systems. *Capacities*—from fractional tonnage to 100 tons Methyl Chloride, 50 tons "Freon-12".

Type TK—with famous 10-second capacity change feature—"3 valves in 1"

Type TCL

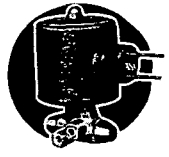
Type TIL



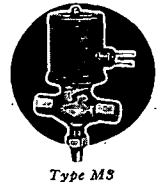
Type TR—Multi-Outlet

ALCO SOLENOID VALVES

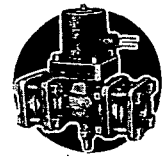
SOLENOID VALVES: for all types of service. *For Liquid:* "Freon"—up to 75 tons. Methyl Chloride—up to 150 tons. *For Suction:* "Freon"—up to 8.8 tons. Methyl Chloride—up to 17 tons.



Type S1



Type M3



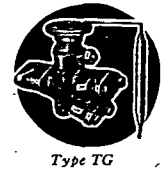
Type R2

ALCO AMMONIA CONTROLS

AMMONIA CONTROLS: *Solenoid Liquid Valves*—up to 172 tons. *Solenoid Suction Valves*—up to 28 tons. *Thermo Expansion Valves*—from fractional tonnage to 60 tons.

Type EPR13

Type M9P



Type TG

ALCO ALSO MAKES: ***Evaporator Pressure Regulators* for "Freon", Methyl Chloride and Ammonia with connection sizes up to 6 in.—*Solenoid Valves* for brine, water, gas, air and steam—*Float Switches*—*High Pressure Float Valves*—*Constant Pressure Expansion Valves*—*Liquid and Suction Line Strainers.***

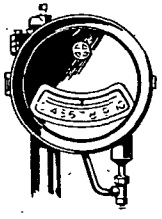
American Meter Company

INCORPORATED

General Offices: 60 East 42nd St., **New York 17, N. Y.**

ALBANY BALTIMORE BIRMINGHAM BOSTON CHICAGO ERIE KANSAS CITY
LOS ANGELES PHILADELPHIA PITTSBURGH SAN FRANCISCO
DALLAS, HOUSTON, TULSA—Westcott & Greis, Inc. ALHAMBRA, CALIF.—Reliance Regulator Corp.
HAMILTON, ONT.—Canadian Meter Co., Ltd.

Measurement and Control of Gas, Oil, Steam, Air and Liquids



FLOWMETERS

Indicating . . . Recording . . . Integrating

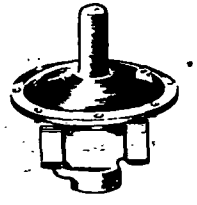
The **Metric Indicating Flowmeter** is an orifice type meter which *indicates* the rate of flow of air, gas, oil, steam or water through a line in which an orifice is installed. The **Integrating Flowmeter** *charts* a totalized reading of the quantity of gas, air, or steam at constant pressure—or for liquids, regardless of the rate.

Metric Flowmeters—known for their simplicity of design—assure trouble-free operation and ease of adjustment.

Write for Bulletin E-3.

RELIANCE REGULATOR

Type A

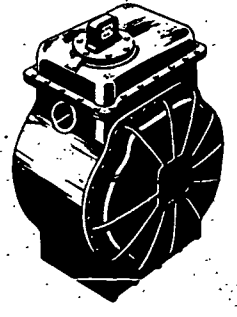


Reliance *Type A* regulators are designed to give absolute control of low pressures and are used extensively for controlling pressure to air and gas appliances requiring positive control of operating pressures.

A complete line of Reliance regulators are notable for exacting duty and minimum pressure loss at required capacity, as well as ease of servicing and low cost of upkeep.

Automatic shut-off valves are available for jobs that require a safety feature, so that when inlet pressures fall too low, the flow of gas through the regulators is stopped.

Write for Bulletin 43.



The Compact

IRONCASE GAS METER

Space requirements of these meters are cut down to the least possible dimensions without sacrifice of **Metric Ironcase** advantages.

Ironcase positive displacement meters set up a standard of unsurpassed accuracy in the measurement of gas by making use of the approved slide valve and full bellows principle . . . then by adding to this principle an extraordinary ruggedness of construction.

The body is cast in one piece. No joints and gaskets between internal parts . . . and no possibility of internal leaks. Special safeguards against friction loss are found throughout each meter.

Low pressure capacities up to 10,000 cu ft at 2 in., differential.

Write for Bulletin EG-40.

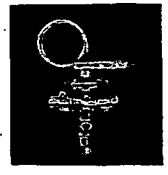
AUTOMATIC PRODUCTS COMPANY

2450 NORTH THIRTY — SECOND STREET

MILWAUKEE WISCONSIN

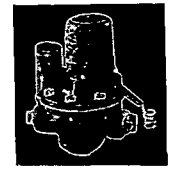
A-P DEPENDABLE CONTROLS

For Heating, Refrigeration and Air Conditioning

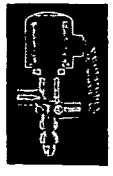


A-P Thermostatic Expansion Valves. Several models and sizes, for capacities up to 16 tons Freon or 32 tons Methyl Chloride.

A-P Solenoid Operated Water Valve. Made especially for Deep Well Cooling.

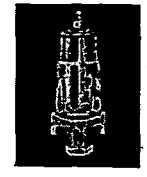


A-P Thermostats.
For Cooling or Heating



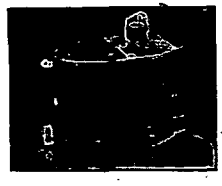
A-P Solenoid Refrigerant Valves. Capacities up to 50 tons Freon.

A-P "Trap-Dri."
Combined Filter-Strainer-Drier
Traps dirt, scale, moisture in refrigeration systems.

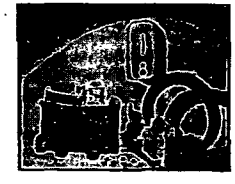


A-P Water Regulating Valves. Capacities up to 1440 Gallons per hour.

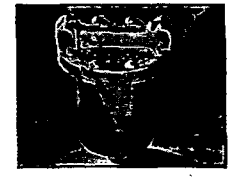
A-P Controls for Oil Burning, Gravity-Feed Heating Plants.



A-P Constant Level Oil Control Valve—With Fuel Compensator. Used on Gravity Oil Burning Appliances.



A-P Complete Furnace Control Set—Made in variety of types for Gravity-Feed Oil Burning Furnaces.



A-P Fuel Oil "Trap-It"—Traps dirt and water in fuel systems. Improves operation of all oil burning devices.

A-P Valve DEPENDABILITY

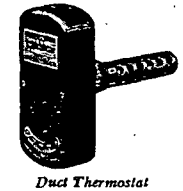
is widely recognized in Refrigeration, Air Conditioning and Heating. This reputation is born of close adherence to a rigid standard of perfection—in materials used, careful testing, inspection and simplicity of construction.

Barber-Colman Company

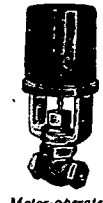
Rockford, Illinois

Automatic Control Systems for Heating, Ventilating, Air Conditioning

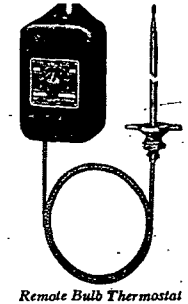
Room Thermostat



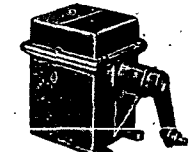
Duct Thermostat



Motor-operated Shut-off Valve



Remote Bulb Thermostat



Stall Type Control Motor

Barber-Colman Controls are all electric; precision built to insure long, continuous and dependable service; easy to install in either new or existing buildings; and ready for instant service, even after long shut-down periods.

Thermostats. All types—room, duct, immersion, air stream and remote bulb. For 2-position, floating, and proportioning control.

Hygrostats. Room and duct types.

Motor-Operated Valves. Packless, packed single-seat, pilot piston, V-ported, balanced, 3-way, and butterfly. For 2-position, floating, or proportioning control. Also Solenoid Valves for air, oil, water and gas. *Motor-operated valves are powered with Barcol motors which have only one moving part and require no attention except oiling; oil submerged operators require no attention.*

Control Motors. Uni-directional, or reversible fixed or adjustable speed. For 2-position, floating or proportioning operation of dampers in heating, ventilating or air conditioning applications. *Oil submerged models have the motor and gear train entirely submerged in oil.*

Program Switches. Automatic contact making mechanism for multi-compressor control or similar applications.

Econostat (not illustrated), a complete self-contained thermostatic unit for automatic regulation of the heat supplied to a building in accordance with outdoor temperatures.

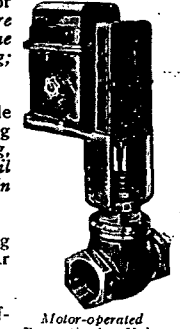
Write for descriptive literature.

LISTED AS STANDARD BY UNDERWRITERS LABORATORIES

(See also Page 1084)

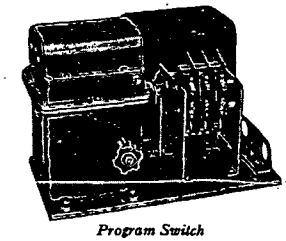
Room Type Microtherm

Hygrostat

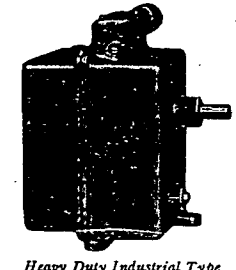


Motor-operated Proportioning Valve

Proportioning Type Control Motor



Program Switch



Heavy Duty Industrial Type Control Motor

Detroit Lubricator Company

Division of American Radiator & Standard Sanitary corporation

General Offices 5900 Trumbull Ave., **Detroit 8, Michigan, U. S. A.**

New York 18, N. Y., 40 West 40th Street
Chicago 5, Ill., 816 S. Michigan Avenue Los Angeles 13, Calif., 320 Crocker Street

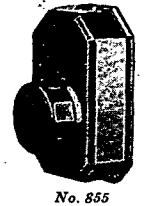
Canadian Representative:
Railway and Engineering Specialties Limited, Montreal, Toronto, Winnipeg

Automatic Controls

Detroit Lubricator Company manufactures a very complete line of electrical controls, designed to open or close an electrical circuit with changes of temperature or pressure. The No. 411 Thermostat (illustrated) is a low voltage model and is made in plain and Day and Night types. All No. 411 Thermostats are available with heat compensation to provide smooth, accurate temperature control.

No. 411

The No. 855 Mercoid Room Thermostat (illustrated) is a line voltage type—available in heating, air conditioning and refrigeration ranges.



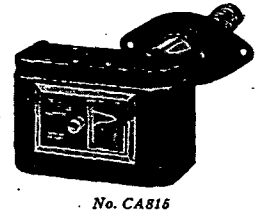
No. 855

For industrial use the No. 250 and No. 450 line of pressure and temperature controls is available, in pressure ranges from 20 in. vac. to 350 lbs, and in temperature ranges from −30° F to 495° F.

Furnace Controls

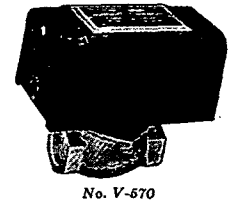
Also available is a full line of blower controls, combination blower and limit controls, such as the CA-815 illustrated, and a special line of water-tight equipment for use in wet locations.

There is a "DL" control available for practically every application where a dependable device is required to open or close an electrical circuit with changes of pressure or temperature. Write for complete information. Our Engineering Department is always ready to make recommendations on any specific problem.



No. CA815

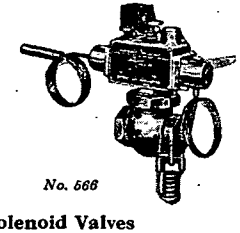
Gas Valves



No. V-570

The No. V-570 Electric Gas Valve is an electrically operated valve for control of gas lines from ½ to 1¼ in. It provides partial opening upon initial operation, permitting quiet ignition of gas. Inexpensive, compact, and attractive in appearance, it employs an easily serviceable bimetal strip motor for actuating force. Write for Bulletin 201.

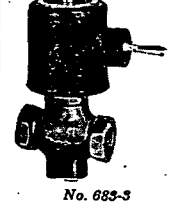
The No. 566 Valve combines in one compact unit all of the functions of a control system for gas heating. All safety functions are mechanical and operate independent of electric current. Adjustable for full snap or any degree of throttling action. Limit control closes valve with a snap action. Valve may be manually operated in case of current failure. Applicable to all gases, natural, manufactured, or mixed. Write for Bulletin No. 80.



No. 566

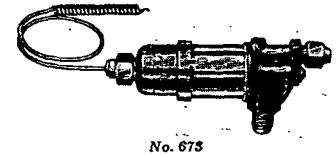
Solenoid Valves

"DL" Solenoid Valves for control of water, air, oil, gas, or refrigerant, embody many desirable features. They are free from A.C. hum and will open against high pressures. Available in all standard voltages and cycles A.C. or D.C. No. 683-3 (illustrated) is a small size valve with ⅜ in. connections. No. 681 is a pilot operated, intermediate size valve, and the No. 686 is a large pilot operated valve with



No. 683-3

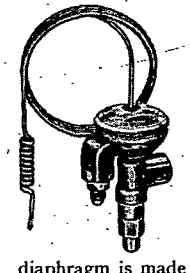
capacity up to 17 tons Freon. No. 686 valve available with flanged connections. No. 681 and 686 are furnished with manual opening feature to permit opening in case of current failure. All models may be taken apart and cleaned in the field without removing from pipe lines. Write for Bulletin No. 199.



No. 673

Expansion Valves

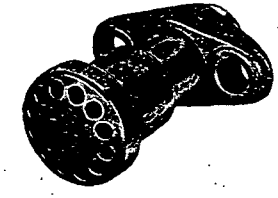
"DL" Thermostatic Expansion Valves are designed to keep the evaporator in a refrigerating system completely refrigerated. All power elements are "gas charged" to a definite pressure, preventing motor overload and providing quicker response and more sensitive control. Capacities from ⅛ ton to 30 tons Freon. The No. 673 valve employs a double bellows as the actuating means, while the "Dura-fram" line is constructed with a single diaphragm power element. All needles and seats are made of Delubaloy, a very hard, corrosion resistant alloy to insure long, trouble-free service. Write for Catalog No. 200 on refrigeration equipment.



"Dura-fram" Expansion Valves

These valves are of single diaphragm construction. The diaphragm is made of special alloy, carefully tested for flexibility to assure uniform capacities and smooth operation. Power elements are gas charged for protection against motor overload and for maximum sensitivity. Needles and seats are hard, corrosion resistant Delubaloy.

Automatic and thermostatic types in capacities from ½ to 20 tons Freon and .9 to .36 tons methyl chloride.



Refrigerant Distributors

Detroit No. 790 Refrigerant Distributors insure a uniform supply of refrigerant to all sections of multiple circuit evaporators and are attached directly to the expansion valve by means of a flanged connection. These corrosion-resistant distributors are available for "Dura-fram" Expansion Valves No. 786, No. 787 and No. 788, and are interchangeable on all types.

They are made in two types, as follows:

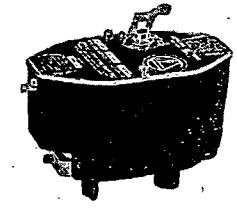
Type A—2 to 8 passes.
6 passes standard.

Type B—9 to 18 passes.
12 and 18 passes standard.

All distributors are designed for use with ¼ in. O.D. copper outlet tubes.

For complete capacity tables write for Bulletin No. 207.

"DL" Float Valves



Automatic compensator assures even fuel flow regardless of fuel temperature—so simple to clean that servicing is no problem.

Available in both manual and automatic types. Manual—single, dual, or non-metering. Automatic types—for furnaces, floor furnaces, water heaters, etc.—with either "high-low" or "high-off" burners, natural or forced draft.

The Fulton Sylphon Company

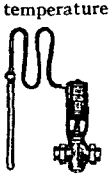
Manufacturers of Sylphon Automatic Temperature Controlling Instruments and Packless Expansion Joints

Sales Representatives in Principal Cities

Knoxville, Tenn.

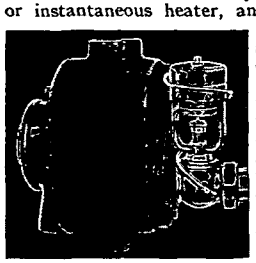
HOT WATER SUPPLY

No. 923 Temperature Regulator—For controlling water temperature in heaters, open or closed tanks and other equipment. Operation unaffected by temperature fluctuations at the valve, either above or below bulb temperature. All parts, except steel adjustment spring, made of non-ferrous metals. May be installed in any position. Ranges from 40 —80 F to 290 —330 F. Bulletin HVG-20.



No. 923 Temperature Regulator

Sylphon Thermostatic Water Mixers Utilize hot water from any storage tank or instantaneous heater, and effectively regulate the amount of cold water required to temper it to the desired degree, actually mixing the hot and cold water together before delivery. Temperature remains constant in spite of fluctuations in supply water temperatures or pressures.



No. 902 Sylphon Thermostatic Water Mixer—14 to 131 gpm depending on water pressure

Four sizes with capacities ranging from 5 to 131 gpm. Bulletin HVG-40.

REFRIGERATION CONTROLS

Adaptable wherever brine is used as the refrigerant. Latest development is a "freeze-proof" valve (illustrated at left on the popular Sylphon No. 945-Z Regulator). Bulletin HVG-20.

No. 945-Z Regulator

PACKLESS EXPANSION JOINTS

The Sylphon Packless Expansion Joint eliminates useless building height, expensive construction and non-revenue producing space. No costly leaks and repairs, no repacking, always tight, allows heating system to operate at full efficiency. Write for Bulletin HVG-140.

No. 110 Sylphon Expansion Joint

SPACE HEATING CONTROL

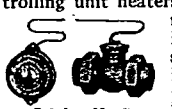
No. 885 Automatic Radiator Valve—For exposed radiation. Small, neat, finely finished, adjustable to room temperature desired. Simply replace ordinary radiator valves with these Sylphon Automatic Regulators—no wiring, piping or auxiliary equipment are required. These valves answer the demand for an inexpensive means of providing accurate, dependable space temperature control in rooms, sections or throughout large buildings, new or old. Similar type valves for concealed radiation — get Bulletin HVG-80.

Sylphon No. 885 Automatic Radiator Valve

No. 890 Electric Radiator Control Valve—For either exposed or concealed radiation. Similar in appearance and action to Sylphon Automatic Valves, but operated by an electric wall thermostat. The closing of the thermostat circuit energizes a low voltage electric heater coil surrounding a bulb containing a volatile liquid. This liquid expansion causes pressure on a bellows in the valve head operating the valve. This provides radiator valve control from a remote location, permits regulation of several radiators from a single thermostat, enables a time switch to be installed, if desired, offers effective zone control of large areas at a fraction of the cost of conventional motor-operated valve systems. Bulletin HVG-70.

Sylphon No. 890 Electric Control Valve

No. 7 Temperature Control—A self-contained, self-powered regulator for controlling unit heaters, wall or ceiling type radiators, heating coils in duct-type heating systems, etc. Quickly installed, holds temperatures within close limits. Valve placed in steam line to one or a battery of heaters, thermostat mounted on wall or column. For use on regular heating pressures up to 15 lb. Similar regulators, Nos. 7-2 and 7-3 for 50 and 75 lb pressure and temperatures up to 170 F. Bulletin HVG-50



Sylphon No. 7 Temperature Control (Self-operating)

HEATING AND AIR CONDITIONING CONTROL

Almost any type of heating, ventilating or air conditioning system can be advantageously controlled wholly or in part by Sylphon Regulators. Basic advantages of Sylphon Controls are:

Modulating—Maintains ideal conditions —not continually correcting too hot, too cold, too humid or too dry conditions.

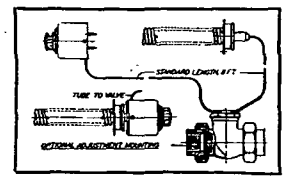
Compensating—Many Sylphon Regulators offer compensating control, automatically raising their low limit setting at a predetermined rate as outside temperatures fall.

Sensitive—Close operating temperature differentials. Quick response.

Simple—in design.

Rugged Construction—To give years of satisfactory service.

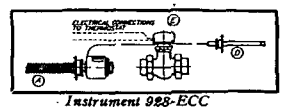
Adaptable—Any one of many combinations of Sylphon Instruments can be arranged to control any air conditioning system and to provide exactly the conditions desired. *Write for Bulletin SAC—820.*



Regulator 928-C

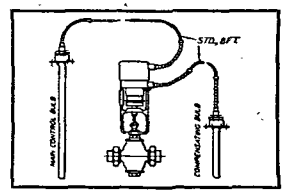
The No. 928-C Regulator—Simple, compact yet highly sensitive. Suitable for modulating control of air temperatures in ducts. Bulb is constructed of numerous coils of copper tubing giving sensitivity to the slightest temperature variation. Packless valve eliminates service problem and makes this regulator ideal for installation in inaccessible locations. Suitable for steam pressures up to 15 lb; other types available for pressures up to 75 lb.

The No. 928-ECC Sylphon Instrument—Room control and low-limit control in a single valve regulator for modulating control of ventilating systems. Main control from an electric room thermostat operating through the electric head "E" on the valve. Low-limit control by Bulb "A" located in discharge duct from the heater. Bulb "D", located in inlet side of the duct to the heater, compensates Bulb "A". Compensating thermostat can be furnished to raise low-limit setting at predetermined rate with falling outside temperature. Suitable for steam pressures up to 15 lb.

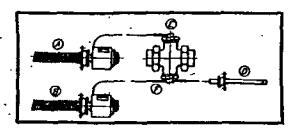


Instrument 928-ECC

Sylphon No. 971 Differential Regulator—For controlling room temperature on the cooling cycle, where chilled water or brine is used as cooling medium and where it is desired to have a gradual increase in room temperature as outside temperature increases. This regulator is modulating in action, thereby affords better control over humidity than is procured when usual on-and-off type control is employed.



Regulator 971

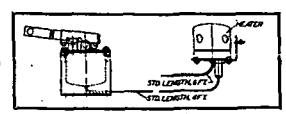


Control 889-C

The Sylphon No. 889-C Control—A modulating, dual-function regulator for control of duct heating and ventilating systems—two independent valves in a single body.

Adjustable Thermostat "A" governing Valve "E" functions to maintain room temperature from temperature of recirculated air. Adjustable Thermostat "B" acts as a low-limit ductstat controlling Valve "F" to maintain minimum discharge air temperature. Bulb "D" compensates Bulb "B" to maintain even discharge air temperature irrespective of demand. Compensated Thermostat "B" can also be furnished to raise its setting at a predetermined rate with falling fresh air temperatures, if desired. Suitable for steam pressures up to 15 lb.

Sylphon No. 371 Positive Type Damper Motor—On-and-off control of dampers. Operation controlled by room thermostat, by hand-operated switch, by motor starting switch, etc. Advantages include: (a) motor returns to closed or safety position in event of current failure; (b) heat-motor bulb and motor separate enhances convenience of installation; (c) damper motor lever adjustable; (d) positive, powerful operation.



Damper Motor 371

GENERAL CONTROLS

FACTORY BRANCHES: Atlanta • Boston • Chicago • Cleveland • Dallas • Denver • Detroit • Houston • Kansas City • New York • Philadelphia • Pittsburgh • San Francisco • Seattle

DISTRIBUTORS IN PRINCIPAL CITIES

HOME OFFICES AND FACTORY: 801 Allen Ave. Glendale 1, California

AUTOMATIC TEMPERATURE—PRESSURE—FLOW CONTROLS

THERMOSTATS

The Trimtherm

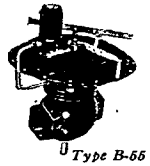
Trimtherm provides accurate, remote control of desired temperature. Streamlined, compact—extends only 7/8 in. from wall, yet sensitive to slightest temperature change (Differential 1/2 F.) Chrome cover, natural ivory base.

MAGNETIC GAS VALVES

Type K-3B

Versatile, two-wire, straight magnetic current-failure valve. Packless. Insures tight shut-off indefinitely. Humless. Size range, 7/8 in. to 6 in. I.P.S. Operating pressures up to 5 lb. Voltages and frequencies, A.C. or D.C. Quiet, positive, trouble-free. Available in explosion-proof housing.

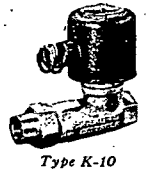
SLOW OPENING GAS VALVES



Type B-66

For industrial and commercial burners and furnaces. Adjustable opening time, 5 to 50 secs. Wide pressure range up to 5 lb. Size 3/4 in. to 6 in. I.P.S. Ample power for louvre control. Damper arm easily rotated. Packless.

MAGNETIC VALVES



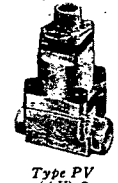
Type K-10

Provides six times more power than ordinary solenoid valves. Controls air, gas, water, light, heavy oils, steam. Positive opening, complete shut-off, packless, hum-free. Available for any voltage, A.C. or D.C., in sizes up to 1 1/4 in. I.P.S., port sizes up to 5/8 in. K-20 Series is designed for applications where single needle port sizes provide sufficient flow capacity. W-3 Series are magnetic 3-way valves for controlling fluid to piston and diaphragm operators on valves, doors, gates, etc.

B-60 GAS ACTUATED PACKAGE SETS

Everything needed in a convenient package for remote gas control—B-60 Valve, Trimtherm (with or without thermometer, or Timer Thermostat), Pilot Burner Generator, and 30-ft wire. No outside current required. Safe, quiet, dependable. Applicable to gas furnaces, floor furnaces, boilers, circulators, gas radiators.

*hi-g MAGNETIC VALVES

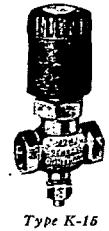


Type PV (AV)-5

Designed for positive operation on aircraft, trucks, tractors, tanks, graders, ships, and other moving equipment. Handle all fluids, vapors and gases on anything that rolls, floats or flies at pressures up to 3000 lb or more. Packless, two-wire, current-failure type, available normally open, normally closed for intermittent or continuous duty.

*Trade Mark—"hi-g" indicates positive ability to function in any position, regardless of vibration, change of motion or acceleration.

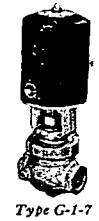
REFRIGERANT CONTROLS



Type K-15

Magnetic piloted, two-wire, current failure, high pressure, packless. Handle large capacities with minimum pressure drop and loss. Tight shut-off. Operates on wide variety of fluids and gases.

HYDRAMOTOR VALVES



Type G-1-7

Simplify valve control installations. Two-wire, current failure, electric-hydraulic operation. Ample motor-driven power, slow opening and closing movement. Operator drives against a spring in one direction; power failure or opening of circuit causes spring to operate in other direction.

MANUAL RESET VALVES

Type MR-1-2

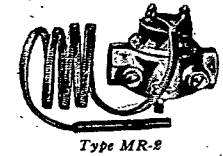
Equipped with manually-reset electromagnetically-held valve operator. Current flowing to operator permits manual opening by turning valve wheel at side. Current failure releases operator allowing valve to close. Trip-free mechanism cannot be opened under unsafe conditions. Once closed, valve is re-opened manually because applying current has no effect.

STRAINERS

Type S-5-1

Actual tests prove importance of strainers in prolonging operating life and reducing leakage of flow controls. S-5 Series STRAINERS come in 8 types; meshes 3/16 in. diam. to 120-per-inch.

THERMOVALVES



Type MR-2

Manually-reset, electromagnetically-held-open valve with current generated by single couple subject to heat of pilot flame. Available 3/8 in. to 1 1/2 in. I.P.S.

THERMOPILOTS

Type A-100

Proven principles of operation insure dependability. Flame applied to thermocouple maintains electrical contact, allowing electrical gas valve to open. When flame fails, Thermopilot opens circuit to main gas valve. Flexible, armored, asbestos-covered cable detachable from relay. 2 or 3-wire control. Electrical rating, 2 amp., 24 volt; 1 amp., 115 volt; 0.5 amp., 230 volt.

THERMAL EXPANSION VALVES

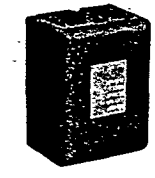
Type V-200 provides unmatched sensitivity and dependability. Handles freon, methyl chloride, sulphur dioxide. Quickly removed orifice cartridges eliminate need for stocking several sizes for low tonnage installation.

GAS FUEL GOVERNORS

Type V-25-1

Throttle gas lines according to boiler pressure applied to diaphragm. Ball bearing thrust adjustment, ground and polished non-corrosive stems, low friction packing gland seal, multiple calibrated springs, high lift for maximum capacity. Suitable for butane, natural or manufactured gas. Available 3/8 in. to 3 in., I.P.S.

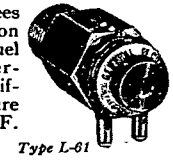
RELAYS AND TRANSFORMERS



Type RS-100

Type RS-100 handles single phase motor loads up to 1 hp or heating loads up to 1.1 kw. Combines double-break relay and integral transformer. Normally open; large double-break contacts. Two-wire control circuit; maximum holding current 0.4 amps. Furnished with 1/2 in. conduit connections and low voltage outlet. A.C. only.

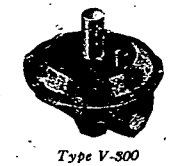
TANK THERMOSTATS



Type L-61

Averages 6 to 10 degrees effective differential on water heater. Saves fuel by eliminating overheating due to large differentials. Temperature range, 60 to 170 F. Also to 230 F.

LOW PRESSURE GAS REGULATORS



Type V-300

New V-300 Series are reliable, trouble-free valves with high capacity, close regulation, yet small and compact. Regulator sizes range from 3/8 to 2 in. I.P.S. Internal parts, corrosion-resistant. Cast iron regulator bodies, long life calibrated springs, properly fitted all-metal valves.

GENERAL CONTROLS manufactures a complete line of Automatic Temperature, Flow and Pressure Controls. For complete specifications write for *Catalog 52B*.

Friez Instrument Division

BENDIX AVIATION CORPORATION

Towson, Baltimore 4, Maryland

Manufacturers of Precision Recording and Measuring Instruments for over 70 years—now will manufacture a complete new line of Automatic Heating, Air-Conditioning and Refrigeration Controls and Accessories.

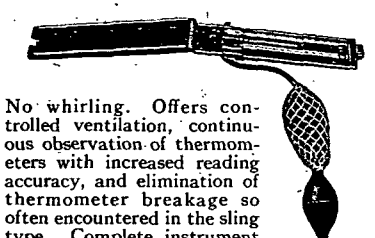
HUMIDISTAT

Employs the ever reliable dual hair element to operate electrical contacts. The ideal control for residential, commercial or industrial humidifying or dehumidifying equipment. Insertion type, not shown, also available in several models.

THERMOSTAT

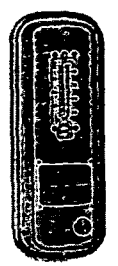
A complete range of highly sensitive and thoroughly dependable thermostats.

PSYCHROMETER, Hand Aspirated Type



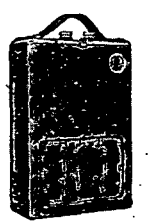
No whirling. Offers controlled ventilation, continuous observation of thermometers with increased reading accuracy, and elimination of thermometer breakage so often encountered in the sling type. Complete instrument with psychrometer slide rule, folds into a protected pocket assembly.

HUMIDITY INDICATOR



Range: 10-95 per cent Relative Humidity. Employs the Friez dual hair element to provide high sensitivity and best accuracy. Includes thermometer as illustrated.

RECORDER, Portable



For surveys, studies, and tests of humidity and temperature. A third pen (optional) provides an on-off record of associated electrical equipment under test. Provides inked record on 3 x 5 cards, suitable for filing.

Write for descriptive literature stating your requirements.

Henry Valve Company

3260 W. Grand Ave., **Chicago 51, Ill.**

PACKLESS AND PACKED VALVES • DRYERS FOR REFRIGERATION AND AIR CONDITIONING • STRAINERS • AMMONIA VALVES • FORGED STEEL VALVES AND FITTINGS FOR OIL, STEAM AND OTHER FLUIDS.

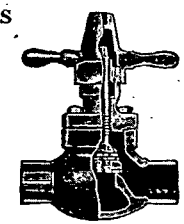
Balanced-Action Diaphragm Packless Valves

VALUE OF BALANCED-ACTION

Regardless of operating conditions or the differential in the pressure above and below the valve seat, balanced-action assures positive and instantaneous opening. The balancing action is really the equalizing of the pressures on the two sides of the valve seat at the instant of opening. This is accomplished by a channel in the axis of the valve stem. When the valve is closed, the upper port of this channel is sealed by the diaphragm assembly itself. At the instant of opening, the pressure above the seat forces the diaphragm assembly upward, exposing the upper port of the balancing channel. The pressure is released through the channel to the region below the seat, equalizing the pressures, thus assuring positive opening. A spring-tensioned ball check seals the channel for diaphragm inspection.

Other Important Features are: Oval handwheel, ports-in-line, non-rotating bearing plate to protect diaphragm from rotating friction of stem, and use of multiple puncture and fracture-proof diaphragms designed to resist wear and corrosion. Available in a complete range of sizes with flare and solder connections.

WING CAP VALVES



Designed especially for Freon and Methyl Chloride. Have patented rotating self-aligning stem disc. Special resilient packing. May be repacked under pressure. Wing cap can be inverted and socket used for operating valve.

Made of non-ferrous alloy to meet government specifications. Solder connections machined directly in valve body.

HENRY STRAINERS

There is a size and type of Henry Strainer for every installation requirement.

Type 895 "Y" Strainer

With solder fittings for use with copper pipe. Welded steel construction. Negligible pressure drop. Screen can be taken out for cleaning without removing strainer from line. Very large screen area. Light weight. Baffle prevents heavy particles injuring screen.

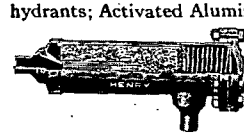
ABSO-DRY PRESSURE SEALED DRYERS

For Refrigeration and Air Conditioning

The exclusive Henry vacuum process first removes every trace of moisture, then the dryer is charged with dehydrated air. Loosening a seal cap prior to installation produces hissing sound, a guarantee of original factory dryness and freedom from leaks.

Other Features of Henry Dryers— *Perforated Dispersion tube* is connected to inlet port and exposes entire volume of dehydrant to penetration by refrigerant. *Minimum* pressure drop. No channelling. *Compression Spring* maintains uniform tension on dehydrant at all times and compensates for changes in volume. *Soldered or Flanged Shells*—models are available with either soldered cap or flanged end shells. Flange is distortion-proof. Shells not exceeding 5½ in. in length are drawn in dies, so that they have only one joint.

TWO DEHYDRANTS—Choice of two dehydrants; Activated Alumina or Silica gel.



Type 757 Cartridge Dehydrator

Flanged shell dehydrator with replaceable cartridge.

Type 743 Refillable Dehydrator

Screen tube assembly soldered to outlet connection. With dispersion tube.

Type 712 Dehydrator

Soldered brass shell with dispersion tube.

APPROVED FOR ARMY AND NAVY USE

Johnson Service Company

AUTOMATIC TEMPERATURE AND AIR CONDITIONING CONTROL

General Offices and Factory

Milwaukee, Wis.

Branch Offices in all Large Cities

JOHNSON TEMPERATURE REGULATING CO. OF CANADA, LTD., 113 SIMCOE ST., TORONTO, ONT.
HALIFAX, N. S. MONTREAL, QUE. WINNIPEG, MAN. CALGARY, ALTA. VANCOUVER, B. C.

PRODUCTS AND SERVICES

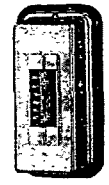
Manufacturers, Engineers, and Contractors—For automatic temperature and humidity control systems applied to all types of heating, cooling, ventilating, and air conditioning installations.

Space Control—Automatic control of room temperatures and humidities, applied to radiators, unit ventilators, unit heaters, and heat delivery ducts. Johnson "Duo-Stats" to maintain the proper relationship between outdoor and heating system temperatures for groups of radiators, or "heating zones." A complete line of devices for automatic control of air conditioning systems, heating, cooling, humidifying, dehumidifying.

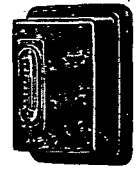
Process Control—Automatic temperature and humidity control devices for manufacturing and industrial processing, applied to tanks, dryers, vats, kettles, curing rooms, coolers, kilns, etc.

Nation-wide Service—Johnson sales engineers, technicians, and trained installation men are available at all branch offices. None of the men in the nation-wide Johnson organization are agents, jobbers, or part-time representatives. All are salaried employees, devoting their entire efforts to the interests of the JOHNSON SERVICE COMPANY and its customers.

Send for Bulletins describing the detailed characteristics of any of the Johnson devices.

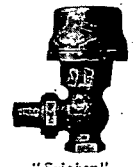


Single Room Thermostat



Dual Room Thermostat

Room Humidostat



"Syiphon" Radiator Valve

JOHNSON THERMOSTATS

Room Thermostats—Intermediate (gradual) or positive (snap) action, maintaining temperatures accurately within one degree above or below point of setting. "Dual" (two-temperature) and "Summer-Winter" types, as well as standard instruments. Various types of covers allow wide selection of adjusting features, guards, and method of mounting. Red-reading thermometers with magnifying tube attached to each cover.

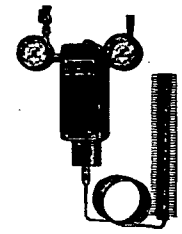
Insertion and Immersion Thermostats—Control temperatures in ducts, tanks, and similar locations. High grade insertion or immersion thermometers for mounting adjacent to the thermostats, including the distinctive Johnson insertion thermometer, with red-reading mercury column in heavy lens glass tube and 9-in. scale with patented adjustable tilting feature. Also, separable socket liquid thermometers.

Extended Tube Thermostats—Liquid-filled or vapor tension type, to measure temperatures at a point remote from the location of the operating mechanism. Various types of bulbs. Connecting tubing up to 50 ft in length for vapor tension.

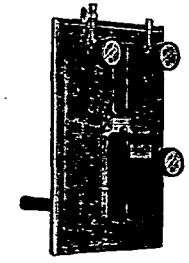
Special Thermostats—For applications encountered in industrial control, including the "Record-O-Stat," combination extended-tube temperature controller and recorder. Full 10-in. chart. Single or duplex type, the latter controlling and recording both wet and dry bulb temperatures.

Remotely Adjusted Thermostats—A distinctive Johnson feature, applied to various types of instruments where readjustment must be accomplished from a remote point, such as another thermostat or a manual switch.

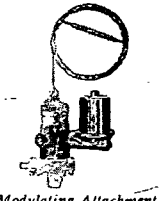
Johnson Sensitivity Adjustment—An important development in automatic temperature and humidity control for air conditioning. A unique and convenient means of adjusting the sensitivity of Johnson thermostats and humidostats, on the job, balancing "time-lag" with respect to the capacity of conditioning apparatus. "Hunting" and temperature fluctuation prevented. Available on all Johnson gradual action insertion and immersion thermostats, insertion humidostats, and certain room type thermostats and humidostats.



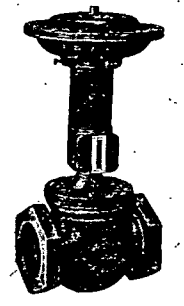
Extended Tube Thermostat



Remotely Adjusted Duct Thermostat



Modulating Attachment for Expansion Valves



Rubber Diaphragm Coil Valve

JOHNSON HUMIDITY CONTROL

Johnson Humidostats—Automatically control the supply of moisture delivered by a humidifier or by other means, maintaining a constant percentage of relative humidity. Available in both room and insertion patterns and with various types of elements as determined by requirements, the most sensitive controlling within 1 per cent at relative humidities as high as 95 per cent at 100 F. Humidostatic elements are wood-strip, human hair, animal membrane, or other suitable substances as selected.

Johnson Humidifiers—"Steam grid" type (perforated pipe supplied with low pressure steam) or pan type with copper evaporating pan, brass heating coils, and float control.

JOHNSON VALVES

Johnson Diaphragm Valves—Simple and rugged. Seamless metal bellows and heavy spring operate the valve stem. Available, if desired, with diaphragms of special molded rubber, resistant to aging, heat deterioration and oxidation. No complicated moving parts. Made in all standard sizes and patterns. Direct acting (normally open) or reverse acting (normally closed). Also, three-way mixing and by-pass valves. For steam, water, brine, and freon.

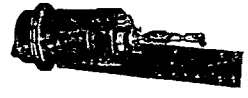
Johnson "Streamline" Diaphragm Valves—Modulating discs and special internal construction, insure superior gradual control . . . Where maximum power is required for repositioning at the slightest demand of controlling instruments, Johnson molded rubber diaphragm valves are fitted with Johnson's dependable pilot feature, for smooth gradual operation, independent of friction and pressure variations.

JOHNSON DAMPERS AND SWITCHES

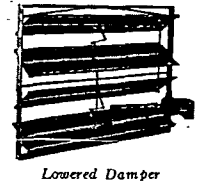
Standard Johnson Dampers—Steel blades in flat steel frames with adequate bracing to form a rigid assembly. Finished in two coats of black lacquer. Special corrosion-resisting finishes on order. Angle iron frames optional. **Special Dampers**—Galvanized iron, monel metal, aluminum, copper, rust-resisting steel, etc. Brass pins in steel bearings or ball bearings.

Johnson Damper Motors—In principle, similar to valves. Seamless metal or specially molded rubber diaphragm operates damper through suitable linkage. Various types of brackets. Distinctive Johnson "Piston-type" damper motors afford long travel at full power, a feature not found in other such devices. With or without pilot mechanism, as described above under "Valves."

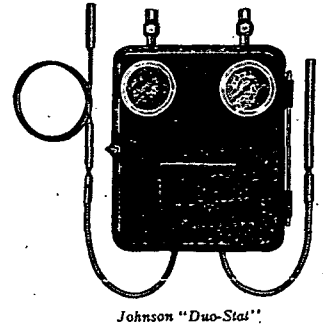
Johnson Pneumatic Switches—Various patterns for operation of dampers and for placing thermostats and other devices in and out of service, as required, from remote points. Standard switchboards are oiled slate. Ebony asbestos, polished oak, and genuine or imitation marble on order.



Piston Type Damper Motor



Lowered Damper



Johnson "Duo-Stat"

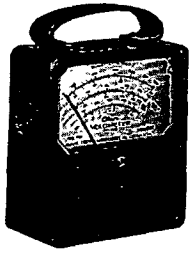
Illinois Testing Laboratories, Inc.

422 North LaSalle Street, **Chicago, Illinois**

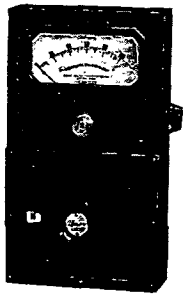
Alnor

Manufacturers of

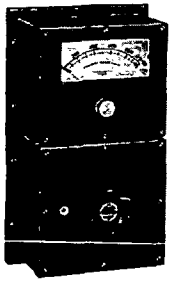
ALNOR Thermometers, Pyrometers, Temperature Controllers, and Air Velocity Meters



The Alnor Velometer is an instantaneous, direct-reading air velocity meter designed for convenient, rapid determination of air velocities in air conditioning, heating and ventilating, and exhaust systems. It gives instantaneous direct readings in feet per minute, without timing, calculations, or reference to tables or charts. Low range air velocity readings are taken by holding the **Velometer** in the air stream. High range readings are obtained by the use of tube connected special jets, available in 17 types providing for convenient use in any location. Accurate information on performance of equipment, duct systems, etc., can be obtained with a few moments inspection with the **Velometer.** It can be effectively used to locate drafts and leaks around windows and doors, or in duct systems.



The Alnor Velometer is built in several standard ranges from 20 fpm to 6000 fpm, and up to 3 in. static or total pressure. Special ranges available as low as 10 fpm and up to 25,000 fpm velocity and 20 in. pressure.



Alnor Electrical Resistance Thermometers, available in single and multi-point types, are ideally suited to air conditioning, heating and refrigeration installations. Thermobulbs may be located at any point up to 1000 ft from the instrument. Indicating thermometers can be furnished with capacities up to 14 points. Standard types include round, rectangular, horizontal edgewise, and portable thermometers. Instruments may be installed in office or engine room and thermobulbs located at any desired point in the building or on the roof. Accurate temperature measurements at the remote locations are obtained merely by moving the selector switch.

Alnor Pyrometers include a wide range of portable and wall mounting types, both single and multi-point instruments, for a variety of industrial temperature measurements. Temperature readings of molten metals, industrial furnaces, and similar equipment, and temperatures of surfaces, either metallic or non-metallic, moving or stationary, are readily obtained with standard **Alnor** Pyrometers.

Bulletins describing **Alnor** instruments and accessory equipment will be sent upon request.

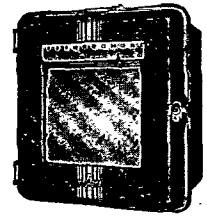
Leeds & Northrup Company

4941 Stenton Avenue, Philadelphia 44, Pa.

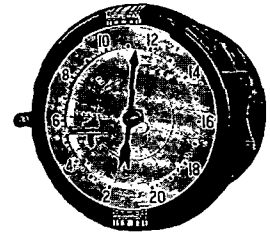
Branch Offices:

BOSTON, BUFFALO, CHICAGO, CINCINNATI, CLEVELAND, DETROIT, HARTFORD, HOUSTON, LOS ANGELES, NEW YORK, PITTSBURGH, ST. LOUIS, SAN FRANCISCO, TULSA

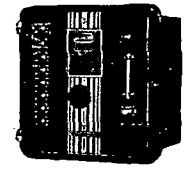
RUGGED, ELECTRICAL-BALANCE INSTRUMENTS



Model S Micromax Recorder
Records from 1 to 16 points on a single strip-chart. Extremely open record. Can also signal or control. (About 1/15th size)



Model R Micromax Recorder
Records 1 or 2 points on a round-chart. Has extremely readable dial. Can also signal or control. (About 1/15th size)



Panel Indicator
Hand-operated. Can be connected through selector switches to any number of points. (About 1/12th size)

Electrical Thermometers for Air Conditioning

No method for measuring temperatures fits the specific needs of air conditioning as does the three-lead, null-type resistance thermometer method. It is independent of distance and disregards all temperatures except those right at points of measurement. Thermohms (electrical resistance thermometers) can be placed anywhere—in rooms, air ducts or water lines. They are connected by simple electrical wiring to instruments at a central location. Instruments may be: Micromax Recorders, Model S for up to sixteen Thermohms, Model R, for related pairs such as wet and dry bulb; indicators with switches for any number of Thermohms; or indicating and recording combinations.

Sound in principle, reliable in operation, instruments and Thermohms are highly responsive, yet ruggedly built. A complete system is economical to install, regardless of distances. It is easy to operate; requires little maintenance. Thermohms and instruments are interchangeable; can be replaced without disturbing wiring or returning anything to the factory.

L&N Resistance Thermometers make it possible to operate efficiently; to maintain comfort or correct process atmosphere constantly . . . so that maximum return is realized on the conditioning investment. Ask for Catalog N-33C.

Jrl Ad N-225(2)

Electrical Instruments for the Steam Plant

Facts needed to run a modern heating plant so as to save fuel, protect equipment, and operate efficiently at varying loads are provided reliably by rugged L&N instruments. Readings can be indicated, recorded, or both. Recorders can be equipped to operate signals or alarms; in some cases to control automatically.

Micromax Model S provides a permanent record on one wide-scale chart of conditions at from 1 to 16 points. Micromax Model R concentrates on conditions at one point, provides a permanent record, has a giant indicating dial. A Panel Indicator provides intermittent checks on conditions at one or several points.

L&N equipments for the steam plant are described in the following publications:

Metermax Combustion Control (for large plants; central stations) Cat. N-01M-163.

L&N Combustion Control, Type P (for industrial and smaller stations) Cat. N-01P-163.

Micromax Temperature Instruments (for the Steam Plant) Cat. N-33-163.

Micromax Temperature Instruments (for Superheated Steam) Cat. N-33-163(1).

Centrimax Flowmeters, Cat. N-28-160.

Micromax CO_2 Recorders Cat. N-91-163.

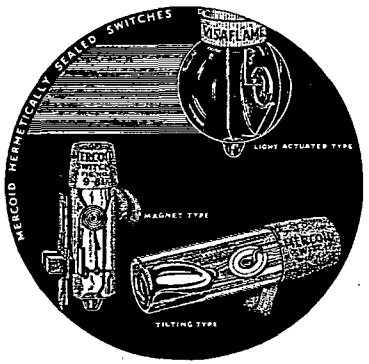
Micromax Smoke Density Recorders, Cat. N-93-163.

Micromax Condensate Purity Instruments (for the Steam Plant) Cat. N-95-163.

The Mercoid Corporation

Main Office and Factory, 4201 Belmont Avenue, Chicago 41, Illinois
New York Office, 205 East 42nd St. Philadelphia Office, 3137 N. Broad St.

AUTOMATIC CONTROLS FOR HEATING, AIR CONDITIONING, REFRIGERATION AND VARIOUS INDUSTRIAL APPLICATIONS



FOR PRESSURE & TEMPERATURE

These controls have a wide range of applications. They are noted for their accuracy and dependable performance. The outside double adjustment feature and visible dial eliminate all guesswork when setting the operating range.

VISAFLAME

The Mechanical Eye Actuated by Light. A positive safety control system for domestic and industrial oil burners. Operates direct from the light of the flame instead of from heat in the stack. Tried and proven to be the most practical method for dependable oil burner performance. (See next page.)

OIL BURNER SAFETY CONTROLS

Type JMI provides positive protection against flame or ignition failure on intermittent ignition oil burners. This control insures having ignition circuit closed before every starting operation of burner. Type JM is used for constant ignition burners.

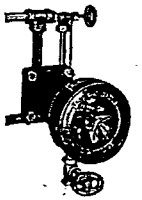
THE ONLY 100% MERCURY SWITCH EQUIPPED CONTROLS

The distinguishing feature of Mercoid Controls is the exclusive use of Mercoid hermetically sealed mercury switches. These switches are not subject to dust, dirt or corrosion, thereby assuring better performance and longer control life. The items shown below are but a few miscellaneous items. See Catalog No. 700 for complete line.

MERCOID THERMOSTATS

Mercoid low voltage room thermostats are known by their trade name Sensatherm. They operate on a total differential of 1 degree F. Type H is the regular popular room thermostat. Type DNH is a hand wound day and night Sensatherm. Type HBH is the two-stage type for control of high-low oil or gas burners. Recommended also for stokers. Type 855 is designed for direct line voltage applications recommended for unit heaters, air conditioning, etc.

LOW WATER CONTROLS



Available also as a combined low water and pressure control to prevent firing into a dry boiler or building up excess pressure. There are a number of different types for various requirements. The illustration shows type provided with quick-hook-up fittings designed in accordance with the *A.S.M.E.* code.

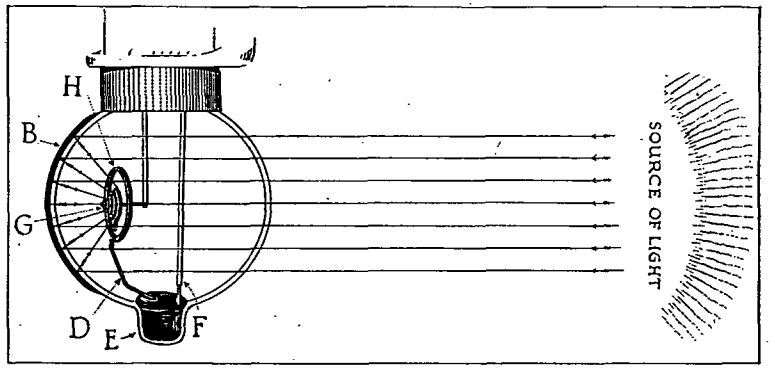
STOKER TIMER CONTROLS

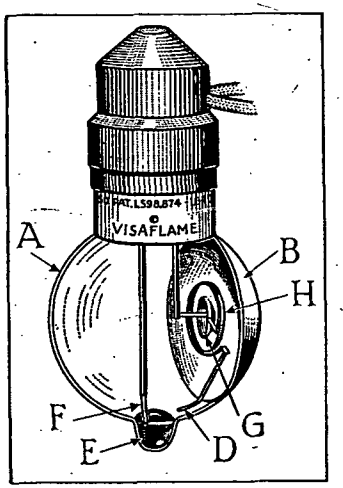
Type TV2 Stok-A-Timer combines a Mercoid Transformer-Relay and a synchronous motor timer mechanism for maintaining the stoker fire during periods when thermostat is not calling for heat. Interval adjustment can be set for ½ hour or 1 hour merely by moving a lever.

MERCOID VISAFLAME CONTROL SYSTEM

(Actuated by Light)

FOR DOMESTIC AND INDUSTRIAL OIL BURNER APPLICATIONS





The Visaflame is a departure from the usual oil burner control operation. Instead of having a safety control on the stack, actuated by the heat from combustion, the Visaflame is located within or adjacent to the oil burner where it uses the light of the flame as a means of actuating the control system. This permits a more direct and positive method for protection against flame failure, power failure, or low line voltage.

The Visaflame System lends itself to streamline burner design as it can be built into the burner unit by the manufacturer, thereby making it possible to test the complete burner with safety device at the factory which also simplifies and reduces the cost of installation in the field.

MANNER OF OPERATION

The Visaflame, when properly focused, closes its circuit in the presence of light and opens its circuit when the light is absent. The light of the flame passes through bulb "A" and is picked up by the concave reflector "B" which concentrates the light waves on the small bimetal coil "G." This bimetal coil is opaque and therefore transmutes the light waves into heat, which causes it to move the attached electrode "D" into the pool of mercury "E." The electric circuit is then closed between the fixed electrode "F" and the movable electrode "D." The outer bimetal coil "H" to which the movable electrode is attached, compensates for varying ambient temperatures.

The Visaflame is available for Intermittent Ignition or for Constant Ignition oil burners. Also for manually operated industrial burners. Adaptable to high-low flame operation.

Minneapolis-Honeywell Regulator Company

2958 Fourth Ave., So., Minneapolis 8, Minn. Cable Address: MINNREG, MINNEAPOLIS

Electric or Pneumatic Control Systems for Heating, Ventilating, Air Conditioning

BROWN INSTRUMENTS for Indicating, Recording, Controlling

Factories: MINNEAPOLIS, MINN., PHILADELPHIA, PA., WABASH, IND., CHICAGO, ILL.

Branch Offices or Distributors are located in all principal cities

ALBUQUERQUE, N. M.	CLEVELAND, OHIO	HOUSTON, TEXAS	OMAHA, NEB.	SALT LAKE CITY, UTAH
ALLENTOWN, PA.	COLUMBUS, OHIO	INDIANAPOLIS, IND.	PEORIA, ILL.	SAN FRANCISCO, CAL.
ATLANTA, GA.	DALLAS, TEXAS	JACKSON, MICH.	PHILADELPHIA, PA.	SEATTLE, WASH.
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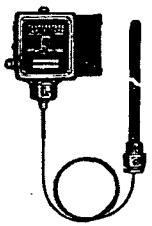
In Europe: LONDON, ENGLAND; STOCKHOLM, SWEDEN; AMSTERDAM, HOLLAND

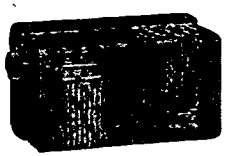


Temperature Controller

AUTOMATIC CONTROLS FOR EVERY APPLICATION

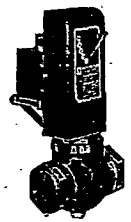
Minneapolis-Honeywell manufactures a complete line of electric, pneumatic, and self-contained controls and regulators for every type of heating, ventilating, and air conditioning installation. In addition, the Brown Instrument Division of Minneapolis-Honeywell manufactures a complete line of indicators, recorders, and controllers for Industrial Process applications.

Each of the branch offices of Minneapolis-Honeywell maintains a staff of experienced engineers who are qualified to give unbiased advice on any type of control application and to install and service control equipment of any type. They are prepared to assist in the writing of specifications and to furnish control layouts and cost estimates without charge.



Line Voltage Thermostat

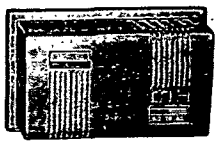
Minneapolis-Honeywell, with 55 years of experience in the control field, is the only company which is prepared to furnish every type of control, whether electric, pneumatic, or self-contained for any type of installation. This eliminates the necessity of purchasing controls from more than one company, which often results in split responsiblity and unsatisfactory results.



"Modutrol Valve"

THE MODUTROL SYSTEM OF ELECTRIC CONTROL

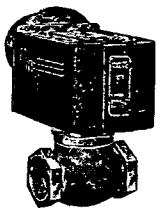
The Modutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Electric Controls or Self-contained Automatic Valves used to govern the operation of air conditioning or heating systems other than the small domestic installations. A wide variety of both modulating and two position motors, controllers and valves are available thus making the Modutrol System extremely flexible as to the selection of control equipment to produce the desired results.



"Gradustat" Pneumatic Thermostat

THE GRADUTROL SYSTEM OF PNEUMATIC CONTROL

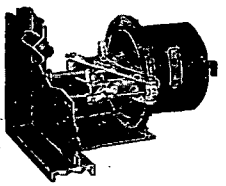
The Gradutrol System designation is applied to any combination of Minneapolis-Honeywell Automatic Pneumatic Controls used to govern the operation of air conditioning or heating systems. Such features as infinite positioning with the Gradutrol Relay and accurate graduation of valve and damper motors makes the Gradutrol System a truly remarkable advance in pneumatic control of commercial air conditioning and space heating installations.



"Grad-U-Valve" Pneumatic Control Valve

COMBINATION ELECTRIC AND PNEUMATIC SYSTEMS

The outstanding advantages of both the electric Modutrol System and pneumatic Gradutrol System of control may be combined in a single installation. Thus maximum flexibility and low installation cost are obtained. Minneapolis-Honeywell can offer either an electric or pneumatic system, or a combination of the two. This is your guarantee of an unprejudiced recommendation.

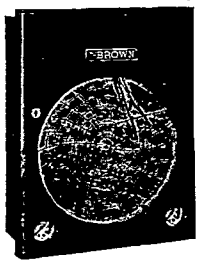


"Grad-U-Motor" Pneumatic Damper Motor

BROWN INSTRUMENTS

The extent to which air conditioning equipment is being used in office buildings, theatres, stores, industrial buildings, etc., has opened up a wide demand for indicating and recording resistance thermometers because the temperatures throughout these air conditioning systems should be checked periodically in order to obtain the best results at minimum operating cost. To obtain uniform conditions from modern equipment, it is necessary that the engineer in charge of operation have a visual picture of actual conditions.

Brown Resistance thermometers are available for indicating, recording, and controlling service and are applicable to all types of air conditioning and space heating installations.



Brown Air Operated Controller Recorder

In addition to Resistance Thermometers, The M-H Brown Instrument Division manufactures:

Thermometers	Flow Meters
Hygrometers	CO_2 Meters
Pressure Gauges	Tachometers
Vacuum Gauges	Liquid Level Gauges
Potentiometer Pyrometers	Protectoglo System

RESPONSIBILITY FOR ENTIRE CONTROL SYSTEM

Minneapolis-Honeywell Regulator Co. is equipped to assume the entire responsibility for any control installation, thereby eliminating the difficulties and misunderstandings which division of responsibility may create.

The Meriam Instrument Company

10984 Madison Avenue, **Cleveland 2, Ohio**

Representatives in Principal Cities

MANOMETERS, METERS AND GAUGES FOR ACCURATELY MEASURING PRESSURES, VACUUMS AND FLOWS OF LIQUIDS AND GASES

Clean Out U-Type Manometer

U-TYPE MANOMETERS

For measuring pressure, vacuum or differential from only a few ounces to many pounds.

In the Clean Out design (patented) the Pyrex U-tube and housing can be disconnected from the head and cleaned without disconnecting the piping. Available in 8 sizes from 6 in. to 50 in. for pressures up to 100 lb per sq in.

Gland Packed type available for 250 lb per sq in. *Write for Bulletin 1.*

"H" Type Flow Meter

FLOW METERS

The H-Type Flow Meter—Illustrated at the left is a direct reading well type manometer for measuring the flows of liquids, gases and steam across calibrated plates inserted in the line. The single straight Pyrex glass tube is gland packed top and bottom. Overflow well prevents indicating fluid from being blown into the line if the differential pressure should exceed the meter's range. Plain or flush type mounting for line pressures up to 250 lb per sq in. *Send for Bulletin 18.*

6 In. Mercury Pressure Gauge

MERCURY PRESSURE GAUGES

For measuring low range air and gas pressures. Gauges are direct reading and so designed that the mercury can not be lost from the gauge either through spilling or overpressuring.

Available in 3 and 6 in. sizes to measure 22 oz and 48 oz respectively. Unit scale graduations are in ounces. Each gauge is furnished complete with pet cock and necessary mercury. *Write for Bulletin No. 3.*

A-203 Table Mounting Well Type Manometers

A-275 Wall Mounting

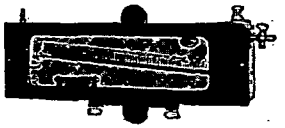
WELL TYPE MANOMETERS

For measuring pressure and vacuum, and differential directly from the scale, using water, red oil or mercury, without adding distances above and below reference zero.

Model A-203—Furnished in wall, table and flush front mounting styles. Ranges 30 in. to 100 in. for pressures up to 250 lb per sq in.

Model A-275—Furnished in wall mounting and flush front styles only. Ranges from 6 in. to 24 in. for pressures up to 150 lb per sq in. *Send for Bulletin B.*

DRAFT GAUGES



Single Tube Draft Gauge

Model GP Draft Gauges take accurate readings of low pressures, vacuums and differentials in fractional inches of water. Furnished in single tube, also in 2-tube and 3-tube models for taking multiple readings on the same gauge. *Send for Bulletin No. 4.*

Gauge Pulsation Absorber

GAUGE PULSATION ABSORBERS

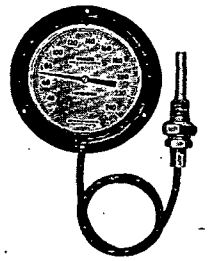
The Gauge Pulsation Absorber provides a simple, effective method of preventing "hammer" and pulsing flow in a line from affecting the accuracy of gauge readings, and protects the gauge from damage. *Send for Bulletin No. 2.*

Moeller Instrument Co.

132nd St. & 89th Ave., **Richmond Hill 18, New York**

Representatives in Principal Cities

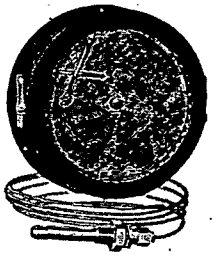
Manufacturers of Indicating and Recording Thermometers, Psychrometers, "U" Gauges, Mercury Vacuum Gauges, Mercury Barometers, Hydrometers, Engraved Stem Thermometers, etc.



MOELLER DIAL THERMOMETERS

Mercury filled. Made in several sizes and in standard ranges from minus 40 to plus 1000 F. Either in the rigid connection or with flexible capillary tubing. All kinds of connections.

INDUSTRIAL THERMOMETERS Red reading columns, in V shaped case with glass front in all forms, sizes, and connections.

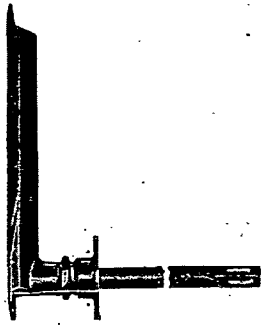


INDICATING AND RECORDING PSYCHROMETERS

WRITE FOR CATALOGUE

MOELLER RECORDING THERMOMETERS Mercury filled, in chart ranges from minus 40 to plus 1000 F. Made in 10 in. and 12 in. uniform chart graduations. All kinds of connections. Square or round cases.

AIR DUCT THERMOMETERS For hot and cold air ducts, or designed for both cold and hot ducts. Made in several sizes. Various stem lengths with union connection and flanges for insulated and non-insulated ducts.



The Palmer Company

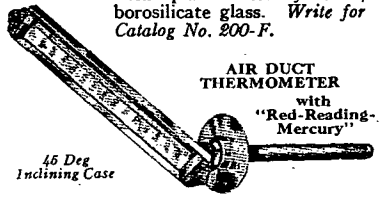
Main Plant: 2506 Norwood Ave., **Cincinnati (Norwood), Ohio**

Canadian Factory: King and George Sts., Toronto

Manufacturers and Originators—"Red-Reading-Mercury" Thermometers

"RED-READING-MERCURY"

Industrial Thermometers—These mercury tubes will show a bright RED color, visible at a great distance. The color is reflected and cannot fade. (Patented by Palmer). Thoroughly annealed and guaranteed permanently accurate. Costs no more. STRAIGHT, ANGLE, SIDE - ANGLE, RECLINING AND INCLINING Case, OBLIQUE STEM, etc. 7, 9 and 12 in. case, with or without glass front. Standard 3½ in. stem and longer lengths. Fittings: Fixed Thread, Union, Separable-socket and Adjustable or Union Flange. All ranges up to 750 F or 400 C. For ranges up to 1000 F or 550 C, with plain mercury tube, borosilicate glass. *Write for Catalog No. 200-F.*

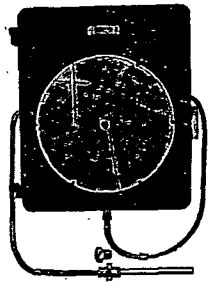


45 Deg. Inclining Case

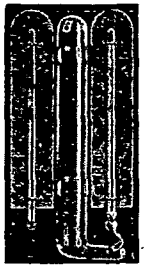
AIR DUCT THERMOMETER with "Red-Reading-Mercury"

9 in. case, Flanged Union Connection, standard stem length extending into Air Duct 12 in. and 18 in. Range +30 and +160 F. Easy to install.

RECORDING THERMOMETERS



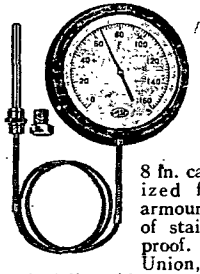
Mercury Actuated. 12 in. die-cast aluminum case. Wrinkle or Satin finish. All parts are rust-proof. Flexible armoured tubing and bulb of stainless-steel. Fittings: Plain, Union, Separable-socket and adjustable or union flange. All ranges up to 1000 F or 550 C. Guaranteed extremely accurate and sensitive. *Write for Cat. 45.*



WALL HYGROMETER and SLING PSYCHROMETER

Wet and Dry bulb; Mercury tube, with RED column. Chart furnished. Guaranteed sensitive and accurate. *Send for Bulletin No. 500.*

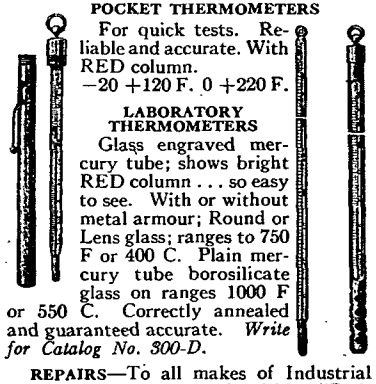
DIAL THERMOMETER



Mercury Actuated

8 in. case. Black rubberized finish. Flexible armoured tubing and bulb of stainless-steel. Rustproof. Fittings: Plain, Union, Separable-socket and Adjustable or Union flange. All ranges up to 1000 F or 550 C. Guaranteed sensitive and accurate. *Write for Cat. 46.*

POCKET THERMOMETERS



For quick tests. Reliable and accurate. With RED column.
−20 +120 F. 0 +220 F.

LABORATORY THERMOMETERS

Glass engraved mercury tube; shows bright RED column . . . so easy to see. With or without metal armour; Round or Lens glass; ranges to 750 F or 400 C. Plain mercury tube borosilicate glass on ranges 1000 F or 550 C. Correctly annealed and guaranteed accurate. *Write for Catalog No. 300-D.*

REPAIRS—To all makes of Industrial Mercury Thermometers, furnishing "Red-Reading-Mercury" tube, at no extra cost and replacing all worn or broken parts, making the thermometer as good as new. Guaranteed accurate.

Penn Electric Switch Co.

Goshen, Indiana

Offices

ATLANTA; BOSTON, NEWTON, MASS.; CHICAGO; DALLAS; DAYTON; DETROIT; MILWAUKEE; MOLINE, ILL.; NEW YORK; PHILADELPHIA; ST. LOUIS

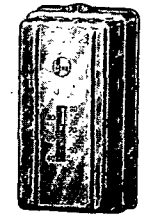
Export—13 East 40th St., NEW YORK 16, N. Y.

Representatives—GARLAND-AFFOLTER ENGRG. CORP., San Francisco, Los Angeles, Seattle, Portland; SPECIALTY SALES CO., Salt Lake City; FORSLUND PUMP AND MACHINERY CO., Kansas City; VINCENT BRASS AND COPPER CO., INC., Minneapolis; H. M. OLMSTEAD, Denver

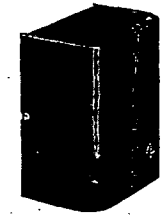
In Canada—POWERLITE DEVICES, LTD., PENN ELECTRIC SWITCH DIV., Toronto, Ont.

Distributors and Jobbers in All Principal Cities

Automatic Controls for Heating, Refrigeration, Air Conditioning, Engines, Pumps and Air Compressors



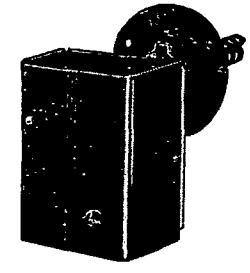
Thermostats



Stoker Timers

HEATING CONTROLS

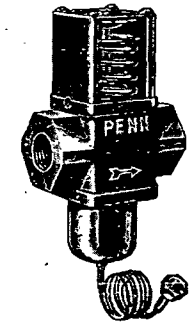
The Penn line of controls includes models for every type of heating service on steam, vapor, hot water or warm air systems whether gas, oil or coal fired. In this complete line there are: Low and line voltage Room Thermostats—Stoker Timers—Oil Burner Stack Switches—Hot Water and Warm Air Temperature Controls—Vapor Controls—Steam Pressure Controls—Relays—Humidistats—Day-Nite Tem Clocks—Damper Motor Controls—Boiler Water Feeders—Boiler Water Level Controls—and Solenoid Valves. Condensed description and specifications are given in Bulletin 1508-K. *Write for your free copy.*



Oil Burner Stack Switches

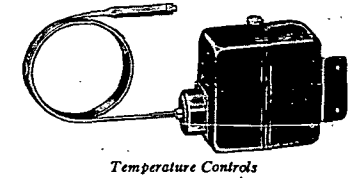
REFRIGERATION CONTROLS

Whatever your need may be in refrigeration controls—choose the exact type to fit the job from Penn's complete line of automatic controls. Pressure and temperature controls are available with or without calibrated adjustments for single phase or polyphase service. Also available are: Cooling Room Thermostats—Humidistats—Relays—Solenoid Valves—and Water Regulating Valves. All of these controls are described and illustrated in condensed Bulletin 1487-M. *Write for your free copy.*

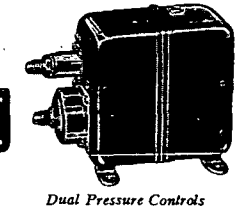


Water Valves

Write for catalog on Penn Controls to cover your particular applications.



Temperature Controls



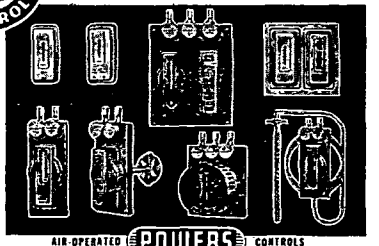
Dual Pressure Controls

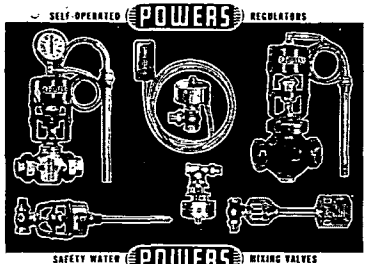
THE POWERS REGULATOR CO.

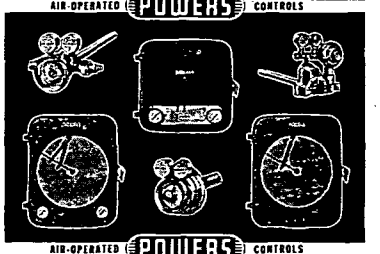
Over 50 Years of Temperature and Humidity Control—Offices in 47 Cities

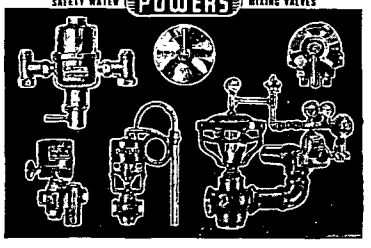
NEW YORK 17, N. Y., 231 East 46th St.—**Chicago 14,** 2719 Greenview Ave.—LOS ANGELES 5, 1808 W. Eighth St.—BOSTON 15, 125 St. Botolph St.—PHILADELPHIA 32, 2240 N. Broad St.—GREENSBORO, Jefferson-Standard Bldg.—ATLANTA 3, Bona Allen Bldg. CLEVELAND 13, 2012 West 25th St.—KANSAS CITY 6, 409 East 13th St.—SEATTLE 4, DETROIT 2, Boulevard Bldg.—CINCINNATI 2, American Bldg.—NEW ORLEANS 12, Balter Bldg.—ST. LOUIS 3, 2726 Locust St. 500 First Ave., S.—DALLAS 1, 2438 Brandon St.

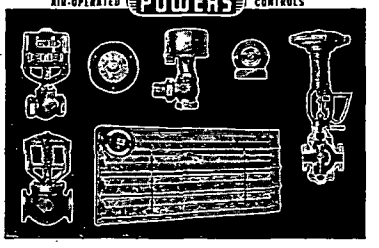
A very complete line of Temperature, Humidity, Pressure — indicating, controlling, and recording regulators — for heating and air conditioning systems, industrial processes and all types of hot water heaters.











Over 50 years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings have given us a wealth of experience from which you can draw in selecting the proper type of control for any purpose.

Catalogs and Bulletins describing any or all of our products furnished upon request. Phone or write our nearest office. See your phone directory.

Spence Engineering Company, Inc.

28 Grant Street, **Walden, N. Y.**

SPENCE METAL DIAPHRAGM "DEAD END" REGULATORS
Advantages of Spence Regulators

Dead-end Shutoff—Spence Regulators are guaranteed to hold a dead-end.

Single Seat—Spence design makes possible a balanced single seat even in large sizes.

Metal Diaphragms—Under normal conditions never require replacement.

Accurate Regulation—Regardless of fluctuations in either load or initial pressure.

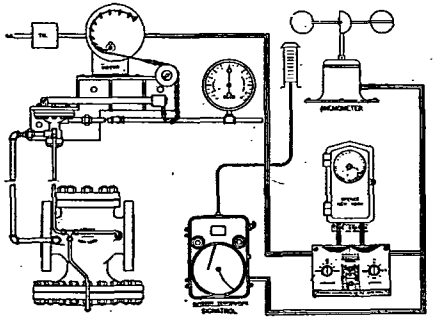
SECO Metal—Guaranteed to resist the wiredrawing action of steam.

Interchangeable Pilots—Any type of pilot will fit any size main valve.

Accessibility—Pilot is connected to main valve with unions.

No Stuffing Boxes—All main valves and most pilots are packless.

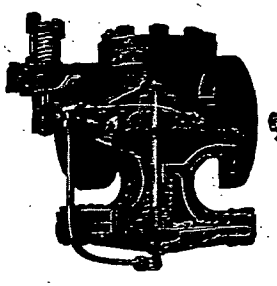
Spence Weather Compensator and Orifice Zone Control System



This simple, dependable Control, when installed on a properly designed orificed heating system, will show a substantial degree-day steam saving, at a low maintenance cost.

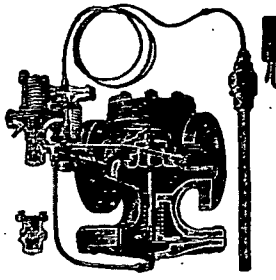
The delivery pressure of the Regulator is automatically adjusted in direct proportion to the building heat losses. In other words, as the losses become greater, steam pressure on the system is automatically increased.

Any number of zones can be controlled by one automatic Signatrol, automatic Wind Loss Compensator (Anemometer), Time Switch and Master Control Panel equipped with Manual and Automatic Dials for each zone. In this way each zone can be set individually and at the same time be under the Master Control.



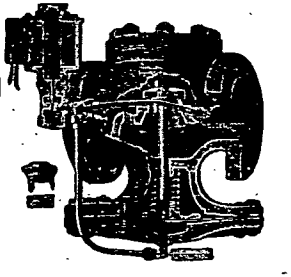
Pressure Regulator—Type ED

Designed to regulate a steady or varying initial pressure so as to maintain a constant, adjustable, delivery pressure. Applicable to heating systems, power plant operations, or manufacturing processes.



Combined Temperature and Pressure Regulator—Type ETD

Self-contained, pilot operated, dead-end. Designed to control flow of fluid to a heating or cooling element, so as to maintain a constant, adjustable temperature, and protect the element against excessive pressure.



Electrically Operated Valve—Type EM

Can be opened or closed independently by an electrical switch.

Type ET—Same as ETD except pressure control is omitted.

Order a SPENCE Regulator for 40 days' free trial.

Fall-O-Matic Universal Pipe Intersection Cutter.

Taylor Instrument Companies

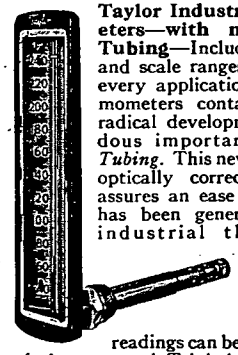
Rochester, N. Y., U. S. A.

IN CANADA—TAYLOR INSTRUMENT COMPANIES OF CANADA, LTD., TORONTO

NEW YORK CHICAGO BOSTON PHILADELPHIA LOS ANGELES PITTSBURGH CLEVELAND ST. LOUIS BALTIMORE SAN FRANCISCO CINCINNATI TULSA DETROIT ATLANTA MINNEAPOLIS WILMINGTON

Manufacturing Distributors in Great Britain, Short & Mason, Ltd. London

Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure, Humidity, Flow and Liquid Level

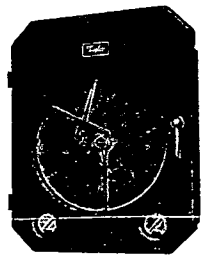


Taylor Industrial Thermometers—with new "BINOC" Tubing—Includes many styles and scale ranges with bulbs for every application. These thermometers contain a new and radical development of tremendous importance—*"BINOC" Tubing*. This newly designed and optically correct glass tubing assures an ease of reading that has been generally lacking in industrial thermometers. *"BINOC" Tubing* more than doubles the angle of vision within which readings can be made. Because of the patented Triple-lens construction its broad mercury column can be read easily and accurately with both eyes. Bore reflection is absent.

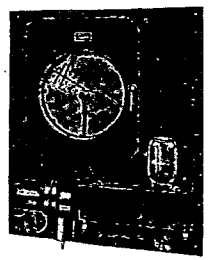
Taylor "BINOC" Pocket Test Thermometer—Ideal for frequent testing of important temperatures. Taylor patented *"BINOC" Tubing* eliminates juggling and guesswork. High accuracy—Easier to Read.

The New Taylor "Fulscope" Recording Controller—An air-operated controller that gives practically any character of process control regardless of *time lag* in apparatus.

Available for controlling temperature, pressure, humidity, rate of flow, liquid level. Where extreme load changes or badly balanced operating conditions exist, precision control can be maintained by the automatic reset feature. For applications where a record is not needed, Taylor supplies an Indicating "Fulscope" Controller.



Taylor Biram's Anemometer—Ideal for measuring air velocities with the fan revolutions indicated on the dial. Various models for a wide range of air speeds and registration limits.



Taylor Recording Hygrometer—Records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.

Type shown has motor-driven fan for conditioned rooms or passages where circulation is poor. Furnished without fan for installations where circulation across bulb is good.

Taylor Sling Psychrometer—The advantage of this form of Wet- and Dry-Bulb Hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as the whirling bulbs are subjected to perfect circulation. Two accurate etched stem thermometers are mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened.

These thermometers have scales of 20 to 120 degree F, graduated in ½-deg divisions. A copper case protects the tubes when not in use.

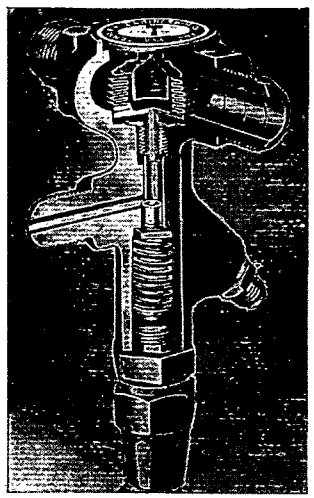
Taylor also offers a complete line of the famous Taylor Recording and Dial Thermometers, Self-Acting and Type "P" Controllers, the 10-BG Hygrometer and many types of Humidiguides.

Tenney Engineering, Inc.

26 Avenue B — Newark 5, New Jersey

Telephone Bigelow 8-3905

Manufacturers of Automatic Temperature, Humidity and Pressure Control Equipment



HALF ACTUAL SIZE Model TS-1 TENNEY Thermostatic Expansion Valve

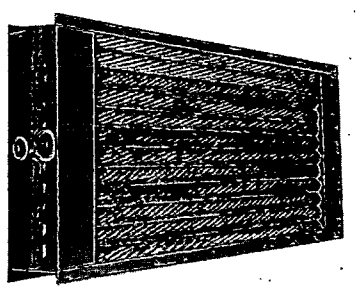
New TENNEY Thermostatic Expansion VALVE

Has no feeler bulb. Unaffected by ambient temperature, entering warm air or warm suction lines. Responds instantly to changes in suction vapor conditions. Special power element in the Valve design facilitates complete closing above a definite evaporative pressure. Design assures positive action—no sticking or binding. Push pin packing not required. Valve can operate in any position. Outstanding features stamp it as first forward improvement in expansion valve design in 20 years. Particularly adaptable to modern evaporators with forced air, small tubes, short passes and distributor header combinations. Ideal for small evaporators or modern close coupled coil and machine combinations.

TENNEY Feather Fin Coils

for COOLING and HEATING
DIRECT EXPANSION STEAM HEATING
WATER COOLING and HEATING

Adequately meets modern demand for equipment that gives high operating performance and efficiency. Low air resistance. Simplified component parts. Ratio of extended fin surface carefully engineered to provide best operating results. Easy to mount and install. All joints copper welded. Design embodies distributing header refinements.



TENNEY Feather Fin Coil

Send for New Valve Bulletin or Feather Fin Coils Catalog, prices and discounts.

White-Rodgers Electric Company

1293 Cass Avenue, **St.Louis, Mo.**

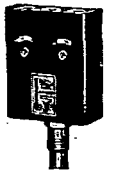
Distributors in Principal Cities

HYDRAULIC-ACTION CONTROLS

Hydraulic-Action temperature controls combine the powerful uniform expansion and contraction of a solid liquid charged stainless steel diaphragm with mechanical simplicity to provide accuracy and dependability at all times.

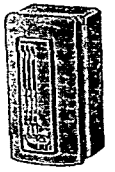
Take advantage of **Hydraulic-Action** on your next installation. Specify White-Rodgers Controls. The latest condensed control catalog is awaiting your request. Write for it today!

Single speed fan control-cover removed showing visible dial.

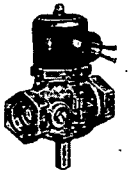


Dual Immersion Control—Limit-Circulator or Summer-Winter service.

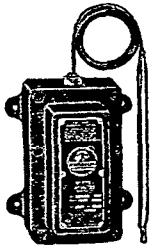
Line voltage Thermostat for Unit Heater and Air-Conditioning Installations.



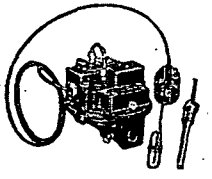
Low Voltage Room Thermostat—anticipating type.



Solenoid Gas Valve—High plunger torque and silent operation.



Explosion-Proof Thermostat for hazardous locations—remote type.



Diaphragm Gas Valve with puff bleed and built-in mechanical limit control.

Steam Pressure Control—for safety limit service.

8 EXCLUSIVE FEATURES OF WHITE-RODGERS HYDRAULIC-ACTION TEMPERATURE CONTROLS

1. May be mounted at any angle or position, above, below or on level with control point.

2. Hydraulic-Action Principle incorporating solid-liquid filled bulb and capillary provides expansion force comparable to that of a metal bar.

3. Diaphragm motion uniform per degree of temperature change.

4. Power of solid-liquid charge permits unusually sturdy switch construction resulting in positive contact closure.

5. Heavier, longer-wearing parts are possible because of unlimited power.

6. Dials are evenly and accurately calibrated over their entire range because of straight-line expansion.

7. Controls with remote bulb and capillary are not sensitive to change in room temperature. Accuracy of control is not affected by temperature changes in surrounding area.

8. Not affected by atmospheric pressure. Works accurately at sea level or in the stratosphere without compensation or adjustment.

International Heating & Ventilating Exposition

THE AIR CONDITIONING EXPOSITION

Permanent Address—Grand Central Palace, **New York 17, N. Y.**

EXPOSITIONS HELD

The first in Philadelphia, 1930.
The second in Cleveland, 1932.
The third in New York, 1934.
The fourth in Chicago, 1936.
The fifth in New York, 1938.
The sixth in Cleveland, 1940.

SEVENTH EXPOSITION TO BE HELD IN CLEVELAND

The first post-war Heating and Ventilating Exposition will be held in Cleveland at Lakeside Hall, January 27 to 31, 1947. This will be the first of a new series, now scheduled on an annual basis. Response to the initial announcement indicates that manufacturers have a keen interest in exhibiting their newest products of research and development.

There has developed a tremendous pent-up demand for new equipment that will be a challenge to the industry. The manner in which you meet this opportunity will be a guide to your future progress. As in the past, the Exposition will be ready to help you by offering a timely, quick and effective means of renewing old contacts, developing new ones, and actually demonstrating your products.

These expositions have been and will be held coincident with the Annual meetings of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and are directed by the International Exposition Company, under the auspices of the A.S.H.V.E.

EXHIBITORS

Comprise leading firms in each phase of the industry; number has varied from 150 to 327 exhibitors.

EXHIBITS

These range from and comprise all the types of articles discussed or advertised in this copy of THE A.S.H.V.E. 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 and 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, 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.

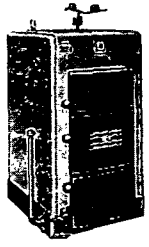
VISITOR ATTENDANCE

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. In 1940 at Cleveland, there were 20,652 registered visitors.

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.

AMERICAN RADIATOR & Standard Sanitary

New York CORPORATION Pittsburgh

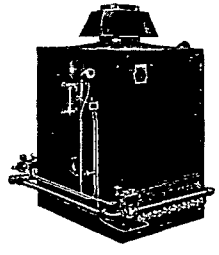


SEVERN BOILER FOR COAL (stoker or hand-fired), or OIL

An exceptionally efficient Boiler with many new features for convenience and economy. Ratings: Steam—350 to 780 sq ft, Water—560 to 1250 sq ft, installed radiation.

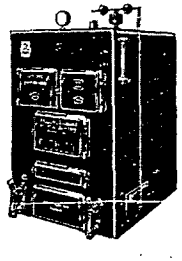
EMPIRE GAS BOILER

Designed by experts to burn gas efficiently, economically. All controls concealed. Ratings: Steam—163 to 1097 sq ft, Water—135 to 1755 sq ft, installed radiation.



STANDARD GAS BOILER

Basically the same as the Empire Gas Boiler shown above but without jacket. Ratings: Steam — 400 to 11,905 sq ft, Water — 135 to 19,048 sq ft, installed radiation.

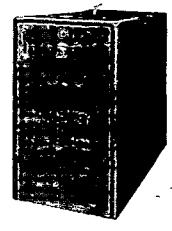


REDFLASH BOILERS—For All Fuels

Economical heat for any size or kind of building. Attractive red jacket, fully insulated. Ratings: Steam—770 to 9900 sq ft, Water—1230 to 15,840 sq ft, installed radiation.

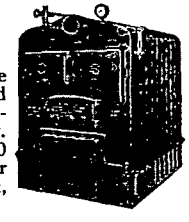
ARCOLINER FOR COAL (stoker or hand-fired) or OIL

An attractive Boiler of advanced design for heating smaller homes inexpensively and well. Ratings: Steam—180 to 460 sq ft, Water—290 to 740 sq ft, installed radiation.



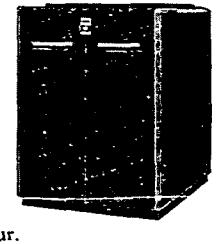
OAKMONT OIL BOILER

A highly efficient moderate priced Boiler for small homes. Also supplied as complete boiler-burner unit with Arcoflame Burner. Ratings: Steam—390 to 810 sq ft, Water—625 to 1295 sq ft, installed radiation.



WATER TUBE BOILERS Coal (Hand or stoker fired) or Oil

For medium to large size buildings. Noted for efficient performance and economy. Ratings: Steam—400 to 5,200 sq ft, Water—640 to 8,320 sq ft, installed radiation.



MOHAWK Winter Air Conditioner Gas-Fired

Provides complete, automatic gas-fired Winter Air Conditioning in small, medium or large sized homes. Capacities range from 43,200 to 216,000 Btu input per hour.

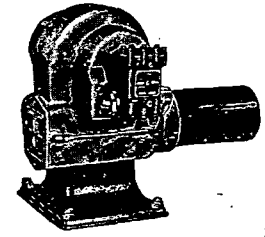
AMERICAN RADIATOR & Standard Sanitary

New York CORPORATION Pittsburgh

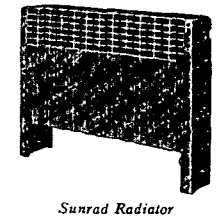
YOU CAN FILL EVERY HEATING NEED FROM THIS COMPLETE LINE

From American - Standard come Boilers of all types and sizes for Coal—stoker or hand-fired—Oil, or Gas, and a full line of Radiators, Convectors, Oil Burners, Domestic Water Heaters, and Accessories—all backed by the undivided responsibility of one of the world's largest manufacturers of Heating and Plumbing Equipment.

ARCOFLAME OIL BURNERS



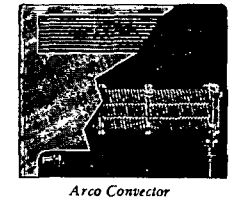
The Model "C" Arcoflame has a capacity of up to 3 gallons per hour. The Model "L" (not shown) from 3 to 7 gallons per hour. Both embody unusual and highly efficient features.



Sunrad Radiator

SUNRAD RADIATOR

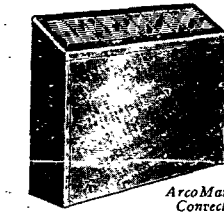
Installed recessed or free-standing. Requires no enclosure. Two sizes: 20 in. high x 5 in. wide = 2 sq ft, and 23 in. x 7½ in. = 3 sq ft.



Arco Convector

ARCO CONVECTOR

For convection heating at its best. Available in four widths and in virtually any desired length.



Arco Marine Convector

ARCO MARINE CONVECTOR

Especially designed for Marine use. Non-ferrous. Highly efficient. For all systems except one-pipe steam.

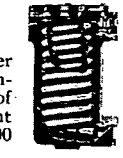
ARCO RADIATOR

The modern, slim type radiator that occupies less space and gives more heat. It comes in four narrow widths and in four heights —19, 22, 25 and 32 inches.

Arco Radiator

EXCELSO INDIRECT WATER HEATER

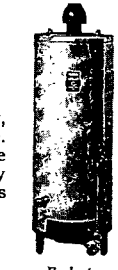
Provides domestic hot water during heating season without care and expense of separate heater. Efficient and economical. 30 to 2500 gallon capacities.



Excelso

BUDGET GAS FIRED WATER HEATER

Heats water automatically, stores it for instant use. Jacket finished in white enamel, black trim. Thrifty and dependable. Three sizes —20, 30, and 40 gallons.



Budget Gas Fired

No. 861 Hurivent Valve (for mains)

No. 300 Multiport adjustable Valve (for radiators)

No. 999 Arco Packless Steam Radiator Valve

THE BABCOCK & WILCOX COMPANY

85 Liberty Street — **New York 6, N. Y.**

Manufacturers of

Water-Tube Boilers — **Chain-Grate Stokers**

Oil Burners — **Seamless Steel Tubing and Pipe**

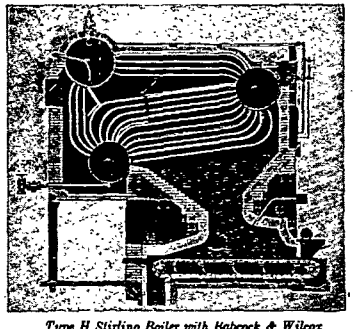
Branch Offices and Representatives in all Principal Cities

Type H Stirling Boiler

The Babcock & Wilcox Type H Stirling Boiler is a highly efficient unit built for moderate pressures at moderate prices.... and is designed to occupy minimum floor space and head room for the heating surface required.

This boiler is built in four classes and 36 sizes ranging from 691 to 6225 sq ft of heating surface, and can be designed for operation with any fuel and every method of firing.

The moderate price is due only to the simplicity of design, efficient production methods and superior shop equipment.



Type H Stirling Boiler with Babcock & Wilcox Chain-Grate Stoker

ADVANTAGES OF THE BABCOCK & WILCOX TYPE H STIRLING BOILER:

Unusual steaming capacity for the floor space and head-room required.

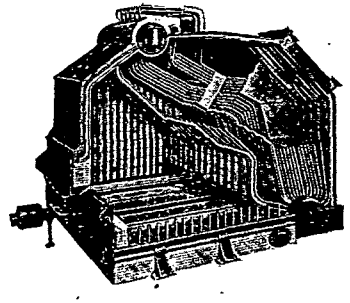
The choice of three locations for gas exit reduces cost of flues and breeching.

Distribution baffles make effective all of the heating surface.

Tube renewal is facilitated by correct tube spacing, and a tube removal door.

The boiler is supported by a structural-steel framework entirely independent of the brickwork.

A complete table of sizes and dimensions will be sent upon request. Simply ask for Bulletin G-8-C.



B&W Integral-Furnace Boiler, Type FF

Many of the advantageous features incorporated in large B&W central-station boilers are now available for the first time in the B&W Integral-Furnace Boiler, Type FF, which is offered in sizes ranging from 1353 to 6506 sq ft heating surface.

DISTINGUISHING FEATURES INCLUDE:

A completely water-cooled furnace. The construction provides water cooling for front and rear (or bridge) walls, as well as side walls and roof.

A furnace arrangement in which the primary combustion zone is followed by an open pass, thus making use of a principle of combustion that was first developed and used successfully in the B&W Open-Pass Boiler for central stations. This design insures mixing of the gases while at high temperatures, thereby aiding efficient and smokeless combustion.

Cyclone Steam Separators, which provide dry steam at high boiler-water concentrations independently of normal variations in water level, and increase circulation by eliminating steam from the water.

These, with related features, result in a boiler that is outstanding for economy of fuel and maintenance and for ease of operation. Write for Bulletin G-34.

The Bigelow Company

105 River Street, New Haven 3, Connecticut

MANUFACTURERS OF

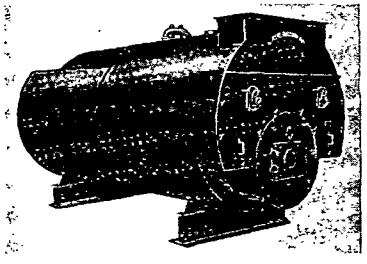
FIRE TUBE BOILERS — **WATER TUBE BOILERS**

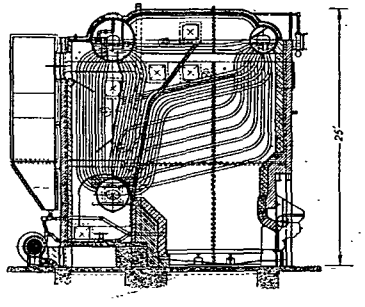
SALES REPRESENTATIVES IN PRINCIPAL CITIES

Catalog Furnished on Request

BIGELOW SCOTCH TYPE HEATING BOILER

A compact, self contained, highly efficient oil fired boiler that requires low head room and has low water line. All parts internally and externally are readily accessible. The boiler is of welded construction and contains no water leg or stay-bolted surfaces. Built in units with steam rating from 1,820 to 42,500 sq ft.



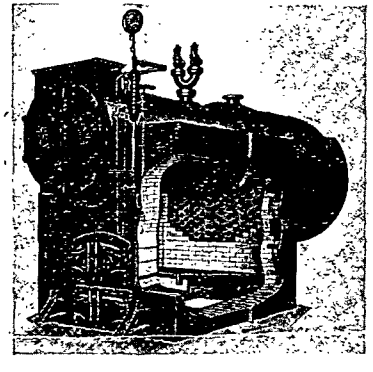


BIGELOW TYPE F WATER TUBE BOILER

Cut shows boiler equipped with spreader type stoker. The flexibility of design permits oil and other type stoker firing. Partial water wall surface is provided without extra headers, downcomers and circulating tubes. The Type F Steam Generating Unit is built in sizes ranging from 10,000 to 60,000 lb of steam per hour.

BIGELOW TWO-PASS BOILER

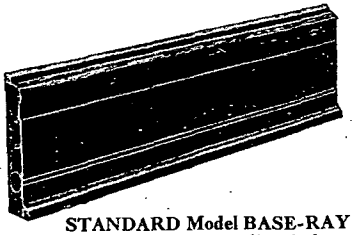
This boiler is designed to meet heating and power requirements, especially where space limitations prevail. The unit may be oil, stoker or hand fired. The elimination of special brick shapes and stay-bolts reduces the cost of maintenance to a minimum. The Two-Pass boiler is built in standard sizes from 25 hp to 250 hp.



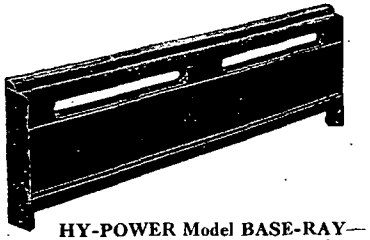
Burnham Boiler Corporation

Irvington-on-Hudson, N. Y.—Zanesville, Ohio

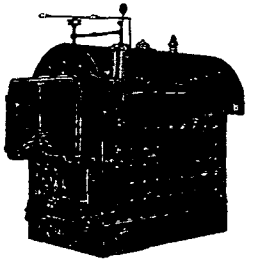
There's a Burnham for Every Purpose—**Catalogs Sent on Request**



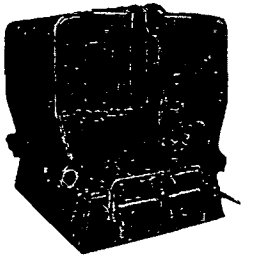
STANDARD Model BASE-RAY—Ratings—1.25 sq ft per lineal foot. Tappings—¾ in. at bottom only of both end panels. Sections are 7 in. high, 1¾ in. thick—come in 12 and 24 in. lengths.



HY-POWER Model BASE-RAY—Ratings—2.08 sq ft per lineal foot. Tappings—¾ in. at bottom only of both end panels. Sections are 7 in. high, 2 in. thick and 24 in. long.

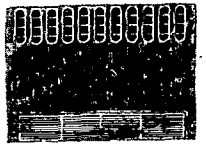


Welded Steel Boiler—Capacities from 2500 to 35,000 sq ft for Steam and 4800 to 56,000 sq ft for Water. Furnished for coal, oil or stoker firing.



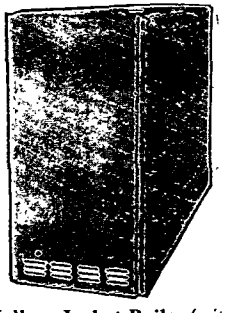
50 In. Twin-Section—4500 to 14,600 sq ft for Steam and 7200 to 23,360 sq ft for Water.

No. 1, 2, 3 and 36 in. Series—All Fuel. 230 to 4920 sq ft Steam and 370 to 7880 for Water.



Burnham Radiant Radiator—Two heights. 20 and 23 in.

Burnham Slenderized Radiators (not shown) made in three to six tubes in all heights from 19 in. to 33 in.



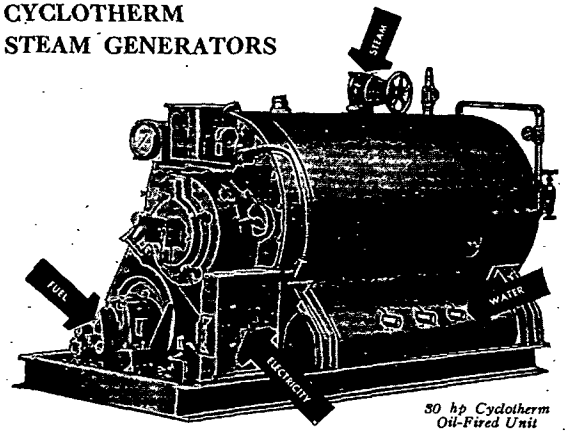
Yellow-Jacket Boiler (with extended Jacket)—All Fuel Convertible. 305 to 935 sq ft for Steam and 490 to 1495 sq ft for Water.

Cyclotherm Corporation

90 Broad Street, **New York 4, N. Y.**

CYCLOTHERM STEAM GENERATORS

Arrows indicate the 4 connections necessary to put a Cyclotherm into operation.



80 hp Cyclotherm Oil-Fired Unit

Wide range of models in sizes from 5 to 200 hp with operating pressures from 15 to 200 psi.

COMPACT—require minimum of floor and head space.

SIMPLE TO INSTALL—No expensive stack required, just connect fuel, water, electricity and steam outlet. Requires no special base.

COMPLETELY AUTOMATIC — for efficient and economical operation.

BURNS FUEL OIL or GAS—also available with combination oil and gas burner.

FUELS

Up to 60 hp No. 3 oil or lighter. 80 to 200 hp No. 6 oil or lighter. Natural or manufactured gas.

All sizes can be fired by natural or manufactured gas when equipped with combination burners, firing either oil or gas.

AUTOMATIC CONTROLS

10 to 80 hp—"on and off" burner operation.

100 to 200 hp—uniform flame modulation from 30 per cent load.

Automatically protected against low water or flame failure and motor overloads.

Firetube construction conforms to *ASME* Boiler Code requirements throughout. Will meet boiler code requirements of any city, state or federal regulations.

STANDARD EQUIPMENT INCLUDES:

Steam valves

Feed water injector

Condensate return and storage tank

Make-up valve

Motor-driven feed water pump

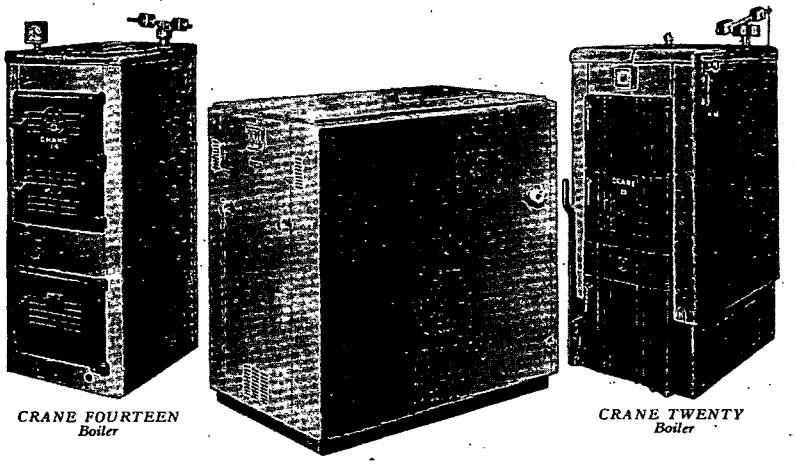
Write, wire or phone for more complete information.
CYCLOTHERM CORPORATION, 90 Broad Street, New York 4, N. Y.

Crane Co.

BOILERS, RADIATORS, FURNACES, VALVES, FITTINGS, PIPE, STEAM SPECIALTIES, PLUMBING AND HEATING MATERIALS

General Offices: 836 South Michigan Avenue, **Chicago 5, Illinois**

Nation-Wide Service Through Branches, Wholesalers, Plumbing and Heating Contractors



CRANE FOURTEEN Boiler

CRANE SIXTEEN Boiler

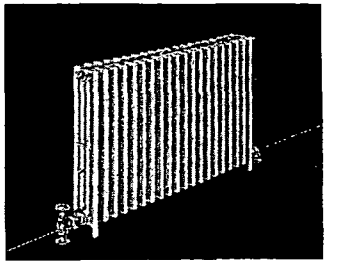
CRANE TWENTY Boiler

These three boilers from the complete Crane line are now available. They offer new design and the latest in engineering achievement, assuring the utmost in heating comfort with even greater heating economy.

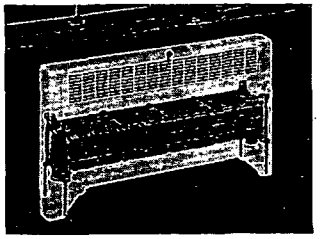
And in the complete Crane line you will also find quality equipment to satisfy every heating requirement—steam, hot water or warm air—for all fuels—hand fired or fully automatic boilers for homes, public buildings and industrial plants—warm air furnaces for all types of installations. The Crane line also includes oil burners, stokers, radiators, controls, water specialties and all necessary piping.

For complete information on Crane heating equipment, consult your Crane Branch or Crane Wholesaler.

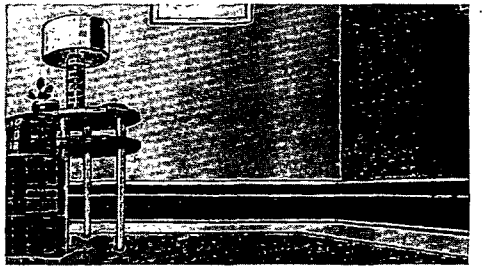
CRANE RADIATORS AND CONVECTORS



COMPAC RADIATORS: Slender in line—modern in design— constructed to give more heat and occupy less space. For free-standing or recessed installation.

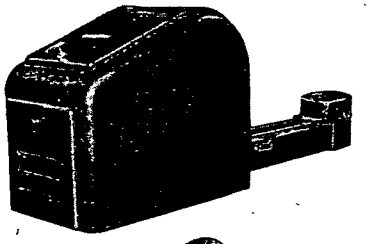


CONVECTORS: For attractive installations that harmonize perfectly with room decorations. Flush, extended or free-standing types for new construction or remodeling.



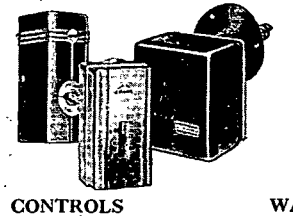
RADIANT BASEBOARD HEATING

Something new in heating! Radiant Baseboard Heating, developed by Crane Co. Steam and hot water are circulated through the baseboard panel—no radiators or grills necessary.



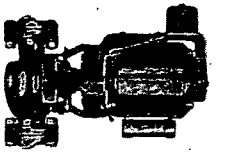
STOKERS

Newly designed and engineered stokers give greater economical heating comfort. They are made in a wide range of sizes and models, suitable for any type of stoker installation.



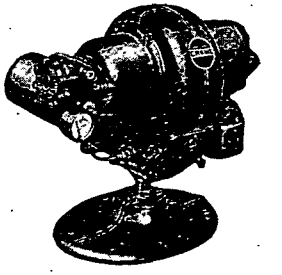
CONTROLS

The complete line of controls includes thermostats, stack switches, limit switches, aquastats, damper controls—everything necessary for every heating system.



WATER SPECIALTIES

Crane hot water specialties are made to suit every installation. They include water heaters, circulators and flow control valves.



OIL BURNERS

Newly engineered. Crane oil burners provide heat comfort at low fuel costs. Simplified design and sturdy construction reduce maintenance to a minimum.

Fitzgibbons Boiler Company, Inc.

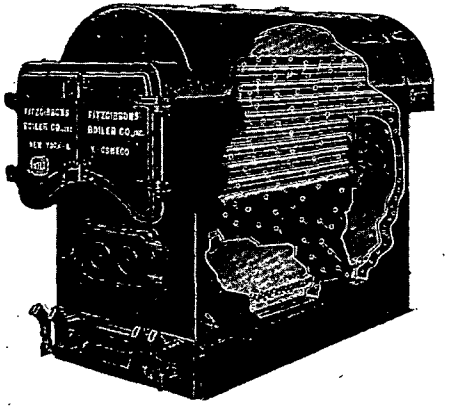
Established 1886

General Offices: Architects Bldg., 101 Park Avenue
New York 17, N. Y.

Manufactured at: OSWEGO, N. Y.
Sales Branches in Principal Cities

PRODUCTS—STEEL HEATING and POWER BOILERS for all fuels and all heating systems. Capacities to meet requirements of any building. Built and rated according to *A.S.M.E.* and *S.B.I.* Codes.—AIR CONDITIONERS for "Split-Systems" and for Direct-Fired installations in residences of all sizes.

Member, Steel Boiler Institute and Indoor Climate Institute.



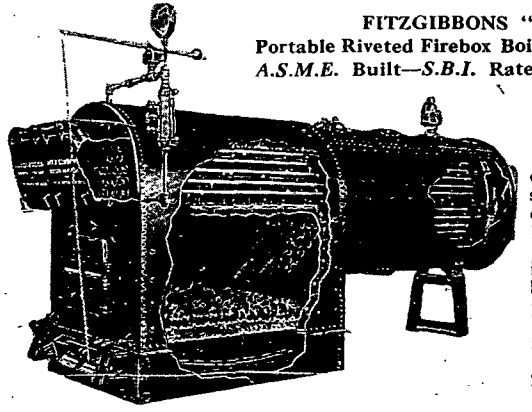
FITZGIBBONS "D" TYPE

Welded Steel Firebox Boilers —For 15 lb W.S.P.
***A.S.M.E.* Built—*S.B.I.* Rated —Hartford Inspected**

The Fitzgibbons "D" Type Boiler is of all-welded steel construction. Quick steaming, thorough combustion, long flue travel, high efficiency of heat transfer and fuel economies have won for the "D" Type a reputation for superior heating service with any fuel in any heating system. Ratings are in accordance with the new standards of the *Steel Boiler Institute*.

Oil, Gas, Stoker, 2680 to 42,500 sq ft steam.

Hand fired Coal, 2680 to 35,000 sq ft steam.



FITZGIBBONS "P" SERIES

Portable Riveted Firebox Boilers—100-150 lb W.S.P.
***A.S.M.E.* Built—*S.B.I.* Rated—Hartford Inspected**

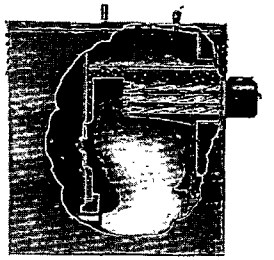
The riveted construction of the Fitzgibbons "P" series is designed for boilers to be operated up to 150 lb w.s.p.

This boiler is being used in many hospitals, and buildings requiring high pressure steam.

Oil, gas, stoker and hand fired types.

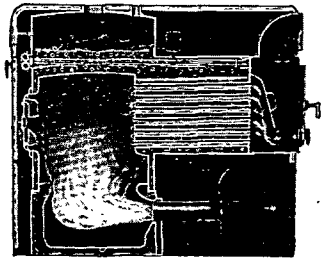
Ratings, horsepower—25 to 250 hand fired. 30 to 261 mech. fired.

FITZGIBBONS RESIDENTIAL STEEL HEATING BOILERS



THE 400 SERIES

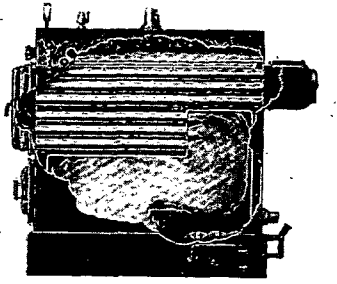
A boiler for mechanical firing that brings to the small home the many advantages of Fitzgibbons steel boiler construction. Electrically welded in a one-piece unit, it combines strength and durability with leak-proof, crack-proof construction. The Fitzgibbons principle of a single pass of many small tubes simplifies cleaning and assures high operating efficiencies. Five sizes—320 to 900 sq ft steam, net radiation load.



THE "OIL-EIGHTY" AUTOMATIC

This graceful, efficient and economical domestic heating boiler can be combined with any good oil burner to form a heating unit of outstanding merit. Fitzgibbons steel construction combines strength and resiliency with life-time, crack-proof, rust-resisting durability.

Many medium-sized and large homes have been well served by the Oil-Eighty during the war years of fuel stringency. In six sizes—1326 to 2680 sq ft steam. E.D.R.

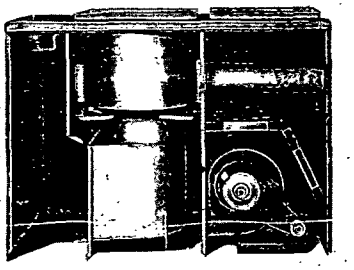


THE R-Z-U JUNIOR

The Fitzgibbons R-Z-U Junior is in effect a large boiler scaled down to sizes for the class of larger residences and smaller business, institutional and public buildings. It has large boiler qualities in fuel economy, quick steaming, convenience of operation. It is adapted for steam, vapor, vacuum or hot water systems, and for burning coal hand fired, or for mechanical firing with oil, gas or stoker.

In sizes up to 2900 sq ft steam hand fired and 3500 sq ft steam mechanical fired. E.D.R.

FITZGIBBONS DIRECTAIRE CONDITIONER



The unit provides automatic temperature control, automatic humidity control, filtered air, circulated air, summer comfort.

Fitzgibbons steel crack-proof construction, prevents all chance of combustion gas leakage into the air stream. The Directaire operates at extremely low fuel cost, with quiet performance.

Installation is easy and rapid.

For mechanical firing.

Sizes from 65,000 to 600,000 Btu per hour.

Farrar & Trefts

Incorporated

20 Milburn Street, **Buffalo 12, N. Y.**

ATLANTA, GA.
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HONOLULU, HAWAII
MILWAUKEE, WIS.
MINNEAPOLIS, MINN.
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NEW ORLEANS, LA.
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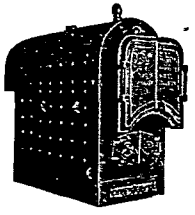
SAN ANTONIO, TEX.
SAN FRANCISCO, CALIF.
TAMPA, FLA.
TOLEDO, OHIO
WASHINGTON, D. C.

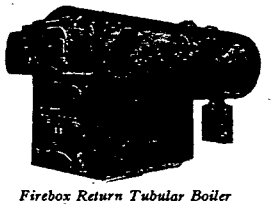


The Bison Compact Boiler Series 100 and 900

The **F&T Bison Compact Welded Heating Boiler** is more than just another boiler. It has been designed carefully so as to have a large furnace volume, the proper volume of water, just the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result is a remarkably steady water line—**A Balanced Boiler.**

This boiler requires a minimum amount of floor space and is easy and inexpensive to install. It is reasonable as to first cost and economical in operation. Construction is in accordance with the *A.S.M.E.* Code for 15 lb working pressure and boilers are designed for hand firing with anthracite or bituminous coal or for mechanical firing with oil, gas or stoker. There are various sizes available from 1,800 to 35,000 sq ft of steam radiation, all ratings as required by the *Steel Heating Boiler Institute.*

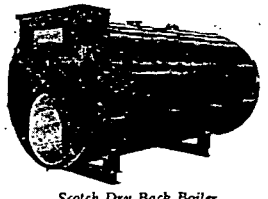
The **Bisonette Compact Boiler** has the same characteristics as the larger Bison Compact Boiler. It has been designed for installation in large residences and small business establishments where the advantages inherent in a **Steel** boiler are desired.



Firebox Return Tubular Boiler Series 500 and 600

Firebox Return Tubular Heating Boilers are **Quality Boilers.** They are constructed to measure up to the high standards set by Heating Engineers and will give unfailing service under all conditions. Being economical to install and operate, they are highly favored by Architects and Engineers for heating Schools, Hospitals, etc.

There are two types of Firebox Boilers, the Up-Draft Type and the Down-Draft Type. Both types are made of welded or riveted construction for heating purposes at 15 lb working pressure and riveted, or, Class 1 fusion welded x-rayed and stress-relieved for power purposes at 100, 125 and 150 lb working pressure in accordance with the *A.S.M.E.* Code. Sizes from 4.500 to 35,000 sq ft of steam radiation, as rated by the *Steel Heating Boiler Institute,* are designed for hand firing with coal or for mechanical firing with oil, gas or stoker.



Scotch Dry Back Boiler Series S-200

The **Bison Low Pressure Scotch Dry Back Boilers** are *carefully proportioned* and *balanced.* They are designed for hand, oil, gas or stoker firing, for ratings from 15 to 250 hp. These boilers operate efficiently and carry sustained overloads. The *Front Smokebox Doors Open Sideways* giving easy access to the tubes.

F & T boilers are designed so that the round furnace is always longer than the tube length which increases the furnace volume. This gives a large combustion volume in proportion to horsepower rating which makes the boilers very *economical to operate* and exceedingly *"Quick Steamers."*

"Catalogs on Request"

The International Boiler Works Company

350 Birch St., **East Stroudsburg, Pa.**

SALES OFFICES IN PRINCIPAL CITIES

FOR years "Fuel-Saver" Boilers Type C have met the requirements for low cost heating in office and apartment buildings, hotels, schools, theatres, industrial plants, etc.

Their design and construction makes them especially suitable for post-war heating requirements:

QUICK STEAMING

Due to rapid and positive internal water circulation.

MAXIMUM HEAT ABSORPTION

Due to effective distribution of heated gases.

EASE OF CLEANING

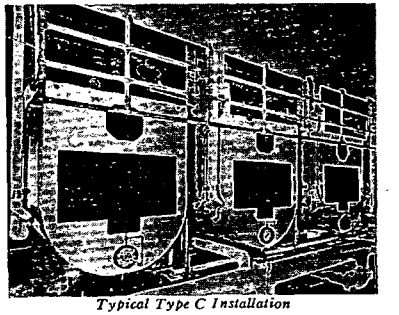
Due to accessibility of heating surfaces.

"FUEL-SAVER" Boilers have cut fuel costs in thousands of heating installations.

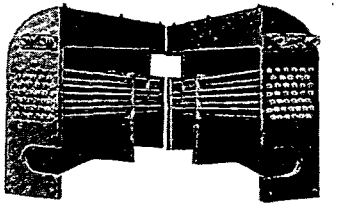
Complete range of standard sizes rated in accordance with *S.B.H.I.*–15 lb *A.S.M.E.* standard—for hand, stoker, oil or gas firing.

Type C twin section—a heating boiler in halves. For installation where Type C one piece cannot be carried through existing passages.

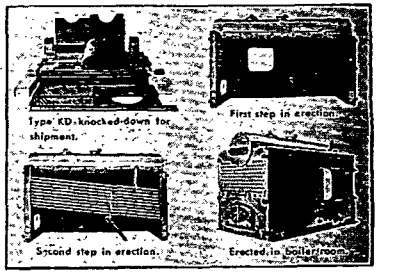
Type KD—knocked down—a heating boiler designed for shipment so that sections can be carried through a door or window. Eliminates expensive cutting or patching of building. Reduces time out when in need of steam.

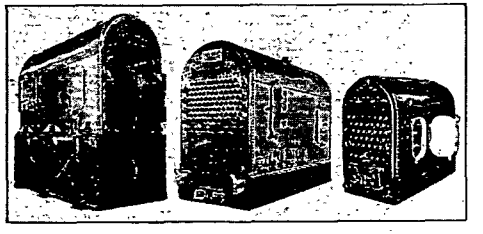


Typical Type C Installation



Halves of twin section ready for bolting





Type CR water-tube power boiler for processing and power. 100-125-150 W.S.P. *A.S.M.E.* standard.

Complete range of sizes—10 to 300 hp for hand, stoker, oil or gas firing.

Write for bulletins

KEWANEE BOILER CORPORATION

Division of American Radiator & Standard Sanitary Corporation

Kewanee, Illinois

BRANCHES IN 64 PRINCIPAL CITIES

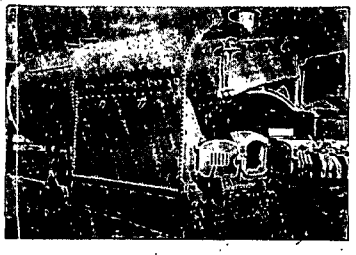
Won for the 6th time

Steel Heating and Power Boilers, Water Heating Garbage Burners, Tabasco Heaters and Tanks.

KEWANEE STEEL HEATING BOILERS

Kewanee offers a dependable line of Steel Boilers built for heating every size building with high efficiency burning any kind of fuel. There are 380 standard sizes and 33 types of Kewanee Boilers; most are kept in stock ordinarily for immediate delivery.

Seventy-eight years of intensive study and effort are back of Kewanee Boiler designs. They are all constructed in our extensively equipped factory at Kewanee, Illinois, in conformity with these Codes: *American Society of Mechanical Engineers* for construction, and for rating with the *Steel Boiler Institute* Simplified Practice. The Kewanee series include:

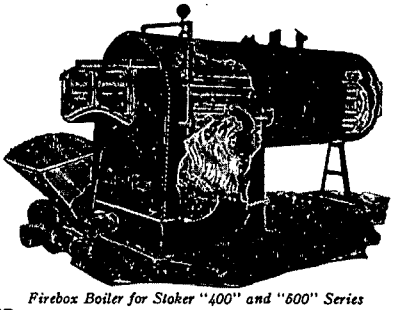


Firebox Boiler Portable Up-draft Type "400" and "600" Series, hi-Pressure

HEAVY DUTY RIVETED FIREBOX TYPES: 1,240 ft to 42,500 ft. Brickset and portable settings, Updraft and Downdraft Smokeless Furnace, Single-pass tubes for rear smoke outlet; Two-pass tubes for front smoke outlet.

WELDED BOILERS: 2,200 ft to 42,500 ft. Direct Draft or Smokeless Arch with Corrugated Crown Sheet. Rear Smoke outlet and Weld + Rivet for front Smoke outlet.

RESIDENCE STEEL BOILERS: 225 ft to 2,924 ft. Square and Round Type "R" with and without Jackets and Hot Water Heating Coils for Storage Tank or Instantaneous flow.



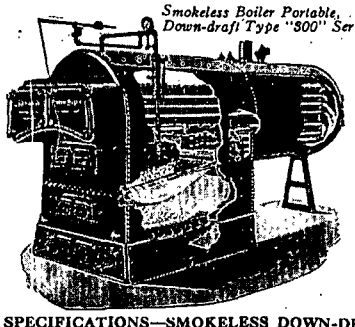
Firebox Boiler for Stoker "400" and "600" Series

SPECIFICATIONS—PORTABLE UP-DRAFT BOILER

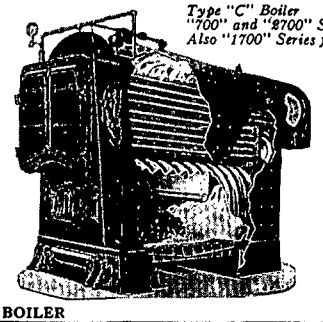
Boiler No.	576	577	578	579	580 480	581 481	582 482	583 483	584 484	585 485	586 486	587 487	588 488	589 489	590 490
Rated Steam Capacity:															
Coal.....Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker.....Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length .In. x Ft In.	42x8-7	42x9-6⅛	48x8-10	48x9-6½	54x11-5	54x13-1½	60x13-4½	60x15-5½	66x15-6½	66x18-0½	72x17-0	78x17-7⅛	78x21-3½	84x20-7	84x23-4
Overall Height Shell.....In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line.....In.	70	70	73	73	79½	79½	84½	84½	89½	89½	94	96½	96½	105	105
Approximate Weight: Coal..Lb	6100	6700	7300	7900	10200	11400	13300	14800	17300	19700	22000	24200	28400	32300	36100
Oil...Lb	5500	6100	6600	7100	9300	10400	12200	13600	15800	18100	20300	22300	26400	30200	33800

Rated Capacity for Water Boiler is 60 per cent greater than Capacity for Steam Boiler.

Tables for two series of Boilers list maximum dimensions only.

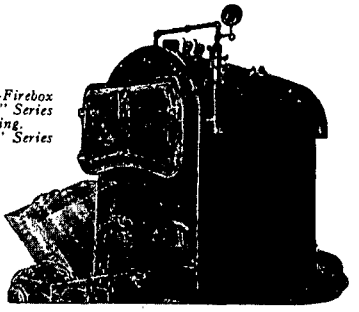


Smokeless Boiler Portable, Down-draft Type "300" Series



Type "C" Boiler "700" and "2700" Series for Coal. Also "1700" Series for Oil.

At Right: Type "C" Hi-Firebox Boiler, "7L70" Series for Stoker Firing. Also "27L70" Series Hand Fired.



SPECIFICATIONS—SMOKELESS DOWN-DRAFT BOILER

Boiler No.	376	377	378	379	380	381	382	383	384	385	386	387	388	389	390
Rated Steam Capacity:															
Coal.....Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker.....Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width and Length.....In. x Ft In.	42x8-3	42x9-3	48x8-7⅛	48x9-5	54x11-2⅛	54x12-11	60x12-8⅛	60x14-9	66x14-11	66x17-4	72x16-5	78x17-0⅛	78x20-7⅛	84x19-11	84x22-8
Overall Height Shell.....In.	80	80	86	86	94	94	101	101	107	107	113	115	115	125	125
Height of Water Line.....In.	70	70	73	73	78	78	85	85	88½	88½	94½	95	95	107½	107½
Approximate Weight: Coal.....Lb	6800	7500	8100	8800	10000	11100	13600	15300	18000	20500	22900	25100	29500	33600	37500
Oil.....Lb	6200	6900	7400	8000	9100	10100	12500	14100	16500	18900	21200	23200	27500	31500	35200

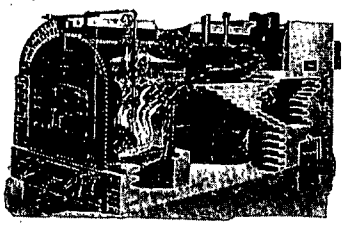
SPECIFICATIONS—TYPE "C" WELDED BOILER

*Boiler No.	2773	2774 2774	2775 2775	2776 2776	2777 2777	2778 2778	2779 2779	2780 2780	2781 2781	2782 2782	2783 2783	2784 2784	2785 2785	2786 2786	2787 2787	2788 2788	2789 2789	2790 2790
Rated Steam Capacity:																		
Coal.....Sq Ft	2200	2600	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker .Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width & Length, In.xFt in.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10⅛	42x8-6⅛	42x9-2	48x9-4⅛	48x10-7	54x9-11	54x11-2⅛	60x11-6⅛	66x12-3⅛	72x12-1⅛	72x13-4⅛	78x14-9⅛	84x14-2⅛	84x15-11⅛
Overall Height Shell...In.	77½	77½	77½	77½	83½	83½	83½	86½	86½	99	99	108½	112	118	118	122	135	135
Height of Water Line..In.	69	69	69	69	72	72	72	73	73	85	85	94	95	101	101	103	114	114
Approximate Weight:																		
700 Series, Coal.....Lb	....	3900	4400	5000	5500	6000	6500	7500	8400	9700	11000	12900	14900	16600	18400	22000	25200	28400
2700 Series, Coal....Lb	3400	3800	4300	4800	5300	5800	6300	7200	8100	9400	10600	12500	14400	16100	17900	21200	24400	27500
Oil.....Lb	2900	3300	3700	4100	4600	5000	5400	6100	6900	8000	9100	10700	12300	13800	15400	18100	20900	23400

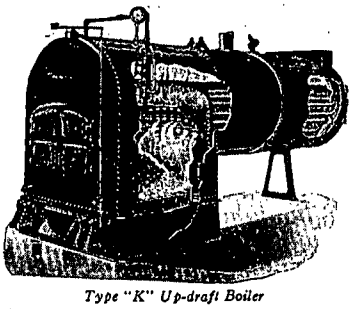
SPECIFICATIONS—TYPE "C" HI-FIREBOX WELDED BOILER

*27L70 Boilers have hand-fired coal grates and ratings, with 7L70 dimensions.

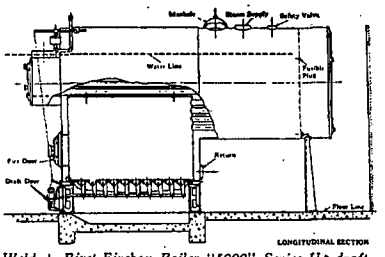
Boiler No.	7L73	7L74	7L75	7L76	7L77	7L78	7L79	7L80	7L81	7L82	7L83	7L84	7L85	7L86	7L87	7L88	7L89	7L90
Rated steam capacity,																		
Stoker.....Sq Ft	2680	3160	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Width x Length, In. x Ft In.	36x5-10	36x6-4	36x6-10	36x7-9	42x7-10⅛	42x8-6⅛	42x9-2	48x9-4⅛	48x10-7	54x9-11	54x11-2⅛	60x11-6⅛	66x12-3⅛	72x12-1⅛	72x13-4⅛	78x14-9⅛	84x14-2⅛	84x15-11⅛
Overall Height.....In.	82½	82½	82½	82½	88½	88½	88½	94½	94½	108	108	119½	123	131	134	140	157	161
Height of Water Line..In.	74	74	74	74	77	77	77	81	81	94	94	105	106	114	117	121	136	140
Approximate Weight..Lb	3100	3500	3900	4400	4900	5400	5800	6700	7600	8800	10000	11800	13600	15300	17000	20100	23200	26100



Brickset Boiler Up-draft Type



Type "K" Up-draft Boiler



Weld + Rivet Firebox Boiler "5000" Series Up-draft. Also "6000" Series Smokeless Down-draft

SPECIFICATIONS—BRICK-SET AND TYPE "K" PORTABLE UP-DRAFT BOILERS

Boiler No.	3 3K	4 4K	5 5K	6 6K	8 8K	9 9K	10 10K	11 11K	12 12K	12 13K	13 14K	14 15K	15 16K	17 18K	18 20K	19	20
Rated Steam Capacity:																	
Coal.....Sq Ft	1240	1380	1800	2200	3000	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000
Oil, Gas or Stoker.....Sq Ft	1770	2020	2190	2680	3650	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290
Width and Length...In. x Ft In.	30x9-10	36x8-10	36x10-4	36x11-10	42x11-4	42x12-10	48x11-10	48x13-4	48x14-10	54x15-3	54x17-9	60x17-0	60x19-6	66x19-6	72x19-6	72x16-0	72x18-0
Overall Height Shell.....In.	59	65	65	65	71	71	77	77	77	83	83	93	93	99	105	107	107
Height of Water Line.....In.	52	55	55	55	58½	58½	61	61	61	66	66	75	75	80	85½	85½	85½
Approximate Weight: Coal....Lb	3500	3900	4200	4700	5700	6400	7000	7600	8200	9500	10700	12600	14500	17600	20700	19400	21900
Oil.....Lb	3130	3550	3800	4200	5100	5800	6400	6900	7400	8600	9700	11500	13300	16100	19100	17700	20000

SPECIFICATIONS—WELD + RIVET FIREBOX BOILER "5000" SERIES

Boiler No.	5076	5077	5078	5079	5080	5081	5082	5083	5084	5085	5086	5087	5088	5089	5090
Rated Steam Cap: Coal....Sq Ft	3500	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, gas or stoker.....Sq Ft	4250	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Diam. x Length.....In. x Ft In.	42x7-3	42x8-0	42x8-11	42x9-10	48x9-9	48x11-1¼	54x11-1	54x12-9	60x13-2½	60x15-3	66x14-5½	66x16-0½	72x16-8½	78x17-5½	78x19-8½
Overall Height.....In.	81	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line.....In.	70	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Approx. Weight: Coal.....Lb	5100	5600	6100	6600	7500	8400	9600	10800	12600	14400	16100	17900	21200	24400	27500
Oil.....Lb	4500	5000	5400	5800	6600	7400	8500	9600	11100	12800	14400	16000	19200	22300	25200

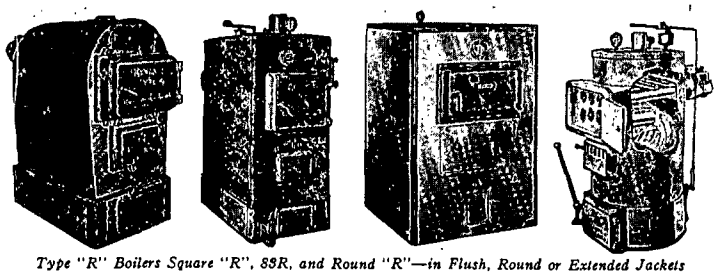
SPECIFICATIONS—WELD + RIVET SMOKELESS BOILER—"6000" SERIES

Boiler No.	6077	6078	6079	6080	6081	6082	6083	6084	6085	6086	6087	6088	6089	6090
Rated Steam Capacity: Coal.....Sq Ft	4000	4500	5000	6000	7000	8500	10000	12500	15000	17500	20000	25000	30000	35000
Oil, Gas or Stoker.....Sq Ft	4860	5470	6080	7290	8500	10330	12150	15180	18220	21250	24290	30360	36430	42500
Diameter x Length.....In. x Ft In.	42x8-2	42x8-8	42x9-6	48x9-7	48x11-0	54x10-8½	54x12-4½	60x12-9½	60x14-9½	66x14-0½	66x15-7½	72x16-4½	78x17-1	78x19-4
Overall Height.....In.	81	81	81	86½	86½	92½	92½	100½	100½	105	105	112	117	117
Height of Water Line.....In.	70	70	70	73½	73½	78	78	84	84	86	86	94½	96	96
Approximate Weight: Coal.....Lb	5800	6300	6800	7700	8600	9900	11200	13100	14900	16600	18300	21500	24500	27300
Oil.....Lb	5100	5600	6100	7000	7800	8900	10100	11900	13500	15100	16700	19800	22700	25300

KEWANEE TYPE "R" RESIDENCE BOILERS

Kewanee Type "R" Boilers are especially designed and constructed to meet all heating and hot water requirements for homes and small buildings. Every kind of solid fuel, coke, all grades of hard or soft coals and their briquette or treated forms are burned with excellent results. Also, any liquid fuel, oil, and natural or commercial gas can be used with high efficiency.

Standard snug fitting jackets, Rex or Regal extension styles enclosing burners are available for Round "R", Square "R" or 83R Boilers. Hot Water Copper Coil. 55 and 65 gal for Round "R". Capacities up to 720 may be ordered for Square "R" and 83R Boilers.



Type "R" Boilers Square "R", 83R, and Round "R"—in Flush, Round or Extended Jackets

SPECIFICATIONS—RESIDENCE BOILERS SQUARE "R" ROUND TYPE "R"

*Boiler No.	742	743	745	746	747	748	733	734	735	736	1737
Net Load Steam, Coal....Sq Ft	790	1000	1350	1600	1780	1960	225	330	450	570	.....
Oil, Gas or Stoker.....Sq Ft	840	1120	1470	1900	2160	2380	275	400	550	700	900
Width and Length.....In. x In.	32x39½	32x45½	32x45½	32x51½	32x57½	32x63½	.....	.....	.....	.....	.....
Diameter Firebox Inside....In.	.....	.....	.....	.....	.....	.....	17	20	23	26	26
Overall Height Shell Top....In.	59½	59½	70¼	70¼	70¼	70¼	54	54¾	58¼	61	61
Height of Water Line.....In.	48	48	58½	58½	58½	58½	46	44	45	49	49
Approximate Weight: Coal..Lb	2150	2360.	2800	3050	3300	3550	700	880	1040	1280	.....
Oil....Lb	1900	2060	2500	2730	2920	3125	600	800	920	1130	1400
Standard Jacket, Crated....Lb	205	225	175	190	200	225	.....	.....	190	215	.....

Boiler No.	83R1	83R2	83R3	83R4	83R6	83R7	83R8	83R9	Oil and Gas; or Stoker 083R.
N. Load Steam..	901	1105	1326	1513	2091	2363	2652	2924	
Wt. Jacketed....	1600	1800	2000	2200	2800	2900	3100	3300	

*Boiler series 1742-1748, 1733-1737 for oil, gas or stoker; 2742-2748, 2733-2736 for anthracite. Kewanee Indirect Hot Water Heating Coils for Type C, Square and Round "R" Boilers; 55 sizes, 55 to 1520 gal.

KEWANEE STORAGE WATER HEATERS

Kewanee Storage Heaters—use exhaust or live steam. 15 standard Coil Elements in 29 standard size storage tanks. Capacities 95 to 2240 gals per hr.

KEWANEE SCOTCH MARINE BOILERS

Kewanee Scottie Junior for High Pressure Steam in small industrial usage. 6 sizes 6 to 30 hp at 100 lbs steam pressure.

Kewanee Welded Scotch Marine, Low Pressure Steam. 18 sizes, 2680 to 42500 sq ft, mechanically fired.

KEWANEE HI-TEST BOILER

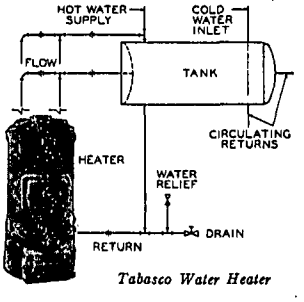
Kewanee Hi-Test Fusion Welded Series for High Pressure Steam in power or industrial process. 6 stock sizes, 50 to 150 hp, 125 and 150 lbs steam working pressure. All *A.S.M.E.* Code.

KEWANEE WATER HEATERS AND TANKS

All-Weld Water Heaters, Tabasco or Garbage Burning, 5 sizes to heat 165-700 gal 50 deg per hour. Extra Heavy W. P. 100 lb per sq in. 150 lb Test.

Water Heating Garbage Burner

Tanks 66 to 1700 Gal



Tabasco Water Heater

THE NATIONAL RADIATOR COMPANY

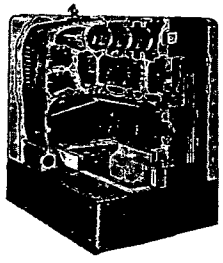
MODERN DESIGN **HEATING SYSTEMS**

Johnstown Pennsylvania

Branch Offices:

BALTIMORE	2622 Matthews Street	PHILADELPHIA	401 North Broad Street
BOSTON	167 Bridge Street, Cambridge	PITTSBURGH	304 Arrott Building
CHICAGO	Daily News Building	RICHMOND	Richmond Trust Building
NEW YORK	60 East 42nd Street	WASHINGTON	4034 Georgia Ave., N.W.

Nos. 1-2-3 SERIES HEAT EXTRACTORS



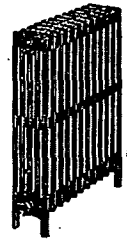
Cast iron boilers incorporating modern design features. Numerous flueways, with added extended surface (fins) on firebox crown and in flues. Readily convertible for all fuels and methods of firing. Built-in tankless and storage tank water heaters available.

NET I=B=R RATINGS

Hand, Stoker, Oil-fired

Steam —170–2,300 sq. ft.
Hot Water—270–3,680 sq. ft.

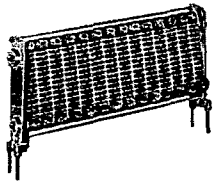
ART RADIATORS



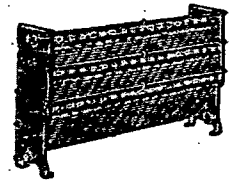
The National "Art" Radiator is a compact, thin tube radiator available in a complete range of sizes. Straight lines and receding plane surfaces provide an attractive modern appearance. Can be furnished legless, with high legs, or with special harmonizing pedestals.

OIL HEATING UNITS—Cast Iron and Steel in a Wide Variety of Types and Sizes
GAS HEATING UNITS—Handsome, Compact and Efficient—Enclosed Controls and Draft Diverter

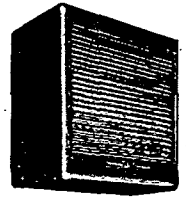
AERO CONVECTORS



The exclusive design feature of sloping sections exposes a greater amount of heating surface to the flow of air traveling through the convector. More heating surface can be arranged in less space with this design and additional sections are easily added. Wide fin spacing prevents clogging. Numerous convector sizes and possible arrangements, and various types and styles of enclosures available, provide flexibility for practically any desired installation requirement.

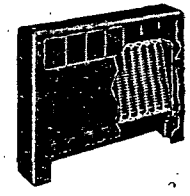


UNIT HEATERS



Where a large volume of air is to be heated in stores, garages and workshops, the NRC Unit Heaters will give efficient diffusion of warmth to all areas. Available in horizontal or vertical shaft types of heaters. For steam or hot water.

ENCLOSURES

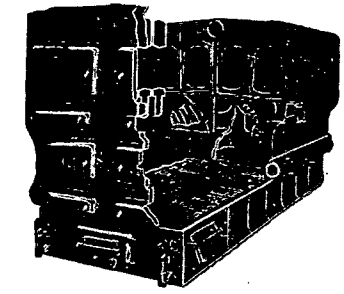


ACCESSORIES

Floor plates, valves, traps, circulating water pumps, controls, etc., are furnished for complete installations.

SERVING THE HEATING INDUSTRY SINCE 1894

Nos. 40-42-48 SERIES—Super-Smokeless and Standard



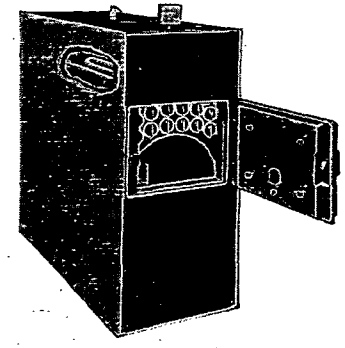
Large amount of prime heating surface, long flue travel and adequate flue-ways provide outstanding performance—regardless of whether hand, oil or stoker fired. Super-Smokeless Boiler (illustrated) delivers preheated air in fine jets into path of gases, burning unconsumed fuel. Twin sections (No. 48 Series only) pass readily through small doorways and facilitate assembly.

NET RATINGS (Bonded):

Nos. 40-42 Series—Steam—2,010 to 7,880 sq ft.
—Hot Water—3,216 to 12,608 sq ft.
No. 48 Series —Steam—3,830 to 12,900 sq ft.
—Hot Water—6,128 to 20,640 sq ft.

NATIONAL PREMIER STEEL BOILERS

Residential Series—Jacketed and Unjacketed



Low pressure steel boilers for residences and small commercial buildings. Electrunite tubes assure longer, trouble-free service. True-arch crown provides maximum firebox height and prevents sludge from accumulating over hottest part of boiler.

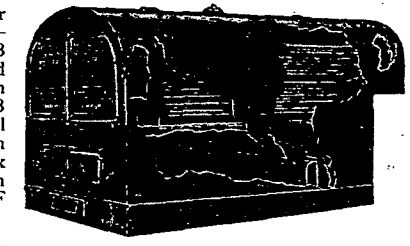
SBI NET RATINGS (Jacketed Boilers Only):

Oil-Fired and Oil Heating Unit —Steam 275 to 680 sq ft.
—Hot Water—440 to 1,090 sq ft.

SBI RATINGS (Unjacketed Boilers):

Type DB, Hand-Fired —Steam—485 to 2,200 sq ft.
—Hot Water—775 to 3,520 sq ft.
Type MB, Stoker-Fired } —Steam—735 to 2,680 sq ft.
Type OB, Oil-Fired } —Hot Water—1,175 to 4,280 sq ft.

Commercial Series—Unjacketed



Hand-fired Smokeless Boiler with rear smoke outlet (Type SB) illustrated—without smokeless feature is Type DB (Direct Draft). Shaking apparatus and water-cooled bridgewall eliminated on Type MB (Stoker-fired) and Type OB (Oil-fired) Boilers. Three-pass gas travel —recessed smoke box provides flush front. Built-in tankless and storage tank water heaters available. Boilers with front smoke outlet (Types SF, DF, MF and OF) can be furnished.

	SBI RATINGS:			
Built-in Tankless Hot Water Heaters	Type DB and SB	Hand-Fired	—Steam	—1,800 to 35,000 sq ft.
			—Hot Water	—2,880 to 56,000 sq ft.
	Type MB	Stoker-Fired	—Steam	—2,190 to 42,500 sq ft.
	Type OB	Oil-Fired	—Hot Water	—3,500 to 68,000 sq ft.
Built-in Storage Hot Water Heaters	Type DF and SF	Hand-Fired	—Steam	—3,000 to 35,000 sq ft.
			—Hot Water	—4,800 to 56,000 sq ft.
	Type MF	Stoker-Fired	—Steam	—3,650 to 42,500 sq ft.
	Type OF	Oil-Fired	—Hot Water	—5,840 to 68,000 sq ft.

CATALOGS WITH COMPLETE DATA AND DIMENSIONAL INFORMATION WILL GLADLY BE FURNISHED

Preferred Utilities Manufacturing Corporation

Executive Offices

1860 Broadway, **New York 23, N. Y.**

FACTORY—Danbury, Conn.

Branch Office: BOSTON, MASS.

Representatives in Principal Cities

PREFERRED UNIT STEAM GENERATOR

The Modern Packaged Power Plant

The Preferred Unit Steam Generator is a heavy duty, oil-fired, portable, automatic steam plant with all the equipment necessary to produce steam—economically.

The guaranteed minimum thermal efficiency of 80 per cent is made possible by the efficient design of the burner and the four pass arrangement of the heat-absorbing surfaces.

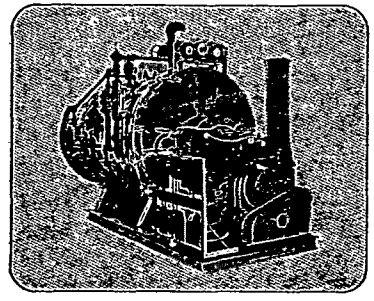
Sizes range from 20 to 500 bhp—for pressures up to 200 lb psi.

Induced Draft—The Preferred Unit Steam Generator operates with induced or "pull through" draft without the need of a stack. Air volume is automatically controlled to meet the requirements of complete combustion. The use of induced draft eliminates the very real danger of forcing the products of combustion into the boiler room, and contributes to the long life of refractories which have stood up remarkably well under the most trying operating conditions.

Heavy Oil Types—These units are made in all sizes, up to and including 500 bhp for use with the heavier grades of bunker oils—for either fully-automatic or semi-automatic operation.

Light Oil Types—For use with lighter oils the Preferred Unit Steam Generator is made in sizes up to 100 bhp and for pressures up to 200 lb psi.

For further details refer to *Bulletin No. 1000.*



Unit Steam Generator

OIL BURNERS

Preferred oil-burners are made in both the horizontal, rotary-cup and the mechanical-atomizing or gun types. Each type is suitable for manual, semi-automatic or fully-automatic operation.

The rotary-cup burner is suitable for use with either light or heavy oil and has several exclusive features. One important feature is its 4 to 1 throttle control which automatically delivers the correct mixture of oil and air for efficient combustion—at all rates of firing. Oil viscosity is controlled by a patented Volumeter. No other method of firing permits such perfect utilization of the source of heat.

Preferred mechanical-atomizing burners are designed to burn light oil only and for use where Diesel or lighter oils are used.

All Preferred burners are supplied with safety devices which prevent firebox flooding, or damage due to shut-down because of electric power failure or overload, or from foreign matter in the fuel oil.

OTHER PREFERRED PRODUCTS

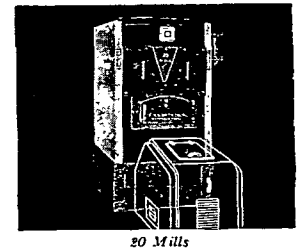
Water Heaters . . . . . **Draft-A-Justors**
Oil Burner Accessories . . . . . **Boiler Room Compounds**

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



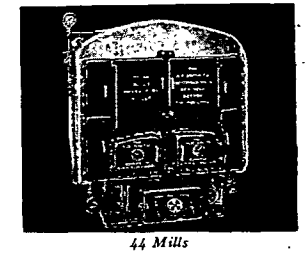
20 Mills

15-20-25 SMITH-MILLS BOILERS

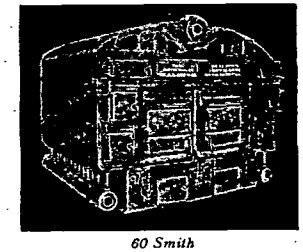
Capacities 200 sq. ft. to 2275 sq. ft. steam radiation. This complete line of modern push nipple boilers is available in models for oil, gas, stoker and hand firing. Provisions have been made for built-in domestic hot water heaters and controls.

MILLS WATER TUBE BOILERS

Capacities 900 sq. ft. to 13,380 sq. ft. of steam radiation. Independent header type construction—literally thousands of these famous Mills boilers have been installed in camps, cantonments, bases, war housing and other essential construction during the past few years. Models for hand and all types of automatic firing.



44 Mills



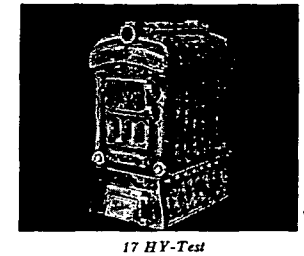
60 Smith

42 AND 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 process requirements as well as for heating and domestic hot water.

SMITH HY-TEST BOILERS

Smith Hy-Test Boilers for hot water supply, are available 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 high pressures before shipment. The No. 17 series, for example, is tested at 350 lbs. hydrostatic pressure—the highest test pressure of any cast iron boiler made.



17 HY-Test

Complete catalogue information describing Smith boilers is filed in current issues of Sweet's "Engineering" and Domestic Engineering Catalogue Directory

Spencer Heater

Division—The Aviation Corporation

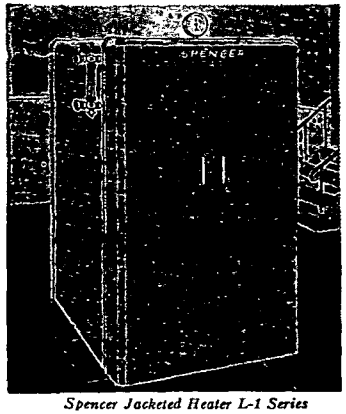
Williamsport, Pa.

Sales Representatives in Principal Cities

AN Avco PRODUCT

Spencer Automatic Magazine Feed Heaters are furnished in cast iron sectional types—and steel tubular types for larger buildings—for steam, vapor and hot water heating. There is a size and capacity for every type of building, to provide economical and convenient heat—safe, dependable, sure.

COMFORTABLE HEAT AT LOW COST



Spencer Jacketed Heater L-1 Series

Why Spencer Heaters perform so satisfactorily can best be explained by an inspection of their design and construction. The Spencer principle, illustrated in the cross-sectional view, is simple:

Once a day fuel (No. 1 Buckwheat Anthracite or small size by-product coke) is put into the magazine. It fills the sloping grate to the level of the magazine mouth. The fire bed always stays at the proper level, for as fast as fuel burns to ash, it shrinks and settles on the sloping grate; and more fuel rolls down automatically over the top of the fire bed. Fuel feed is by gravity alone, in just the right amount to keep the fire always burning at its most efficient combustion point.

This explains why a Spencer Automatic Magazine Feed Heater always gives the same uniform, satisfying heat, and burns less fuel. These exclusive Spencer advantages are available in all types of the magazine feed heaters and boilers.

Coal — Coke — Gas — Oil — Spencer J and L series heaters and M series boilers are primarily designed to burn low cost No. 1 Buckwheat Anthracite or small size coke.

If at any time a property owner desires to burn more expensive fuels—oil or gas—his Spencer Heater can be readily converted and will show a high efficiency.

Jacketed Covering—Illustrated in the attractive metallic jacket of the deluxe enclosing type for Spencer Cast Iron Heaters, either with or without the enclosing jacket doors.

Spencer Heavy Duty Tank Heaters—With the automatic magazine feed construction, they provide ample domestic hot water at lowest cost, and with a minimum of tank heater attention.

Cutaway sectional view Spencer Cast Iron Heater

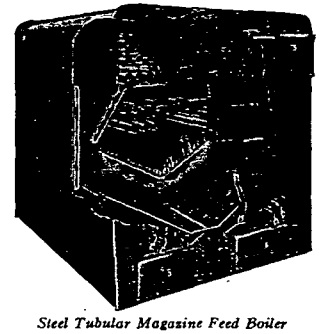
SPENCER ALL YEAR SYSTEM

In addition to the excellent heating facilities afforded by Spencer Magazine Feed Heaters, Year Round Domestic Hot Water Service can also be provided and assures at all times an ample supply of domestic hot water at lowest cost. Complete data for installation and operation upon request.

SPENCER STEEL TUBULAR MAGAZINE FEED BOILERS

For large buildings we recommend Spencer Steel Tubular Magazine Feed Boilers, burning low cost No. 1 Buckwheat Anthracite or coke.

In the cross-section diagram, part of the fire bed is cut away to show the sloping grates and the two magazines filled with fresh coal, ready to feed down automatically by gravity to the fire. These boilers are built in two vertical sections for ease in handling and installation—a great advantage on replacement jobs, eliminating the necessity of costly tearing out of walls or partitions. Combination water and fire tube construction; built to *A.S.M.E.* standards.



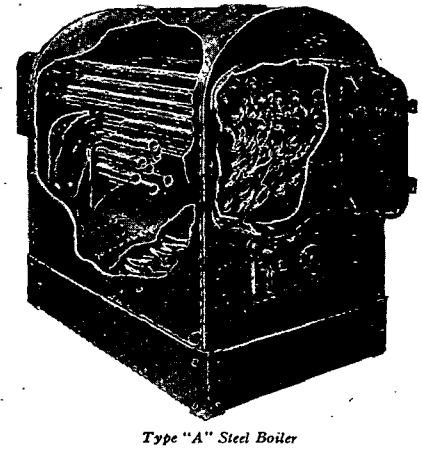
Steel Tubular Magazine Feed Boiler

SPENCER STEEL TUBULAR BOILERS For Oil, Stoker, Gas or Hand-Firing

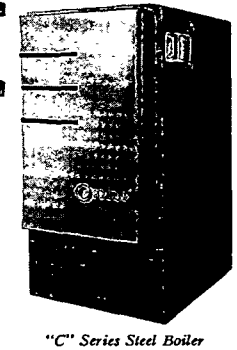
For more than 50 years, Spencer has been building, in the opinion of experts, one of the most efficient, economical and dependable automatic coal burning boilers on the market. With this background of experience, Spencer Engineers developed the Spencer Steel Tubular Boiler for oil, gas, stoker and hand-firing—the "K" and "C" series for residential use, and the Type "A" for larger buildings. They are better boilers both for the property owner and for the architect or engineer who specifies them.

The high sustained efficiency of these boilers means adequate heat for a lower fuel cost. Design is of the three pass type. Combustion chamber is amply large. Built of best quality open hearth steel boiler plate, and steel tubes. Can be furnished with domestic hot water heating coils, storage tank or instantaneous type.

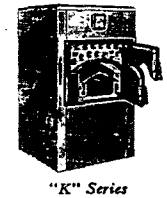
A complete range of sizes from 400 sq ft SHBI net steam rating up. They meet or exceed in every particular the requirements of the *A.S.M.E.* and S. H. B. I. Codes.



Type "A" Steel Boiler



"C" Series Steel Boiler



"K" Series

Every Spencer Boiler is guaranteed to carry more than its full rated load, giving the installer a definite factor of safety.

These boilers have all the advantages of the Spencer exclusive design and are readily adapted to mechanical oil or stoker firing—or hand-fired coal or coke.

Terre Haute Boiler Works Co.

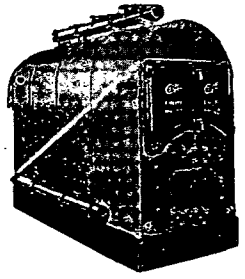
Established 1857

Main Office and Works:
Terre Haute, Ind.

Branches and Representatives in PRINCIPAL CITIES

CLIFF LOLINE SECTIONAL STEEL HEATING BOILERS

For Oil, Gas, and Coal Stoker or Hand Firing



Designed especially for boiler replacement work. Efficient Tube Heating Surface and Generous Firebox Proportions. Only four sections are required for maximum rating. Compactness and accessibility for replacement work in old buildings, etc., eliminates expensive and hazardous excavation or foundation wall removal. Each section will easily pass through the ordinary door opening.

Assembly of Front Header made easy with a Strap and Slip Bolt. Watertight Joint made with Composition Gasket on Boiler Side of Header. Yoke slotted for easy Slip Bolt Assembly. Small lugs on Yoke provide quick alignment of sections. Heavy stay-bolting throughout.

NOTE: Rock wool insulation and baked enamel, steel jacket furnished at extra cost.

RATINGS—Hand Fired—2100-29820; Oil, Gas, Stoker—2550-36210 sq. ft. Steam

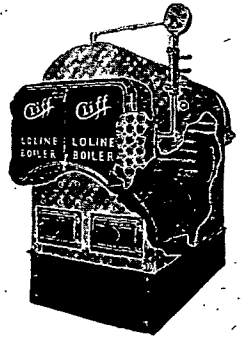
CLIFF STEEL RESIDENTIAL BOILERS

For Oil, Gas, Stoker or Hand Firing

A Three-Pass electric welded Steel Boiler furnished with Red Enameled Steel Jacket and 1½ in. thick rock wool insulation for Residence, Apartments and Smaller Buildings. Will pass thru 31 in. doorway. Designed for 15 lbs. working pressure on steam and 20 lbs. pressure on water. Rated in accordance with S.H.B.I. Code. Submerged type hot water coils can be furnished installed inside of boiler; either tank-heater or tank-less type. Ratings 800-3160 sq. ft. steam.

CLIFF HEAVY DUTY LOLINE BOILERS

For Oil, Gas, Stoker or Hand Firing
Also Hi-Firebox Type
A.S.M.E. Code Construction



Three-Pass compact Fire-Box Boilers. Designed and constructed to A.S.M.E. Standard for 15 lbs. steam or 30 lbs. water working pressures. All boilers comply with the Industry's simplified practice recommendations R157-35 as issued by the U. S. Department of Commerce and the S.H.B.I. Code for the rating of low pressure heating boilers.

Can also be furnished with a Cliff Patented Steel Jacketed Red Enamel Finish having 1½ in. Rock Wool Insulation.

RATINGS—Hand Fired—2600-35000; Oil, Gas, Stoker—3160-42500 sq. ft. Steam.

Write for complete information on CLIFF Boilers

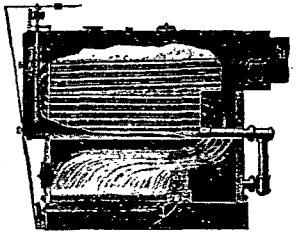
Pacific Steel Boiler Division

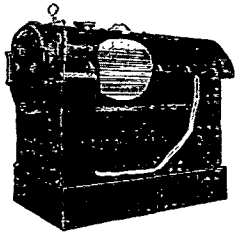
United States Radiator Corporation

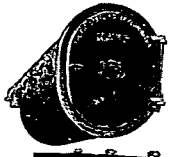
General Offices: **Detroit, Michigan**

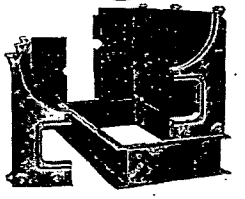
Sales Offices in Principal Cities

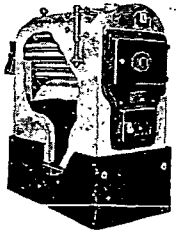
A Complete Line of Low Pressure Steel Heating Boilers











All Pacific Boilers are built using the *A.S.M.E.* Boiler Code Standards as minimums.

DIRECT DRAFT AND SMOKELESS SERIES FOR COAL FIRING

Built in the following capacities for steam: 2200 to 35,000 sq ft.

All Pacific Boilers are built, inspected, and tested under the supervision of the Hartford Steam Boiler Inspection and Insurance Company.

HIGH FIRE BOX SERIES FOR MECHANICAL FIRING, STOKER, OIL OR GAS

Built in the following capacities for steam: 2680 to 56,830 sq ft.

All Pacific Boilers are made of steel with each joint and seam electrically arc-welded —built to last a life-time.

PACIFIC THREE-PIECE CONSTRUCTION

Made up of three parts, shell, firebox and base, Pacific Boilers are particularly adaptable to replacement work. Where necessary Pacific fireboxes can be split (as illustrated) allowing the boiler to be taken into the building in four pieces and erected without welding on the job.

PACIFIC RESIDENTIAL SERIES FOR COAL, STOKER, OIL OR GAS

Built in the following capacities for steam: 320 to 2890 sq ft.

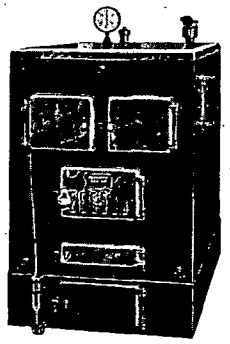
Descriptive Bulletins on Pacific Steel Heating Boilers will be mailed on request.

UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • Branches and Sales Offices in Principal Cities

Detroit, Michigan

CAPITOL RED TOP BOILERS



"B" Series Jacketed Boiler

"A" Series—All Fuels

Boiler No.	I-B-R Sq Ft Direct Cast-Iron Radiation	
	Steam	Water
A-7	340	545
A-8	440	705
A-9	540	865
A-10	640	1025
A-11	740	1185

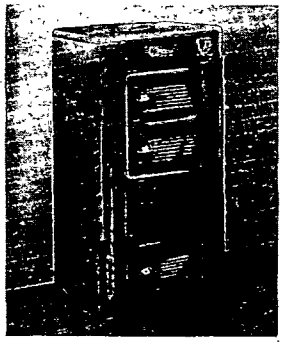
"B" Series—All Fuels

B-7	740	1180
B-8	920	1470
B-9	1110	1770
B-10	1275	2040
B-11	1440	2300
B-12	1580	2530
B-13	1730	2770
B-14	1880	3000

"C" Series—All Fuels

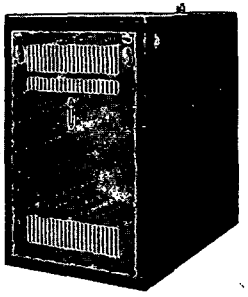
Boiler No.	Direct Cast-Iron Radiator Loads Sq Ft	
	Steam	Water
C-12	2250	3600
C-14	2715	4345
C-16	3180	5090
C-18	3645	5830
C-20	4110	6575
C-22	4575	7320
C-24	5040	8065
C-26	5450	8720
C-28	5800	9280
C-30	6110	9775

NO. 3 SERIES SUNRAY



U. S. Sunray No. 3 Series

Boiler Number	I-B-R Sq Ft Direct Cast Iron Radiation					
	Hand Fired		Stoker Fired		Oil Fired	
	Steam	Water	Steam	Water	Steam	Water
43-S or W	300	480	400	640	400	640
53-S or W	450	720	550	880	550	880
63-S or W	600	960	700	1120	700	1120



U. S. 25 Series Oil Burning Boiler

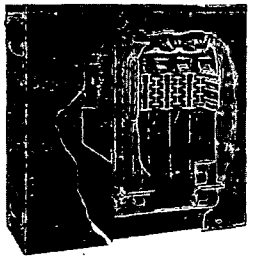
Boiler No.	Net I-B-R Sq Ft Direct Cast-Iron Radiation		Net I-B-R Btu	Gross I-B-R Output Btu
	Steam	Water		
US-25-3	545	875	131,000	198,000
US-25-4	905	1450	217,000	317,000
US-25-5	1255	2025	304,000	435,000
US-25-6	1625	2500	390,000	554,000
US-25-7	1985	3175	476,000	672,000

UNITED STATES RADIATOR CORPORATION

General Offices: Detroit, Michigan • Branches and Sales Offices in Principal Cities

Detroit, Michigan

CAPITOL SUNRAY No. 2 SERIES OIL FIRED BOILER

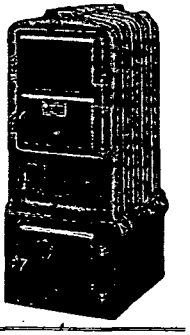


Boiler No.	Net I-B-R Sq Ft Direct Cast-Iron Radiation		Net I-B-R Btu	Gross I-B-R Output Btu
	Steam	Water		
2-03	350	560	84,000	129,000
2-04	500	800	120,000	182,000
2-05	650	1040	156,000	234,000
2-06	800	1280	192,000	285,000

"DEEPFIRE" HOT WATER SUPPLY BOILERS

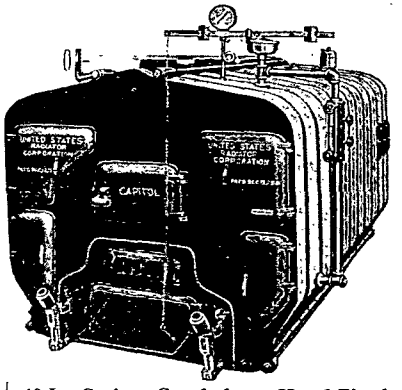
Coal Fired

Tested to 250 lb Hydrostatic Pressure for 100 lb Working Pressure.



Boiler No.	Capacity—Gallons		
	100° Rise 6 Hours	85° Rise 1 Hour	100° Rise 1 Hour
30	792	155	132
40	1512	297	252
50	2160	423	360
60	2700	529	450

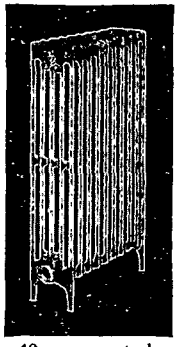
CAPITOL SQUARE SECTIONAL BOILERS



40 In. Series—Smokeless—Hand Fired

Boiler No.	Direct Cast Iron Radiator Loads Sq Ft		Grate Area Sq Ft	Coal Capacity Cu Ft	Outlets	Min. Chimney Sizes	
	Steam	Water				Height Ft	Dim. Inches
740	2480	3970	8.15	10.40	2—5"	50	18x18
840	3070	4910	10.31	13.30	2—5"	55	18x20
940	3660	5855	10.31	13.30	3—5"	60	20x20
1040	4250	6800	12.47	16.30	3—5"	65	20x24
1140	4820	7710	14.63	19.25	3—5"	70	24x24
1240	5320	8510	14.63	19.25	3—5"	70	24x28
1340	5800	9280	16.79	22.20	3—5"	75	24x28

*CAPITOL THINTUBE RADIATORS



3-Tube	
Heights In.	Per Section Heating Surface
25	1.6 Sq Ft
4-Tube	
22	1.8 Sq Ft
25	2.0 Sq Ft
5-Tube	
22	2.1 Sq Ft
25	2.4 Sq Ft
6-Tube	
19	2.3 Sq Ft
25	3.0 Sq Ft
32	3.7 Sq Ft

*1¾ in. Centers.

40 per cent less space needed for these graceful, efficient Capitol ThinTube Radiators.

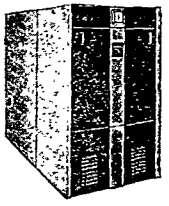
LITERATURE UPON REQUEST

Weil-McLain Company

Manufacturing Division: **Michigan City, Ind. and Erie, Pa.**
General Offices: **641 W. Lake Street, Chicago 6, Ill.**
NEW YORK OFFICE: **501 Fifth Avenue**

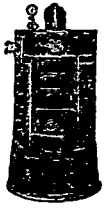
Weil-McLain Boiler and Radiator service is made conveniently available through local stocks carried by Weil-McLain Distributors in most of the important distributing centers.

No. 68 Boiler for Automatic Firing
Boiler is completely jacketed and insulated. Has an integral front burner extension. Net I-B-R Ratings: Steam 390 to 690 sq ft, Water 625 to 1,100 sq ft.

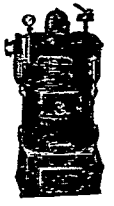


No. 78 Boiler for Automatic Firing
Boiler has insulated enameled de luxe jacket. Front or rear jacket extension available. Net I-B-R Ratings: Steam 530 to 1,130 sq ft, Water 850 to 1,810 sq ft.

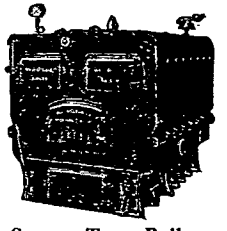
Nos. 57, 67, 77 All-Fuel Boilers
Conversion type boilers for hand or automatic firing. Connected Load Ratings: Steam 210 to 1000 sq ft, Water 340 to 1600 sq ft.



"RO Series" Boiler for Automatic Firing
Jacketed and insulated round boiler for small homes. Connected Load Ratings: Steam 420 and 520 sq ft, Water 630 and 790 sq ft.

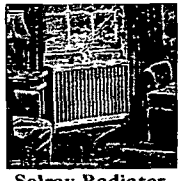


Round-Type Boiler
Unjacketed Round Boiler with corrugated heating surfaces for economical home heating. Connected Load Ratings: Steam 275 to 1,000 sq ft, Water 440 to 1,600 sq ft.

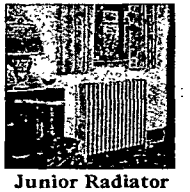


Square-Type Boilers
Sectional boilers for larger installations. Complete range of sizes. Connected Load Ratings: Steam 1,815 to 11,300 sq ft, Water 2,900 to 17,900 sq ft.

Raydiant "Concealed"
A Radiant convector type all cast-iron Radiator. Made in "Concealed," also Partially Recessed types.



Solray Radiator
Free standing all cast-iron Cabinet type Radiator with metal cover top. Available in three depths in 21, 24 and 27 in. heights.



Junior Radiator
Smaller Tubular type Radiation which conserves space. Available in 1¾ in. centers in 3, 4, 5 and 6 tube widths and 13 to 32 in. heights.

The Brownell Company

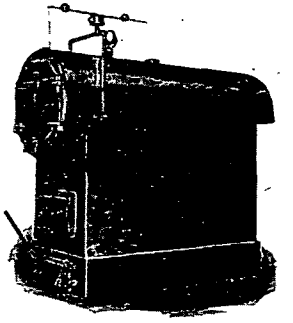
ESTABLISHED 1855

Dayton 1, Ohio

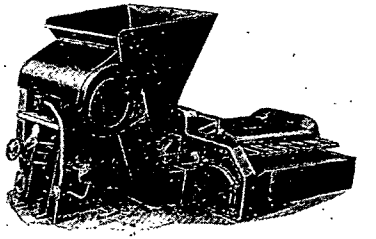
Manufacturers of

BROWNELL BOILERS, STOKERS, AND HEAT EXCHANGERS

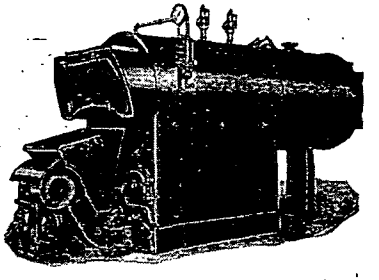
Representatives in All Principal Cities



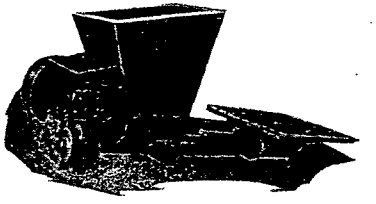
Welded Triple Pass Heating Boilers built in either high leg or low water line types. Hand fired ratings 500 to 35,500 sq ft steam, 800 to 56,800 sq ft water radiation. Stoker, Oil or Gas fired up to 43,100 sq ft steam or 69,000 sq ft water radiation. *A.S.M.E.* Code construction.



Type LR (Low Set) Underfeed Ram Type Stoker. Ideal where height of setting is limited. Sizes to 300 hp; "R" models up to 700 hp. Automatic air volume control. Can be furnished with Brownell exclusive, fully automatic intermittent coal feed control.



High or Low Pressure Double Pass Boiler with Type LR Stoker. Designed and manufactured as a matched unit steam generating plant. Furnished in working pressures from 15 to 150 lb and sizes up to 300 h.p. For power, heating, and process steam. Steam ratings 3,600 to 42,500 sq ft. Water rating, 5,800 to 68,000 sq ft when used with stoker, oil, or gas. *A.S.M.E.* Code construction.



Type C Screw Feed Stoker, proved by years of service to be sturdy, reliable and efficient. Illustration shows dead plates; can also be furnished with dump plates in the larger sizes. 30-300 hp.

Heat Exchangers, Generators, Converters. Hourly capacities from 60 to 4,400 gal; storage capacities, 25 to 1,904 gal.

The illustrations show only a part of the complete Brownell line. We shall gladly send literature describing Brownell equipment. Our field organization is ready to assist in problems of steam generation and heating.

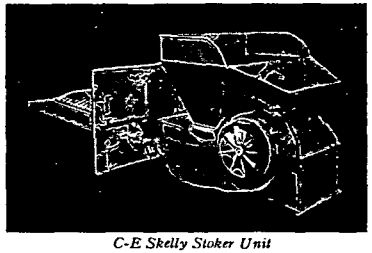
Combustion Engineering Company, Inc.

All Types of Fire Tube and Water Tube Boilers
Mechanical Stokers

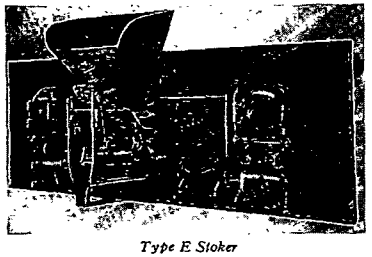
Complete Steam Generating Units
Pulverized Fuel Systems

200 Madison Avenue, **New York 16, N. Y.**
Offices in all principal cities of the United States and Canada

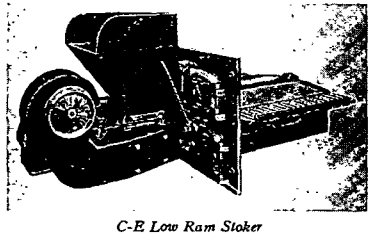
More than 18,500 C-E Stokers purchased to date



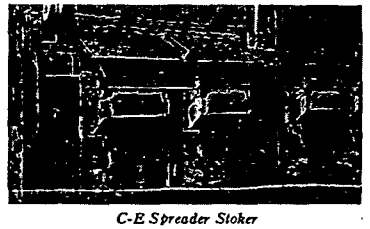
C-E Skelly Stoker Unit



Type E Stoker



C-E Low Ram Stoker



C-E Spreader Stoker

C-E Skelly Stoker—A compact, self-contained unit with integral forced-draft fan, adapted to burn either anthracite or bituminous coal. Alternate fixed and moving grate bars assure lateral distribution of fuel. Automatic control is standard equipment. Approximate application range—20 to 200 rated boiler hp.

Type E Stoker—A single-retort, underfeed stoker with an established reputation of many years' standing for dependable service. Designed to burn a variety of bituminous coals under boilers up to about 600 rated hp. Available with steam or electro-hydraulic drive.

C-E Low Ram Stoker—A single-retort, stationary-grate underfeed stoker for burning bituminous coals under boilers in the upper size range of the C-E Skelly Stoker.

C-E Spreader Stoker—A simple, rugged overfeed stoker designed to burn a wide variety of coals. Fines are burned in suspension and the coarser coal on a grate which may be of either stationary or dumping type. Rate of coal feed and air supply may be regulated over a wide range and are readily adaptable to automatic control. Applicable to boilers from about 100 boiler hp up.

C-E Multiple Retort Stoker—For burning bituminous and semi-bituminous coals under boilers up to the largest sizes.

C-E Traveling Grate and Chain Grate Stokers—Including both Coxe and Green types. Available with grate surfaces suitable for anthracite, coke breeze, lignite or bituminous coal, as required. Traveling grates are all forced-draft types; chain grates are either forced or natural draft types.

C-E Boilers—All fire tube and water tube types in sizes ranging from 25 hp up to the largest. Standard and special designs to suit all conditions of fuel, load and space. Included are all types formerly known by the trade names "Heine," "Walsh & Weidner," "Casey-Hedges," "Ladd" and "Nuway."

Separate Catalogs describing each of these stokers are available. A-531-D

Detroit Stoker Company

Sales and Engineering Offices
General Motors Bldg., Detroit, Mich.

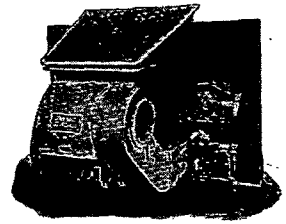
Main Offices and Works at Monroe, Mich.

Since 1898

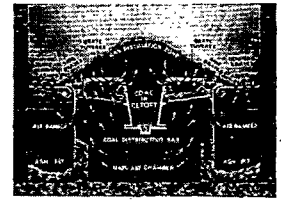
District Offices in Principal Cities

Built in Canada at London, Ont.

Detroit Stokers are unsurpassed for economy and dependability. They include Underfeed and Overfeed Spreader Stokers of many sizes and capacities for all types of boilers. All grades of Bituminous Coal successfully burned. Catalogs of various types, furnished on request.



Detroit LoStoker (brickset type) for horizontal tubular, firebox or water tube boilers.



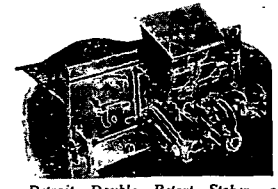
Built to fit the furnace. Arrows indicate flow of air to all parts of the fuel bed including the dumping grates, located at each side.

Detroit Unistoker with Detroit Adjustable Feed (Coal Feed Control) insures accurate fuel and air supply for best economy. Single Retort, side cleaning for boilers approximately 150 to 300 hp with furnace widths from five feet six inches to eight feet.

Double Retort Stoker—a multiple retort stoker having two retorts with the side cleaning feature. For medium sized boilers with wide furnaces.

Detroit RotoStokers are Overfeed Spreader Stokers, having an Overthrow Rotor action, which insures uniform fuel distribution over the entire area. Offers advantages over other firing methods for burning inferior fuels and efficiently handling extremely fluctuating loads.

Detroit UniStoker with Detroit Adjustable Feed provides a wide range of coal feed control.

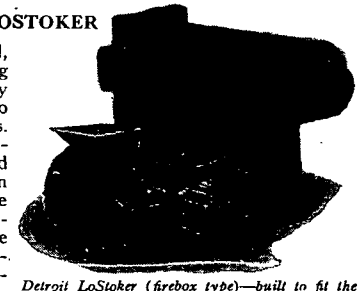


Detroit Double Retort Stoker, a multiple retort side cleaning stoker for medium size boilers.

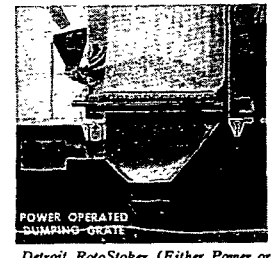
Detroit RotoStoker. Ash removed through doors at grate level successfully burns a wide range of fuels.

DETROIT LOSTOKER

Plunger feed, side cleaning stoker. Many sizes for 50 to 150 hp boilers. Highly efficient—fuel is fed only when needed—none is wasted. Requires little power for operation. Automatically controlled.



Detroit LoStoker (firebox type)—built to fit the furnace of high Firebox Boilers. Coal hopper with agitator designed to clear boiler doors.



Detroit RotoStoker (Either Power or Hand Operated) for large boilers. Especially suited to fluctuating loads.

Iron Fireman Manufacturing Company

Automatic Coal Stokers

Portland, Oregon **Cleveland, Ohio**

Address inquiries to 3815 West 106 St., Cleveland 11, Ohio

Retail Branches or Subsidiaries: CHICAGO, ILL.; MILWAUKEE, WIS.; ST. LOUIS, MO.; NEW YORK, N. Y.; BROOKLYN, N. Y.; TORONTO, CANADA; MONTREAL, CANADA

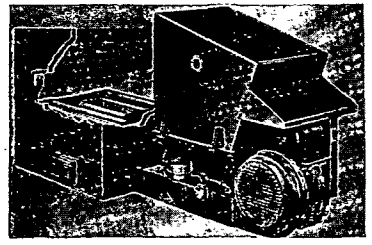
Dealers in Principal Cities and Towns in the United States and Canada

Representation in numerous foreign countries

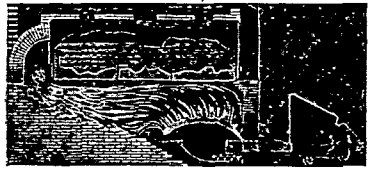
COMMERCIAL AND INDUSTRIAL STOKERS

STANDARD HOPPER MODELS

This series of stokers is the standard of value in equipment for automatically firing boilers ranging in size up to 350 hp. Available in a wide range of coal feeding capacities, lengths and grate arrangements, to fit varied requirements.

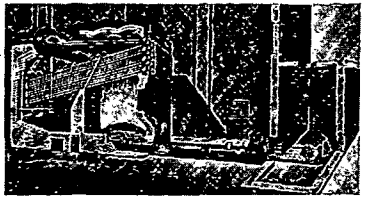


Commercial Installation—Hopper Model



Hopper Model Iron Fireman in Operation in Horizontal Return Tubular Boiler

STANDARD COAL FLOW MODELS



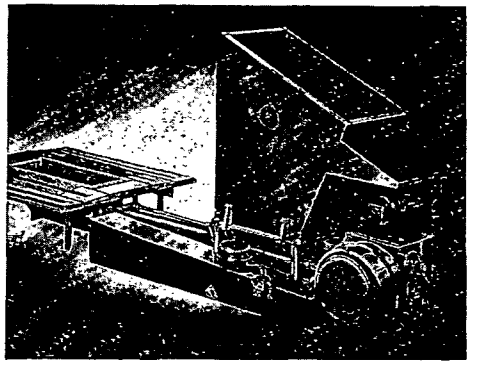
Commercial-Industrial Installation Coal Flow model that carries coal direct from bunker to fire

A heavy duty stoker which combines Iron Fireman's well known firing efficiency with the automatic conveying of coal direct from bunker to fire. An integral coal conveying mechanism eliminates the labor and expense of manual coal handling. The IRON FIREMAN COMMERCIAL-INDUSTRIAL COAL FLOW stoker fires boilers developing up to 350 horsepower. PNEUMATIC SPREADER STOKERS also available in Coal Flow models.

MODEL	OUTPUT RANGE		
	Boiler Horsepower	Equivalent Direct Radiation	
		Steam (240 Btu)	Hot Water (150 Btu)
Coal Flow (available in all models)	3 to 500*	400 to 70,000	650 to 110,000
Commercial and Industrial Standard Underfeed	3 to 350	400 to 50,000	650 to 75,000
Commercial and Industrial Poweram Underfeed	30 to 400	4,000 to 56,000	6,000 to 90,000
Pneumatic Spreader	50 to 1,000*	7,000 to 140,000	11,000 to 225,000

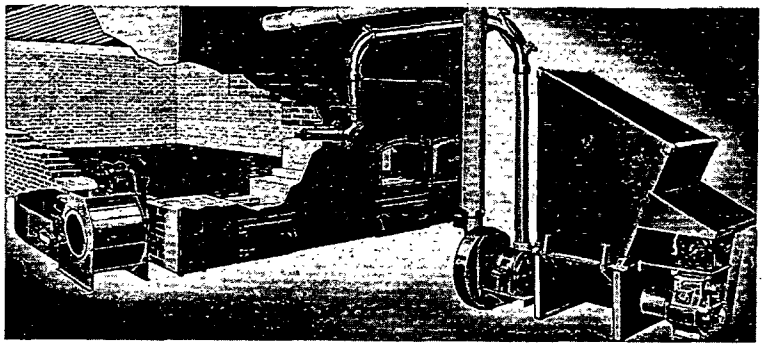
*Multiple units available for larger boilers.

POWERAM STOKERS



Combines ram-type coal distributor system in retort with free-running worm conveyor from coal supply, and has these advantages: Delivers the fuel to the fire bed in a loose, easily aerated condition; the reciprocating pusher blocks insure proper fuel distribution, and make possible the successful and efficient burning of many types of coal which otherwise are impractical to fire automatically with underfeed stokers. Designed for boilers developing up to 400 hp.

PNEUMATIC SPREADER STOKERS



As shown in the illustration above, the IRON FIREMAN PNEUMATIC SPREADER STOKER meters steam size coal from hopper or main coal bunker to transfer housing, where coal is picked up by pneumatic conveyor and delivered to furnace. The conveyor nozzle accurately spreads the larger particles of coal over the entire grate in a shallow, uniform fuel bed. The preheated fines burn in suspension, reducing the cinder carry-over and greatly improving the combustion efficiency and responsiveness, as compared with other stokers which do not preheat fuel. The conveying air provides the overfire air which is essential for efficient combustion. Entering at right angles to the flow of burning gases from the fuel bed, the conveying air produces *maximum turbulence; another requisite of efficient and smokeless combustion.* THE IRON FIREMAN PNEUMATIC SPREADER STOKER was designed to burn efficiently such economical fuels as the lower rank bituminous, and sub-bituminous coals and also lignite. It provides reliability of operation, physical adaptability, ease of operation, and low maintenance which is not afforded by other types of automatic coal burning systems. Pneumatic Spreader stokers are particularly adaptable to operation at high ratings, and as a result are greatly stepping up steam output in many plants throughout the United States and Canada. IRON FIREMAN PNEUMATIC SPREADER STOKERS are made in both hopper and Coal Flow models; the latter carry coal direct from the bunker to the fire.

The Webster Engineering Co.

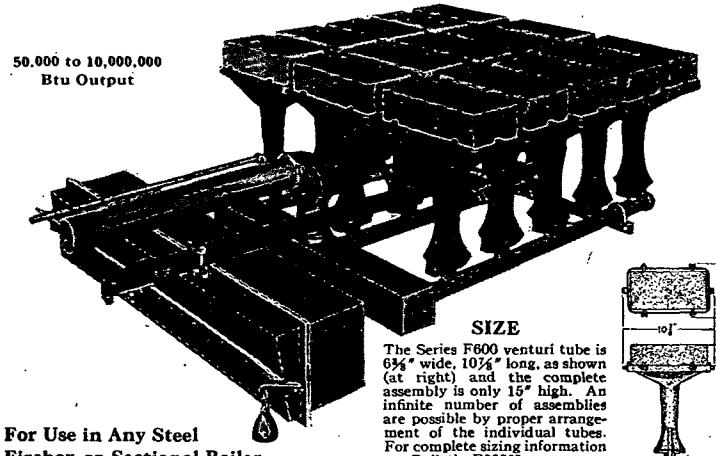
419 West 2nd St., **Tulsa, Oklahoma**

Division of

SURFACE COMBUSTION CORP., TOLEDO, OHIO

WECO-N.G.E. SERIES F600 GAS BURNERS

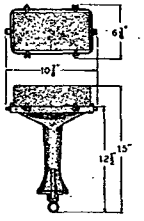
50,000 to 10,000,000 Btu Output



For Use in Any Steel Firebox or Sectional Boiler

SIZE

The Series F600 venturi tube is 6¾" wide, 10¼" long, as shown (at right) and the complete assembly is only 15" high. An infinite number of assemblies are possible by proper arrangement of the individual tubes. For complete sizing information see Bulletin F600H.



Improved venturi and greater port area insure much higher capacities at lower pressures.

Unique baffles at the outlet of the mixing tube make possible perfectly even distribution of flame completely around the baffle brick. As a result the maximum flame length is greatly reduced.

Interchangeable grills with multiple ports can be varied to suit the combustion characteristics of various gases. The proper sizing of these grills prevents any possibility of flash back.

In addition to the above major improvements the F600 possesses the same desirable features that made the 600 so popular.

1. Simple installation requiring no expensive insulated combustion chamber and having no furnace radiation loss.
2. Extreme quietness due to low rate of combustion over a large area.
3. Flexibility from infinite number of possible combinations varying both size and shape to meet load and firebox conditions at various gas pressures.
4. High radiant transmission rate due to radiant temperature of the standard firebrick baffles on the top of the burner tubes.
5. Low draft loss because of ample secondary air openings.
6. Plain gas pilots of heat resistant material and of a design that will not allow flame to pull off.
7. Safety pilot applied in a cool zone in a manner that insures perfect direct ignition of the burner yet allowing the the thermal element to cool quickly upon flame failure.
8. Guaranteed vibrationless under all conditions.

CAPACITY OF SINGLE F600 VENTURI TUBE—No. 17 MTD ORIFICE

Manifold Gas Pressure..	0.5" W.C.	1.0" W.C.	2.0" W.C.	3.0" W.C.	4.0" W.C.	5.0" W.C.	6.0" W.C.	4 oz.	6 oz.	8 oz.
Input—Cu Ft. 1 hr.....	24.5	38.5	58.0	72.0	84.0	94.5	104.0	112.0	138.0	159.5
Output—Sq Ft. St. Rad.	75	117	178	222	258	289	318	343	423	488
Output—Boiler H.P....	.54	.84	1.28	1.59	1.85	2.07	2.28	2.46	3.04	3.4

Automatic Burner Corp.

1823 Carroll Avenue

Chicago 12, Illinois

DOMESTIC BOILER AND FURNACE UNITS
RANGE BURNERS—WATER HEATERS
SPECIAL PRODUCTS

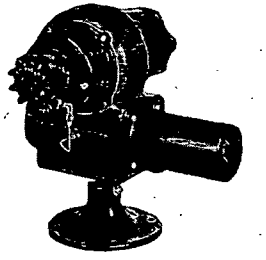
AUTOMATIC OIL BURNING HOT WATER HEATER

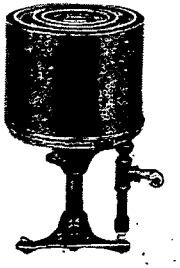
Instant hot water is provided by all A B C models, capacities of 65, 125, 300, or 500 gallons per hour. Extra efficiency is provided by the built-in fire and water tubes.

Each water heater is perfectly matched with the correct specially designed gun type A B C burner. All models are compact and beautifully finished.

PRESSURE TYPE OIL BURNER

The gun type oil burner has the famous A B C Oilairator mechanism that atomizes the fuel, mixes it with air, and delivers it with correct twist and velocity to insure complete combustion. The A B C nozzle combined with the proper choke and turbulator guarantees precise control and flexibility . . . guarantees a thoroughly satisfactory economical home heating unit. Capacity 0.6 to 6.0 gallons per hour.





RANGE BURNER

A B C offers complete interchangeability of parts—precision manufactured—all-steel construction—butt welded all-steel shells that fit tightly to the pressed steel base preventing air leakage and insuring an efficient flame. The full blue flame is secured in a few minutes. Oversized tubing throughout prevents plugging and air pockets. Precision-made one piece valve insures precise oil flow adjustment. The A B C is easy to clean and easy to operate—a well built, adaptable, safe range burner.

Combustion Equipment Division

TODD SHIPYARDS CORPORATION

601 West 26th Street, **New York 1, N. Y.**

NEW YORK, BROOKLYN, ROCHESTER, HOBOKEN, NEWARK, PHILADELPHIA, CHICAGO, SO. PORTLAND, ME., BOSTON, SPRINGFIELD, MASS., BALTIMORE, WASHINGTON, DETROIT, GRAND RAPIDS, TAMPA, GALVESTON, HOUSTON, MOBILE, NEW ORLEANS, LOS ANGELES, SAN FRANCISCO, SEATTLE, TACOMA, MONTREAL, TORONTO, BUENOS AIRES, LONDON.

THE TODD HEX-PRESS REGISTER in combination with the **TODD "VEE-CEE" VARIABLE CAPACITY BURNER** . . . makes possible increased combustion efficiency under almost any type of boiler of 100 hp. capacity or larger, operating at 50 lb. steam pressure or higher.

It provides equal efficiency under either forced or natural draft conditions. The Hex-Press Register assures the most intimate mixture of oil and air as well as quicker, more complete combustion . . . with minimum draft loss at high capacity . . . effecting great economy in maintenance and materially reducing fuel costs.

Through the exclusive "variable range" feature of the "Vee-Cee" Burner, practically unlimited firing range is assured . . . without change of burner tips, oil delivery pressure or angle of spray.

Constant steam pressure can be maintained regardless of demand : . . changing load requirements are met instantly under manual or fully automatic control.

COMBINATION GAS AND OIL BURNERS

For Natural or Refinery Gas and/or Fuel Oil. Available in wide range of capacities. Quickly adjustable for the combustion of either fuel alone, or both in combination. Of special value where fluctuating comparative costs of these fuels call for equipment suited to change-over without time-consuming structural changes.

Maintenance and operation are reduced to a minimum by compactness and simplicity of design . . . accessibility of all parts . . . rugged construction and positive overall efficiency.

Design features eliminate possibility of escaping gas due to structural distortion . . . prevent stratified combustion resulting from improper air distribution and high gas pressure.

Providing sufficient flexibility to care for varying loads, these units assure high furnace temperature and radiant heat transfer with low stack temperature . . . thorough mixture and optimum air-fuel ratio with utmost ease of adjustment.

ROTARY FUEL OIL BURNERS

For firing high or low pressure steam or hot water boilers of all types . . . in smaller factories and industrial plants, laundries, dryers and cleaners, office buildings, hotels, apartment houses. Also applicable to industrial ovens, kilns, etc., where furnace and general physical conditions permit.

Available with manual, semi-automatic or fully automatic control . . . in varying sizes and types . . . for burning light or heavy oil.

Horizontal atomizing cup is rotated by direct-connected electric motor, assuring constant firing as long as motor is in operation. Motors are of extra large frame size, air-cooled and built to withstand long, hard service. Positive air-oil interlocking device automatically shuts off oil supply following any burner stoppage.

Of rugged construction . . . with all parts easily accessible for cleaning or renewing . . . these burners provide a flexible capacity range, with complete and efficient combustion under widely fluctuating loads.

TODD MANUFACTURES: Mechanical Pressure Atomizing Oil Burners—VEE-CEE Variable Capacity Burners—Horizontal Rotary Oil Burners—Oil Burning Air Registers for Natural, Assisted, Induced or Forced Draft—Inside Mixing Steam Atomizing Oil Burners—Combination Gas and Oil Burners—Furnace Doors and Interior Castings for Converting Howden Type Furnace Fronts to oil firing—Oil Burning Galley Ranges—Oil Heating, Pumping and Straining Equipment.

All installations of Todd Equipment are always *individually engineered* to fulfill specific requirements. *Send for descriptive literature.*

Todd engineers are always available for consultation and analysis of combustion problems—without obligation.

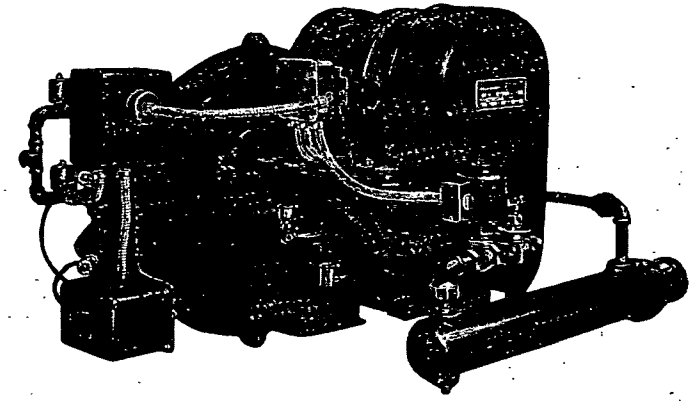
ENTERPRISE

ENGINE & FOUNDRY CO.

18th and Florida Streets • **San Francisco 10, Calif.**

Combustion Equipment Division

Distributors in all Principal Cities



Size "G"—Full Automatic with dual gas-electric ignition for No. 6 oil—capacity 115 B.H.P.

Enterprise Rotary Oil Burners are designed for efficient utilization of No. 3 and No. 5 oils, and when furnished with electric pre-heaters for No. 6, or Bunker oils. Approved by the Underwriters' Laboratories, the sizes range from 12 to 330 boiler hp in the pump type and to 650 boiler hp for gravity feed or with independent pumpset. By addition of a multi-jet gas head in the burner front plate, standard Enterprise Oil Burners can be converted to a combination unit where natural, manufactured, or process gas is available.

Enterprise Burners are available with manual, semi-automatic and full-automatic controls, and low fire, two-position or modulating controls.

Enterprise engineers have developed many applications for these burners where heat is required for industrial processing. The burners are designed for firing high or low pressure boilers of all types for commercial and industrial uses.

One of the features of the Enterprise Burner is the V-Belt Drive which offers a wide range of burner flexibility and makes possible the use of standard and modern type motors. The Enterprise design features durability, simplicity, quiet and economical operation.

S. T. Johnson Co.

940 Arlington Ave., **Oakland 8, Calif.**

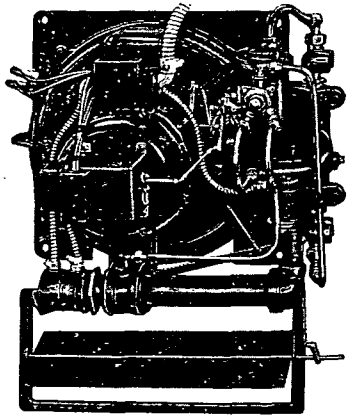
401 No. Broad St., **Philadelphia 8, Pa.**

Builders of Heavy-Duty Industrial Oil Burners

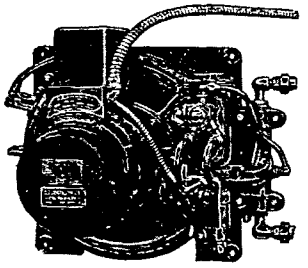
Johnson Industrial Burners are designed to operate on Heavy Oils which produce extra heat at low cost. They increase the capacity of equipment formerly fired with coal and produce desired steam pressures more quickly. Automatic regulation permits the boiler to operate with maximum efficiency at any specified steam pressure, without watching, care or attention, thus reducing labor costs.

They have been installed with marked success in hotels, hospitals, factories, office buildings and other large structures all over America because they provide heating engineers with a wide range of capacities and with every desired feature of economy, performance and automatic control.

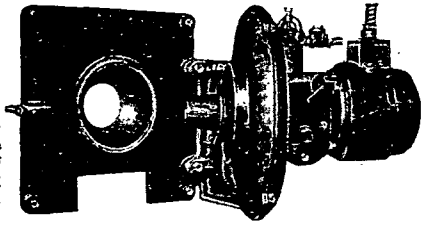
In their design and in their construction, Johnson Burners represent the new and advanced engineering and building techniques backed by 42 years of practical experience.



TYPE 30 AVH
Fully automatic. Pre-heater type. Burns No. 6 Oil. Six sizes, 20 to 800 horse power output.



TYPE 30 AV
Fully automatic. Burns No. 6 Oil. Six sizes, 2 to 100 g.p.h.



TYPE 28
Manual and semi-automatic. With or without built-in pumps. Burns No. 5 and No. 6 Oils. Seven sizes, 2 to 135 g.p.h. Illustration shows burner swung away from fire-hole-plate for easy inspection.

S. T. Johnson Co.

Builders of Domestic and Commercial Oil Burners

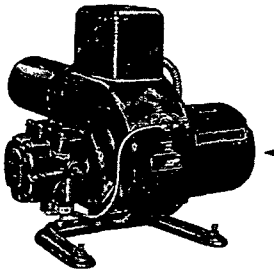
940 Arlington Ave., **Oakland 8, Calif.**

401 No. Broad St., **Philadelphia 8, Pa.**

Self-storage water heaters, separate burner units, burner-boiler units, conditioned air units, range burners and various specialized items comprise the line-up of Johnson light-oil Burners.

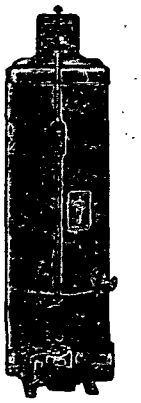
There is a wide range of sizes and capacities in each classification with which heating engineers and contractors can successfully meet every type of problem.

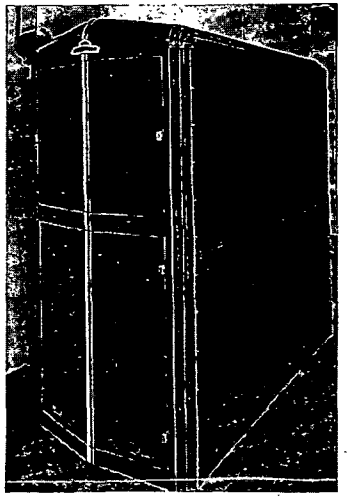
Every Johnson Burner is backed by an unbroken 42-year record of fine engineering and excellent craftsmanship.



BANKHEAT BURNERS
Fully automatic, pressure atomizing type. Sizes up to 20 gal. per hr.

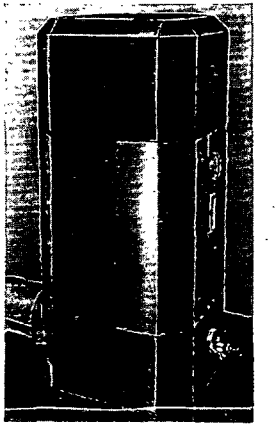
AQULUX WATER HEATERS
Fully automatic, self-storage. Cap.: 100 to 540 g.p.h.





SELECTAIR HEATERS
Hot air, hot water or steam heat. 5 sizes. Bankheat Burners.

ECONOLUX HEATERS
Heat and hot water. Fully automatic Bankheat Burners. 9 sizes.



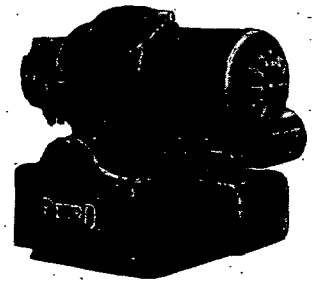
Petroleum Heat & Power Company

Main Office and Factory: **Stamford, Conn.**

Good Oil Burning Equipment . . . "Since 1903" . . . Fuel Oils

PETRO REG. U. S. PAT. OFF — DOMESTIC OIL HEATING EQUIPMENT

PRESSURE ATOMIZING DOMESTIC BURNERS



Petro P 20 Burner

Burner Model Number	Nozzle Size Gal per Hr.	*Total Capacity Steam Sq Ft	*Total Capacity Hot Water Sq Ft	App. Sh'p'g. Wgt.
P-20-A	1.00	350	560	155
P-20	1.25 To 2.50	435 To 875	695 To 1400	155
P-21	2.00 To 4.50	700 To 1575	1120 To 2520	160
P-22	3.00 To 6.00	1050 To 2100	1680 To 3360	170
P-12	6.00 To 10.00	2100 To 3500	3360 To 5600	215
P-13-A	9.00 To 12.00	3150 To 4200	5040 To 6720	285
P-13	12.00 To 18.00	4200 To 6300	6720 To 10080	285

Models P 20-A, P 21, P 22

In domestic heating this group of Petro burners has wide and general application to steam, hot water, or warm air systems.

Burns No. 3 fuel oil (or lighter), heaviest and lowest priced fuel oil approved by UNDERWRITERS for domestic use. They are precision-built atomizing (or "gun" type) burners, with constant electric ignition for reliable safe operation. May be installed with inside or outside fuel storage tanks without adding auxilliary pumping facilities. Order or inquiry should specify the type, size and rating of boiler or furnace to be fired, together with the total load.

Models P 12, P 13-A, P 13

Have wide application in heating large residences, stores, garages and other commercial buildings.

Approved for No. 3 (or lighter) fuel oil.

Petro P 13 Burner

Petro Boiler Units—both cast iron sectional and steel—in a range of five sizes covering normal domestic needs, and Petro Water Heaters and forced warm air furnaces are planned for production in early 1946. Information on this equipment available at that time. Send for catalog of Petro Domestic Oil Heating Equipment.

Petroleum Heat & Power Company

Main Office and Factory: **Stamford, Conn.**

Good Oil Burning Equipment . . . "Since 1903" . . . Fuel Oils

PETRO REG. U. S. PAT. OFF — INDUSTRIAL AND COMMERCIAL OIL BURNING SYSTEMS

FOR AUTOMATIC . . . SEMI-AUTOMATIC . . . OR MANUAL OPERATION

For Unheated No. 5 or Lighter Oil

Model WA—Automatic ignition and operation with synchronized control of oil and air.

Model W-SA—Semi-automatic, i.e.; automatic variation of firing rate with manual ignition; or for manual variation and manual ignition.

For Heated Oils: Heavy No. 5, No. 6 (bunker "C") oil

Model W-AH—Automatic ignition and operation with synchronized control of oil and air and of oil heaters.

Model W-SAH—Semi-automatic with oil heaters, i.e.; automatic variation of firing rate with manual ignition; also available for manual variation and ignition.

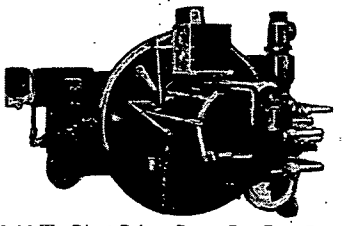
CAPACITIES

Model	Motor H.P.	Max. Gals. per Hour	Rated Cap. B.H.P.	Sq Ft C.I. Steam Rad.
W-2½	⅓	11	37	5,150
W-3	½	15	50	7,030
W-4	½	25	84	11,720
W-5	1	33	110	15,470
W-6	2	45	151	21,100
W-7	2	62	208	29,100
W-8	3	100	336	46,600
W-9	3	145	487	68,000

These burners are available for all electrical current characteristics.

PETRO'S THERMAL VISCOSITY CONTROL

The only dependable and accurate control of viscosity—and hence delivered combustion efficiency,—is through the heat applied to the oil. Petro's Thermal Viscosity System controls this heat-application at its source. Has been used successfully for years, without any need for frequent manual adjustment.



Model W—Direct Driven, Rotary Cup Type Burner Illustration Shows Modutrol Mounted on Burner

This burner is a self-contained assembly of motor, fan, pump, rotary cup atomizer and all air and oil adjustment apparatus.

Interlocking air and oil control mechanism permits any minimum or maximum operation required within the burner's range of operation. Counter-flow Angular Air Vanes at nozzle increase air and oil turbulence and aid efficient combustion of heavy fuel oils.

Special oil adjustment valve meters oil to rotary cup, yet permits manual operation without disturbing permanent burner adjustment.

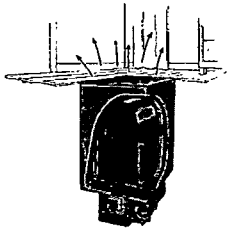
Removable rotary cup and nozzle permits changing shape of flame to suit requirements of any boiler and prevent flame impingement.

Send for catalog of Petro Commercial and Industrial Oil Burners

H. C. Little Burner Co.

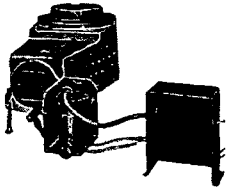
Head Office: **San Rafael, Calif.**

Factory Representatives in Principal Cities



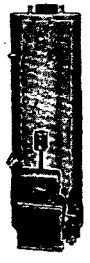
Fully Automatic Oil Burning Floor Furnace

These low cost units are factory assembled, including all controls, ready to hang in floor. Easy installation, no basement needed. Natural draft vaporizing burner, automatic operation, with electric ignition and thermostatic control. Inexpensive to operate. Listed by Underwriters' Laboratories for No. 3 furnace oil. Two sizes: No. **70-42**—47,250 Btu output (22 in. wide x 28 in. long x 44 in. high); No. **100-42**—75,000 Btu output (22 in. wide x 40 in. long x 45½ in. high).



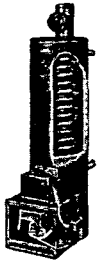
Natural Draft, Automatic, Vaporizing Burner

Natural draft vaporizing burner specifically designed for No. 3 furnace oil and listed by the Underwriters' Laboratories. Model shown (with controls attached) is fully automatic, with electric ignition (no pilot light) and thermostatic control. Manually operated models, which do not require electricity, are also listed for No. 3 oil. Seven sizes: 1.45 qts. to 2½ gal. per hour.



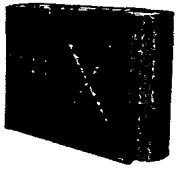
Aquatherm Water Heaters

Economical operation and ample capacity for all domestic requirements. Aquatherm "30" has 28 gal. storage, 40 g.p.h. recovery rate, is 19 in. wide x 21 in. deep x 56 in. Aquatherm "60" has 60 gals. storage, 160 g.p.h. recovery, is 24½ in. wide x 36 in. deep x 68 in. Both units listed by Underwriters' Laboratories for No. 3 furnace oil.



Coil Water Heater

For camps, auto courts, apartments or small hotels. Uses separate hot water storage tank. Also makes excellent boiler for hot water installations in home heating. Two sizes: No. 150—106,960 Btu output, 150 g.p.h. (80° rise); No. 250—160,440 Btu output, 250 g.p.h. (80° rise).

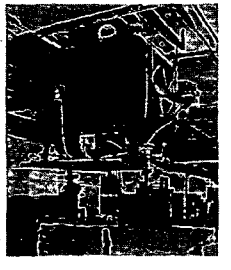


Winter Air Conditioner

Size A, 85,000 Btu output. Size B, 125,000 Btu output. Multivane blower, filters, vaporizing burner.

Ceiling Furnace

Industrial overhead heating saves floor space. Units are factory assembled, including vaporizing oil burner (no refractory brick saves much weight), thermostatic control, high capacity heat distributing fan with ¼ hp motor. Conforms to requirement of the Underwriters' Laboratories for use in garages. 140,000 Btu output.



A complete line of oil-fired small home heating units —plus special industrial units of moderate capacity.

Ray Oil Burner Co.

405 Bernal Avenue
San Francisco 12, Cal.

Since 1872

629 Grove Street
Jersey City 2, N. J.

Distributors in all Principal Cities

Consult your local Telephone Directory

Products: **A complete line of Horizontal Rotary and Pressure Atomizing Oil Burners; Combination Oil-or-Gas Burners; Industrial Gas Burners; Oil Burning Water Heaters; Winter Air Conditioning Units, Commercial Ranges.**

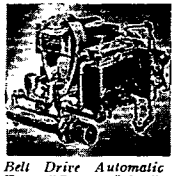
1 to 1000 hp Type AG. Man., Semi-Automatic.

RAY HORIZONTAL ROTARY OIL BURNERS

Built in fully automatic, semi-automatic and manual types; in sizes from 1 to 1000 Boiler hp; to burn all grades of fuel oil.

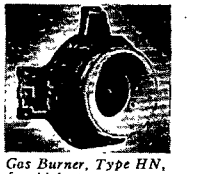
Standard models include both direct and belt drives—the latter being recommended for use where other than 50 or 60 Cycles AC, or only DC is available. Types for straight electric or straight gas ignition; pump or gravity feeds. Direct drive types include a steam turbine driven model.

All 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, for maximum reliability.

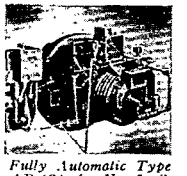


Belt Drive Automatic Type BR-141, # 6 oil.

Semi-Automatic, Type AP-14, for No. 5 oil.



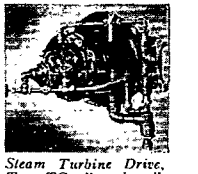
Gas Burner, Type HN, for high pressure gas.



Fully Automatic Type AR-134, for No. 5 oil.

RAY PRESSURE ATOMIZING OIL BURNERS

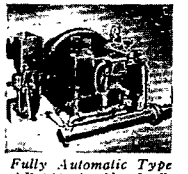
Fully automatic, for No. 3 oil or lighter. AC or DC; capacities to 18 gal/hr.



Steam Turbine Drive, Type TG, all grades oil.

RAY COMBINATION OIL-or-GAS BURNERS

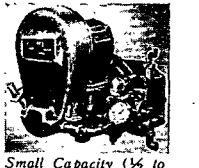
Operate on any grade of fuel oil, any grade of natural or manufactured gas. Oil burner sizes to 1000 boiler hp; maximum gas capacity: 43,600 cu ft/hr. Changeover may easily be made in 2 minutes or less.



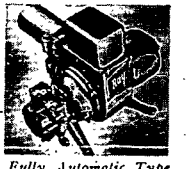
Fully Automatic Type AR-141, for No. 6 oil.

RAY INDUSTRIAL GAS BURNERS

For gas pressures above 1 lb/sq in. May be used alone or in combination with a Ray Oil Burner. Built in eleven sizes; in capacities to 43,000,000 Btu/hr.



Small Capacity (½ to 2½ gph) rotary burner.



Fully Automatic Type JP, # 3 or lighter oil.

RAY WINTER AIR CONDITIONING UNITS

Built in four sizes, with input capacities of 140,000, 230,000, 350,000, 450,000 Btu.

RAY OIL WATER HEATERS

Two sizes. *Capacities:* 45 and 75 gal. *Maximum recovery rates:* 154 and 240 gph.

Ray Oil Furnace, Winter Air Conditioning Unit.

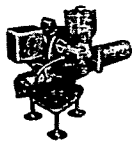
Williams Oil-O-Matic Division

EUREKA WILLIAMS CORPORATION, BLOOMINGTON, ILLINOIS

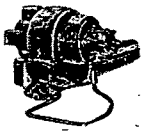
Manufacturers of Automatic and Manually Controlled Fuel Oil Burners — WILLIAMS OIL-O-MATIC HEATING COOLING PRODUCTS — Manufacturers of Year 'Round Air Conditioning Systems, Ice-O-Matic Compressors

OIL BURNERS

Williams Oil-O-Matic Lo-Pressure burners are offered in 5 sizes ranging in capacity from ½ to 15 gallons of fuel oil per operating hour. Easily installed in any type heating plant. Patented Thrift Meter pre-meters oil to exactly meet heating plant needs. Wide, non-clogging orifice handles even the heaviest oils.



Williams Hi-Pressure precision-built burners are available for those who want high quality at minimum cost. 3 sizes with capacities from .65 to 7 gallons per hour.



How to Decide Size of Burner

For low pressure domestic boilers, 1 gal of fuel oil per hour (140,000 Btu) is required for approximately 300 sq ft of steam radiation or its equivalent, or for 480 sq ft of hot water radiation or its equivalent. 70,000 Btu when using hot air furnace ratings. 24 sq ft steam boiler heating surface (or 2.2 hp). For exact data, see Oil-O-Matic Installation and Service Manual.

Oil-O-Matic Lo-Pressure Burners Leg or Flange Mt.

Model	Oil Capacity Min.	Oil Capacity Max.	Motor Hp	Motor Rpm	Atom. Pressure
K-150	.50	1.50	1/10	1800	2 lbs
K-3	1.00	3.00	1/10	1800	2 lbs
K-4.5	1.35	4.50	1/5	1800	2½ lbs
K-7	4.00	7.00	1/5	1800	3 lbs
J-1800	8.00	15.00	1/2	1800	4 lbs

Williams Hi-Pressure Burners Leg or Flange Mt.

HP-1	.65	1.50	1/16	1725	100 lbs
HP-3A	1.35	3.00	1/12	1725	100 lbs
HP-7	3.50	7.00	1/5	1800	100 lbs

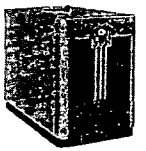
WINTER AIR CONDITIONERS

Williams Oil-O-Matic Winter Air Conditioners automatically draw in air, filter, heat and humidify it, and gently circulate the conditioned air to all rooms. 3 sizes fill all needs. Choice of Lo-Pressure or Hi-Pressure Burner.

Oil-O-Matic Winter Air Conditioners

Model	Btu at Bonnet	C.F.M. (a) ⅛" S.P.	Oil Input G.P.H.	Filters Sq In	Blower Motor
10	100,000	1300	1.00	1000	¼ hp
15	150,000	1800	1.50	1920	½ hp
25	250,000	2400	2.50	2400	⅝ hp

BOILER BURNER UNITS



Williams Oil-O-Matic Boilers are available in 3 models: Lo-Boy (illus.) in 4 sizes for steam, 4 for hot water; Water Base in 3 sizes for steam, 3 for hot water; Residential Steel in 4 sizes for steam, 4 for hot water. All units furnished complete with either Lo-Pressure or Hi-Pressure burners.

Oil-O-Matic Cast Iron Boiler Burner Units

Model	Net Load Ratings I.B.R. Steam Sq Ft	Net Load Ratings I.B.R. Water Sq Ft	Net Load Ratings I.B.R. Btu	Gross I.B.R. Oil Input G.P.H.	Gross I.B.R. Output Btu
LB-5S or 5W	360	580	86000	1.35	130000
LB-6S or 6W	450	720	108000	1.65	162000
LB-8S or 8W	620	990	149000	2.20	222000
LB-10S or 10W	780	1250	187000	2.75	278000

Oil-O-Matic Cast Iron Water Base Boiler Burner Units

16S or 16W	290	460	69000	1.05	105000
20S or 20W	350	560	83000	1.25	126000
24S or 24W	410	660	98000	1.50	147000

Oil-O-Matic Steel Boiler Burner Units

320S or 320W	320	510	76800	1.10	118000
400S or 400W	400	640	96000	1.40	146500
540S or 540W	540	860	129600	1.90	195000
680S or 680W	680	1090	163200	2.30	244000

WATER HEATERS

Oil-O-Matic Water Heaters compact, completely automatic, self-contained units. Lo-Pressure burner handles even heaviest oils for utmost economy.

Oil-O-Matic Water Heaters—Horizontal Type

Model	Storage Capacity	Heating Capacity Per Hr 90° Rise	Oil Input Per Hour Min.	Oil Input Per Hour Max.	Atom. Pressure
W.H.A.	35 gal	60 gal	....	½ gal	2 lbs
W.H.B.	35 gal	120 gal	½ gal	1.00 gal	2 lbs
W.H.C.	100 gal	210 gal	1.00 gal	1¾ gal	2 lbs

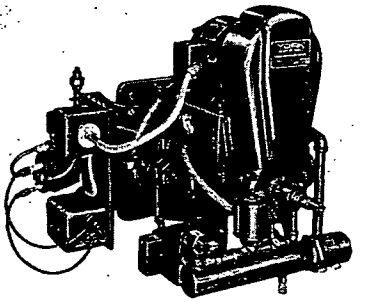
Ice-O-Matic Compressor Units: All temperature ranges. Capacities ¼ hp through 5 hp. Air or water cooled.
Air-O-Matic Absorption Units: 10 and 15 ton nominal capacities. Low pressure steam, water-chilled type, hermetically sealed units.

York-Shipley, Inc.

York, Pa.

YORK HEAT OIL-BURNING EQUIPMENT

INDUSTRIAL DIVISION



Model AHPBDM Pump-Type Burner for Bunker C oil, belt-driven, with fully-automatic and full-modulating firing control.

YORK HEAT INDUSTRIAL HORIZONTAL ROTARY TYPE OIL BURNERS

A range of sizes is available from 45 Boiler hp to 400 Boiler hp. These burners are built in both direct-drive and belt-driven motor applications. Models for using No. 5 and No. 6 oils are available for manual, semi-automatic, and full modulating control.

The basic frame sizes are listed as follows:

A Frame	45 hp to 60 hp
B Frame	60 hp to 125 hp
C Frame	125 hp to 225 hp
D Frame	225 hp to 400 hp

York Heat Horizontal Rotary Burners are equipped with many exclusive and patented advancements, designed to insure high efficiency and superior operation. Outstanding among these exclusive York Heat features are:

(*a*) The Iris Shutter, which gives absolute control of the burner air. It causes the metered volume of air to remain constant, insuring perfect combustion.

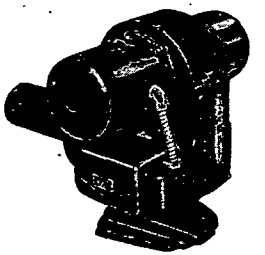
(*b*) The York Heat Flame Former, which makes possible accurate flame-adjustment, to suit the combustion chamber.

(*c*) The torch-type pressure igniter, for smoother and positive automatic starting.

DOMESTIC DIVISION

YORK HEAT SELF-CONTAINED BOILER-BURNER UNITS FOR FUEL OIL

A complete line of fully-automatic self-contained boiler-burner units is available, in capacities of 4 hp to 100 hp. These are constructed for both 15 lb and 100 lb steam operation. Larger sizes are equipped with an oil-burner for No. 5 or No. 6 fuel oil.



Model N-15 York Heat Domestic Oil Burner.

YORK HEAT DOMESTIC OIL-BURNERS

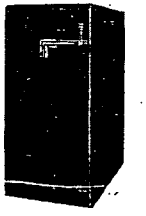
This complete line of pressure, atomizing-type burners carries the Underwriters' and Bureau of Standards approval for liquid fuels up to and including No. 3 fuel oil.

Models and capacities are listed as follows:

C-5	0.7 to 2.25 gph of oil
N-15	1½ to 4½ gph of oil
N-25	3 to 5½ gph of oil
N-35	5 to 10 gph of oil
T-45	8 to 18 gph of oil
T-55	16 to 30 gph of oil

YORK HEAT DOMESTIC HEATING UNITS

York Heat offers a complete line of domestic heating units, in a full range of capacities. These can be furnished for steam, hot water, and forced warm air applications.



York Heat Domestic Boiler-Burner Heating Unit.

The Vinco Company, Inc.

305 East 45th Street — New York 17, N. Y.

Only a clean boiler can be an efficient boiler. A clean boiler means saving fuel, as well as safeguarding boiler metal. Both are now a patriotic duty.

Boiler Cleanser 5 and 5 lb Cans

A positively harmless insoluble powder cleaner for new, remodeled and old heating systems. A unique, scientifically processed compound on a special formula not to be confused with other powder boiler cleaners.

What Vinco Boiler Cleanser Does

VINCO removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water *without the labor, expense, and uncertain results of blowing boilers over the top or of wasting returns.*

By this thorough cleaning Vinco prevents or cures foaming, priming, surging, and slow steaming.

How Vinco Boiler Cleanser Works

Each minute grain of VINCO powder adsorbs several times its own weight of oil, rust and dirt. These larger grains of adsorbed impurities then settle and are drained through the bottom according to directions on each can.

Vinco Guarantees

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients.
2. Purchase price is refunded if results are not as claimed when VINCO has been used according to directions.

Rust Preventer 1 qt cans only

VINCO RUST PREVENTER

When used after VINCO Boiler Cleanser has removed oil, grease, rust, scale and dirt, it will add and keep the rust inhibiting factors at the optimal constant for a year or more. (Test kit below has complete instructions and chart.)

Vinco Test Kit No. 10 (Patent applied for)

VINCO TEST KIT No. 10

for Testing and Treating Heating Boiler Waters

The kit enables the layman to make simple, rapid tests to diagnose and prescribe correct treatment of boiler waters right on the job.

A new time saving method that permits valid conclusions heretofore requiring complicated and often lengthy laboratory analysis and technique.

Each kit has sufficient material for complete tests on 100 jobs.

Refills cost about 2 cents for testing each job.

SPECIFICATIONS FOR COMPLETE VINCO TREATMENT OF NEW OR REMODELED STEAM, VAPOR, OR HOT WATER SYSTEMS

Do not use as a cleaning agent soda or any alkali, vinegar or any acid. Use Vinco.

1. AFTER THE SYSTEM IS TESTED AND TIGHT, USE THE PROPER QUANTITY OF VINCO LISTED.

After this first clean-out of any new or remodeled heating system, Vinco Boiler Cleaner need be used only if more piping, radiation, or another boiler is added to the original installation, or if the system is fouled by unwise cleaning or leak-sealing experiments.

2. After using Vinco Boiler Cleaner, Vinco Field Test Kit should be used to determine and apply the proper quantity of Vinco Rust Preventer. Vinco Rust Preventer should be applied annually or whenever the boiler water is drained for necessary repairs to the system.

SPECIFICATION FOR OLD HEATING SYSTEMS THAT DO NOT PERFORM PROPERLY

Diagnose and treat according to Vinco Field Test Kit. If a test kit is not available, consult table of quantities on this page and follow directions on Vinco cans.

SPECIFICATION FOR HOT WATER SYSTEMS

Use half quantities listed for treatment of steam systems to remove impurities. Then use test kit to determine proper quantity of Vinco Rust Preventer.

CONSULT THIS TABLE FOR NEW AND REMODELED HEATING SYSTEMS AND When a Vinco Field Test Kit No. 10 is not available if cleaning old heating systems.

QUANTITIES OF VINCO (IN POUNDS) REQUIRED FOR HEATING SYSTEMS

(Note that quantities are based on actual installed radiation, not on boiler capacity.)

Sq Ft of Radiation	For Steam or Vapor Systems, to prevent or cure priming or foaming. Also for Hot Water Heating Systems Maintained at approx. 200 F or above.	Annually, to remove rust scale, dirt and for Hot Water Systems below 200 F.
up to 350	3	1½
351 " 600	5	2½
601 " 1100	8	4
1101 " 1400	10	5
1401 " 1800	13	6½
1801 " 2100	15	7½
2101 " 2700	18	9
2701 " 3100	20	10
3101 " 3700	23	11½
3701 " 4200	26	13
4201 " 4600	28	14
4601 " 5000	30	15
5001 " 5300	31	15½
5301 " 5600	32	16
5601 " 5900	33	16½
5901 " 6200	34	17
6201 " 6500	35	17½
6501 " 6800	36	18
6801 " 7100	37	18½
7101 " 7400	38	19
7401 " 7700	39	19½
7701 " 8000	40	20
8001 " 8300	41	20½
8301 " 8600	42	21
8601 " 8900	43	21½
8901 " 9200	44	22
9201 " 9500	45	22½
9501 " 9800	46	23
9801 " 10100*	47	23½

*Above 10100 sq ft use an additional pound Vinco for each additional 300 sq ft of actual installed radiation.

Soot-Off—1 lb cans 60 and 100 lb drums

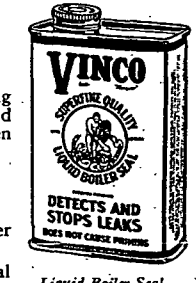
VINCO SOOT-OFF

Safely and thoroughly removes the insulating blanket of soot on fire pot, flues and chimney. It also insures against external corrosion (caused by dampness and soot forming sulphuric acid during summer layoff.) No dangerous chemicals.

REMOVE SOOT WITH VINCO SOOT-OFF SEVERAL TIMES A YEAR

VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid seal. Unique in that it does not induce priming and foaming. It has no unpleasant smell. Makes speedy and permanent repairs of boiler and heating system leaks. Fine to tighten up new jobs. Directions simple.



Liquid Boiler Seal 1 qt. cans only

Quantities

Steam and Vapor Systems—Use 1 quart VINCO Liquid Boiler Seal to each 6 sq ft *grate* area.

Hot Water Systems—Use 2 quarts VINCO Liquid Boiler Seal to each 6 sq ft *grate* area.

McDONNELL & MILLER

Safety Devices for Steam and Hot Water Boilers and Liquid Level Controls

General Offices: Wrigley Building, Chicago 11, Illinois

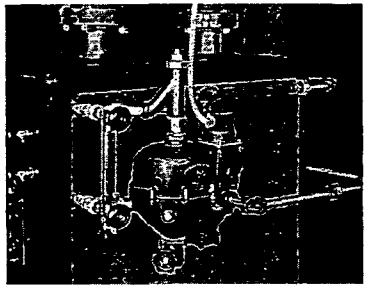
"Doing one thing well"

PRODUCTS:

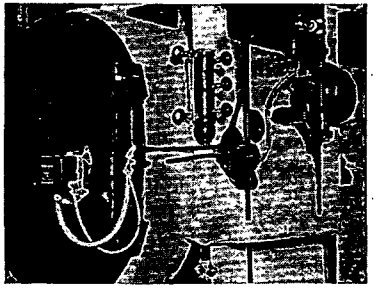
Boiler Water Feeders; Feeder Cut-off-Combinations; low Water Fuel Cut-offs; Pump Controls; Low Water Alarms; Humidifier Water Level Controls; Safety Relief Valves for hot water heating boilers and storage tanks; High and Low Oil Switches and Liquid Level Controllers for a wide range of services.

The service range and types of installations covering most common applications of McDonnell Boiler Water Level Controls are described here. For facts concerning protection of process boilers, or any special information, consult our engineering department.

McDonnell Combined Boiler Water Feeder and Low Water Cut-off



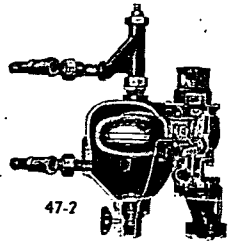
McDonnell No. 47-2 for heating boilers under 5000 sq. ft. capacity. Maximum steam pressure, 25 lbs.



McDonnell No. 51-2 for heating boilers over 5000 sq. ft. capacity. Maximum steam pressure, 35 lbs.

A typical installation of the McDonnell No. 47-2 Combined Boiler Water Feeder and Switch is illustrated above. The water feeder maintains safe level by feeding when water line drops. In case of foaming and priming or failure of water supply, causing water level to drop to ¼ in. in the water glass, low water switch comes into action. For automatically fired boilers switch may be wired to cut current to burner. For hand fired boilers switch may be wired to complete alarm bell circuit. Also furnished for hand fired boilers without switch—designated No. 47.

Installation time is cut to minimum by Quick-Hook-Up Fittings for gauge-glass installation. Valve mechanism is isolated from the heat of the float chamber. Large area straight-through blow-off valve is standard equipment.



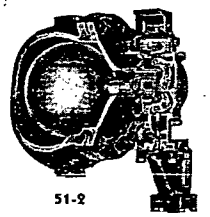
47-2

The McDonnell No. 2 Switch used in the No. 47-2 is Underwriters' approved. Electrical rating: a-c., ¾ hp 110-220 V.; d-c., 10 amp. 115 V.

The illustration above shows a typical installation of the McDonnell No. 51-2 Combined Boiler Water Feeder and Switch used for larger boilers. This combination functions the same as the No. 47-2, the feeder taking care of normal requirements and the switch safeguarding against emergencies by cutting off the burner in automatically fired boilers or completing a low water alarm circuit for hand fired installations. The feeder is also furnished for hand fired boilers (as No. 51) without low water cut-off and alarm switch.

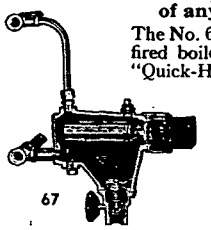
Basic features of the No. 51-2 are the same as in the No. 47-2, except that it is installed with 1 in. equalizing pipe instead of the McDonnell "Quick-Hook-Up" and has the larger feeding capacity required for larger boilers.

For boiler operating at pressures from 35 lb to 75 lb use the McDonnell No. 53-2 (with switch) or No. 53 (without switch). Electrical rating of switches used in No. 51-2 or 53-2 is the same as previously given for the No. 47-2.



51-2

McDonnell No. 67 Low Water Cut-off for automatically-fired steam boilers of any size. Maximum steam pressure, 25 lb



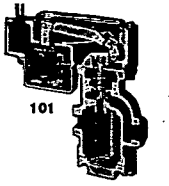
67

The No. 67 takes care of low water cut-off requirements of automatically fired boilers with steam pressure below 25 lbs. It has the McDonnell "Quick-Hook-Up" for quick, easy and trouble-proof installation in the gauge glass tappings; deep sediment chamber with large quick-opening blow-off; packless, non-binding construction; adjustable terminal box to make wiring neat and easy; dependable, snap-action, twin switches.

One switch **closes** on small drop without stopping burner. It can be used to complete an alarm bell circuit, warning of approaching low water, or can be used to complete circuit to the McDonnell No. 101 Electric Water described below. Second switch cuts current to burner if water level drops to ¼ inch in gauge glass.

Underwriters' ratings for both switches: a-c., ¾ hp, 110-220 V.; d-c ¼ hp, 115-230 V.

McDonnell No. 101 Electric Boiler Water Feeders for boilers up to 5000 sq. ft. capacity

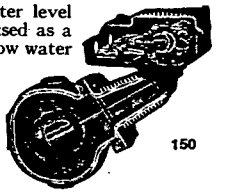


101

The No. 101 is an electric water feeder for use with No. 67 Low Water Cut-off or with McDonnell "built-in" Low Water Cut-offs which are standard equipment on many modern heating boilers. It converts the cut-off into a combination boiler water feeder and low water cut-off as described on the opposite page.

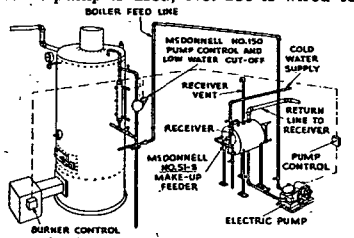
McDonnell No. 150 Pump Control, Low Water Cut-off and Alarm Switch for boilers of any size. Maximum steam pressure, 150 lb

This boiler water level control may be used as a pump control, a low water cut-off, a low water alarm switch, or any combination of these functions. Electrical ratings are: a-c., 1 hp, 110-220 V., d-c, ¼ hp, 110-220 V.—as pump control or cut-off; 1 amp. 110 V, a-c. or d-c—as low water alarm. No. 150 has automatic reset low water cut-off switch but may be ordered with manual reset as No. 150-M.



150

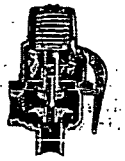
A typical hook-up of the McDonnell No. 150, controlling an electric pump and providing low water cut-off, is shown opposite. When water level drops ¾ in. below normal, No. 150 starts pump and then stops it when normal water line has been restored. If emergency occurs and water level falls to arrow mark on body of control, cut-off switch stops burner. If steam pump is used, No. 150 is wired to control electric valve in steam line to pump. Note that McDonnell No. 51-S Feeder is used to supply make-up water to the pump receiver.



Drawings available covering use of No. 150 on two or more boilers supplied by a common feed pump; ask for No. 150 Data Book.

McDonnell Safety Relief Valves

The No. 29 series of Safety Relief Valves have "snap action"—a mechanical means of opening full orifice capacity at precise instant set pressure is reached. Resulting large discharge capacity makes it feasible to rate them in heat dissipating capacity so they can be matched to Btu. output of unit on which they are installed, thus preventing over-pressures and protecting against explosions.



No. 29

Stainless steel cone replaces conventional composition disc. Long-lived, compact bellows replaces broad diaphragm of ordinary valves. Testing lever is easy to operate. All valves comply with *A.S.M.E.* code.

(1) For Hot Water Heating Boilers

Valve No.	Opening Pressure	Btu. Output
29	29 lbs	156,000
129	29 lbs	350,000

(2) For Domestic Hot Water Heaters and Tanks

Valve No.	Opening Pressure	Max. Water Supply Pres.	Btu. Output
229	75 lbs	50 lbs	316,000
329	100 lbs	75 lbs	380,000
429	150 lbs	100 lbs	432,000

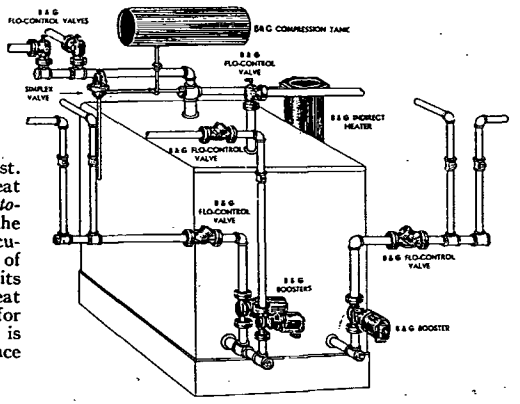
Bell and Gossett Company

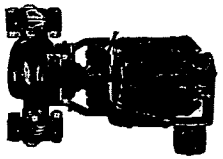
Morton Grove, Illinois

HOT WATER SYSTEMS AND SPECIALTIES

B & G Forced Hot Water Heating Systems

To a postwar world demanding improved ways of doing old things, B & G Forced Hot Water Heating contributes better heating at lower installation and operating cost. It is a system in which the heat input is *accurately and automatically controlled* to equal the heat loss. Water can be circulated through a long range of temperatures, hence permits close adjustment of the heat supply to the actual need for heat. Operating equipment is extremely simple—an assurance of dependable performance.



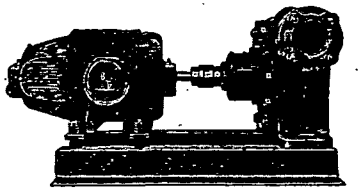


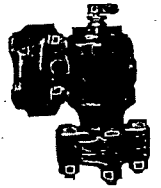
B & G Boosters

An electrically-driven centrifugal pump, used to mechanically circulate water through B & G Forced Hot Water Heating Systems. It is distinguished by genuine oil lubrication, patented water-tight seal, exceptionally quiet operation and precision manufacture throughout.

B & G Universal Pumps

The B & G Universal Pump is designed for large forced hot water heating systems in apartment buildings, office buildings, factories, schools, etc. The installation can be operated as a large single zone or divided into several zones in which circulation of the pumped water in each circuit is controlled by a B & G Motorized Valve, operated by a zone thermostat.



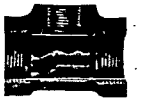


B & G Angle Flo-Control Valves

This valve, installed in the main, shuts off circulation to radiators when heat is not needed, permitting summer operation of a B & G Indirect Water Heater. It also helps maintain a uniform room temperature during the heating season.

B & G Motorized Valves

Thermostatically operated valves used for controlling boiler water flow through the individual circuits of zoned heating systems.

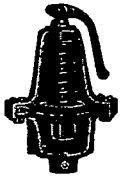


B & G Monoflo Fittings

B & G Monoflo Fittings permit the use of a *single* pipe main, instead of the conventional flow and return lines. It is installed at the junction of the radiator risers to the single main and assures the diversion of the proper amount of heated water into each radiator. Savings in space, labor and materials are obviously effected.

SEE THE B & G HANDBOOK FOR COMPLETE DESIGNING DATA

HOT WATER SYSTEMS AND SPECIALTIES



B & G Relief Valves

For relieving excess boiler pressures in hot water heating systems, and in the lines of service water systems. B & G Relief Valves have the design features which assure dependable service.

B & G Reducing Valves

Fast operating valves for keeping hot water heating systems properly filled. Easily adjusted to meet varying building heights. Also high pressure reducing valves for protection of plumbing fixtures.

B & G Compression Tanks

Essential equipment for closed hot water heating systems. Expansion of heated water is taken up by the tank, providing a cushion of compressed air against sudden pressures and water hammer shocks. On high temperature installations, the Tank develops sufficient pressure to prevent boiling of the water in the system.

B & G Indirect Water Heaters

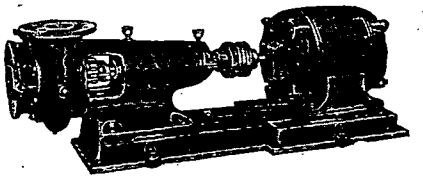
Any steam, vapor or hot water heating boiler can be equipped with a B & G Indirect Water Heater. With the proper electrical controls, the Heater will furnish an ample supply of hot water, *winter and summer*, at very low operating cost. Heater must be used with a storage tank of suitable capacity.

B & G Type "SU" Instantaneous Water Heaters

For heating water with steam. Ideal 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.

B & G Steam Convertors

Extensively used when steam is required in the factory for power or process work, but where the benefits of *mechanically circulated hot water* are desired for the heating system. Steam is passed through the convertor shell, heating the water circulated through the tubing.



B & G Centrifugal Pumps

Design and construction based on years of experience in the industrial field. Rugged, compact units—built to stand up under the strain of continuous operation. Available with semi-open or enclosed impellers—motors flexible coupled or integral with pump. Send for Catalog.

Send for Your Copy of the B & G Handbook

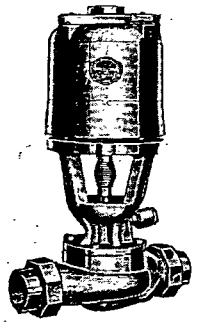
A single authoritative source of information on the design and installation of Forced Hot Water Heating Systems and Service Water Heating Systems. Packed with data and tables of every day value. Your copy will be sent on request.

H. A. Thrush & Company

Peru, Indiana

Representatives in Principal Cities

FORCED CIRCULATING THRUSH FLOW CONTROL SYSTEM OF HOT WATER HEATING AND HEATING SPECIALTIES

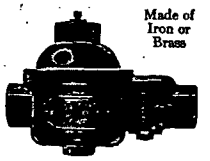


Patent Nos. 2,054,009, 2,111,441, 2,257,867, 2,356,482

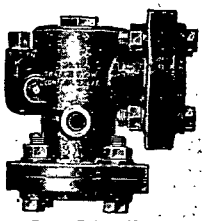
Flexible, economical of fuel, forced circulating Thrush Summer-Winter System of Hot Water Heat is the most satisfactory way to heat buildings. Thrush Systems, Water Circulators, Water Heaters, Pressure Relief and Pressure Reducing Valves and other fuel-saving heating specialties are available. Write today for information or engineering assistance.

THRUSH WATER CIRCULATORS

Five sizes, 1 in., 1¼ in., 1½ in., 2 in. and 3 in., for forced circulation in Hot Water Heating and Domestic Water Systems. Save fuel, insure uniform heating.



Made of Iron or Brass



Patent Reissue No. 19,873

THRUSH FLOW CONTROL VALVES

For use with Thrush Water Circulators

This patented valve prevents circulation when not required. Six sizes, 1 in., 1¼ in., 1½ in., 2 in., 2½ in. and 3 in. Work automatically by pressure head, generated by Thrush Circulator.

Number 4 Illustrated

THRUSH LOW PRESSURE WATER RELIEF VALVES

Protect heating boilers from excess pressures. Made in angle or straight types, of iron or brass, sizes ½ in., ¾ in. and 1 in. Unfailing dependability has been proved by over a quarter of a century of successful operation.

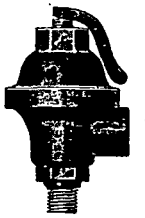
THRUSH PRESSURE REDUCING VALVES

Types for High and Low Pressures

Sizes, ½ in., ¾ in. and 1 in. Low pressure reducing valves to reduce pressure of water entering heating system and maintain water supply in system automatically. High pressure reducing valves for protecting house plumbing and heating equipment from excessive city line pressures.

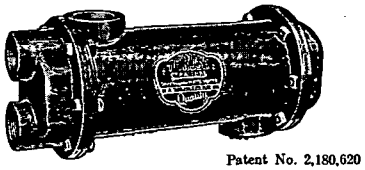
THRUSH AIR-TIGHT PRESSURE TANKS

An essential part of every hot water heating system. Conserves water and fuel, because the heated water expands into the Thrush Pressure Tank and returns to the system as it cools. Adds to the continued operating efficiency of the heating system over a long period of time.



THRUSH HIGH PRESSURE WATER RELIEF VALVES

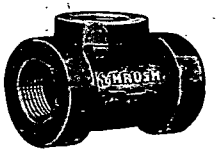
Protect water supply or range boilers and gas or electric water heaters from excess pressure and temperature (some types). Made in angle or straight types, of iron brass, sizes, ½ in., ¾ in. and 1 in., for pressure relief only or combination pressure and temperature relief.



Patent No. 2,180,620

THRUSH WATER HEATERS

Highly efficient heat exchangers or converters. Sixteen sizes, for Hot Water or Steam. Pressure up to 150 lb. Straight tubes readily cleanable. Provide Domestic Hot Water at low cost. Also used industrially for heating or cooling liquids.

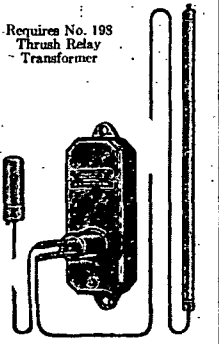


THRUSH SUPPLY TEES FOR ONE PIPE SYSTEMS

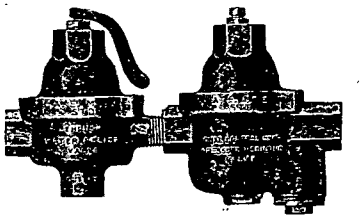
Assure positive diversion to radiators. Made in four sizes, 1 in., 1¼ in., 1½ in. and 2 in., with branch outlets of ½ in., ¾ in. or 1 in.

No. 210 THRUSH DIFFERENTIAL TEMPERATURE CONTROL

Requires No. 198 Thrush Relay Transformer

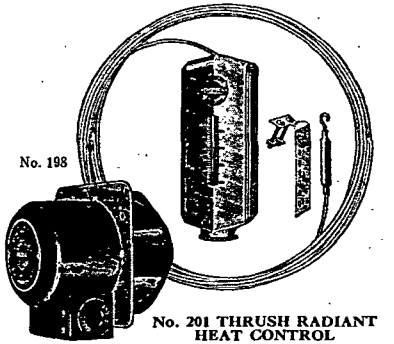


Special dual acting thermostat—one bulb inside and one bulb outside building. Automatically maintains correct indoor temperature with any weather change. A real fuel- and money-saver, especially for apartment houses, or groups of industrial, tourist camp or other buildings with one central heating plant.



THRUSH DUAL CONTROL UNITS

Provide automatic pressure relief, reduce line pressure, automatically fill and maintain water supply in hot water heating system. Built-in strainer. Made in four types, brass or cast iron, ½ in. or ¾ in.

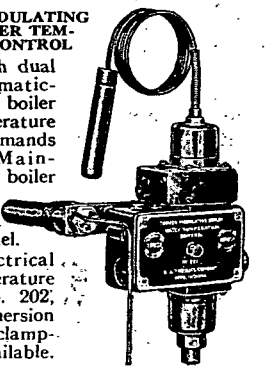


No. 198

No. 201 THRUSH RADIANT HEAT CONTROL

Automatically maintains room temperature within a fraction of a degree. Controls both room and water temperature in the radiators, compensating to prevent variation in room temperature or a lack of radiant heat. Requires No. 198 Thrush Relay Transformer for low voltage.

THRUSH MODULATING BOILER WATER TEMPERATURE CONTROL



No. 203 with dual bulbs, automatically varies boiler water temperature to meet all demands for heat. Maintains correct boiler water temperature, thus saving fuel.

Other electrical water temperature controls, No. 202, No. 205 immersion and No. 206 clamp-on type, available.

WRITE FOR COMPLETE CATALOG

Taco Heaters, Incorporated

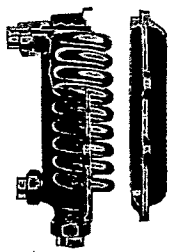
342 Madison Avenue, **New York 17, N. Y.**

TACO HEATERS OF CANADA, LTD., 24 Adelaide St., W., **Toronto**

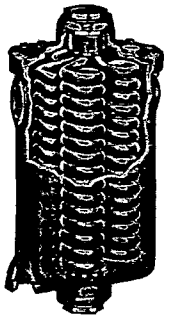
THERE is a storage or tankless, Biltin or external, type Taco indirect water Heater for every job—for use with patented Taco-Abbott System.

Biltin Taco Heaters, which are standard equipment on leading heating boilers, are catalogued only in the boiler catalogues. For additional information on Biltin Tacos write boiler manufacturer or Taco.

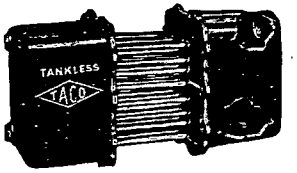
Complete catalogue information on external Taco Heaters is also available for home, apartment house, hotel and housing projects.



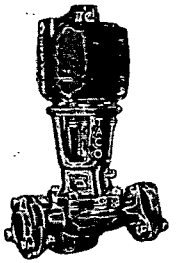
Domestic Tacos



Multi-coil Taco



Tankless Taco Nos. 12, 14, 16 and 18

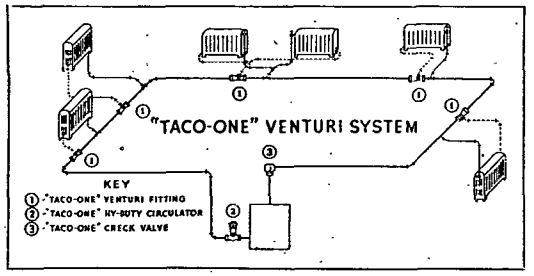


"Taco-One" Venturi System

A forced circulating warm heating system using a single pipe main from boiler to radiators and back again. Small pipes and one pipe main make a neater job, reduce installation costs. Radiators can be placed above and below main. 85 per cent of residential jobs need only one circuit which requires no balancing valves. This revolutionary heating system is made possible by the Taco Venturi Fitting and remarkable Taco Hy-Duty Circulator.

Taco Venturi Fitting—Acts as a suction pump in which the only moving part is water. Water does the trick by producing a vacuum pull that draws water through each radiator, giving positive uniform circulation.

Taco Venturi Fitting (Cross Section)



Buffalo Pumps, Inc.

450 Broadway, **Buffalo, N. Y.**

Sales Representatives

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ATLANTA 2, GA., Mr. J. J. O'Shea, 305 Techwood Drive
BALTIMORE 2, MD., Mr. C. A. Conklin, III, 508 St. Paul St.
BOSTON 76, MASS., Mr. E. D. Johnson, 507 Main St., Melrose Station
CHICAGO 6, ILL., Emmert & Trumbo, 20 N. Wacker Drive
CINCINNATI 2, OHIO, Mr. F. W. Twombly, 626 Broadway
CLEVELAND 13, OHIO, Mr. T. A. Weager, 418 Rockefeller Bldg.
KITCHENER, ONTARIO, Canada Pumps, Ltd., Mr. A. S. Capwell
DALLAS 1, TEXAS, Mr. T. H. Anspacher, 1801 Tower Petroleum Bldg.
DAVENPORT, IOWA, D. C. Murphy Co., Inc., 305 Security Bldg.
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NEW ORLEANS 12, LA., Devlin Bros., 1003 Maritime Bldg.
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NEW YORK 7, N. Y., Mr. W. A. Wagner, Room 1113, 39 Cortlandt St.
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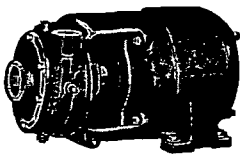
SAN ANTONIO 6, TEXAS, Langhammer Rummel Co., 436 Main St.
SEATTLE, WASH., Consolidated Service & Supply Co., 71 Columbia St.
ST. LOUIS 3, MO., J. W. Cooper, 2726 Locust St.
TOLEDO, OHIO, Mr. C. M. Eyster, 1118 Madison Ave.
SAN FRANCISCO 11, CALIF., Harry W. Parsons, Inc., 24 California St.
WASHINGTON 5, D. C., Mr. G. S. Frankel, 512 Woodward Bldg.

PRODUCTS—A complete line of Single and Multi-stage Centrifugal Pumps and Special Pumps for use in all types of heating and air conditioning installations.

Buffalo Single Suction Close-Coupled Pumps

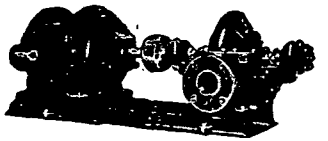


This pump is close-coupled to electric motor, eliminating the necessity for bearings. The impeller is overhung on the motor shaft, providing a compact, easily-serviced unit. Permanent alignment is assured and the pump mounted in this manner requires very little space.

Buffalo Close-Coupled Pumps are suitable for handling hot water with low submergence on suction, or for operating with suction lift as high as 25 ft.

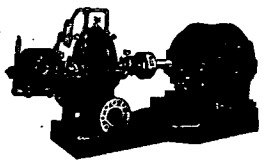
These pumps are also available in special alloys.

Buffalo Double Suction Single Stage Centrifugal Pumps



For general service where clear water is handled you will get top performance with these pumps. They embody all of the accepted modern features of centrifugal pump design. Capacities range from 10 to 50 thousand U.S. gallons per minute.

Buffalo Self-Priming Single and Double Suction Centrifugal Pumps

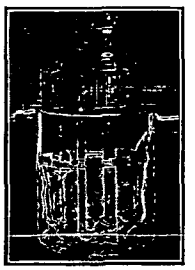


Now available with positive self-priming device built with the pump. This primer is built under license from the Nash Engineering Company and is fully covered by patent.

Buffalo Self-Priming Pumps offer these advantages: (1) All working parts are above the liquid to be pumped. (2) There is complete access to all parts of installation. (3) Rotors are balanced—vibrationless. (4) Buffalo Self-Priming Pumps are very quiet—no long shafts to vibrate and fewer bearings. (5) Constant positive prime obtained without foot valves.

Buffalo Automatic Sump Pumps

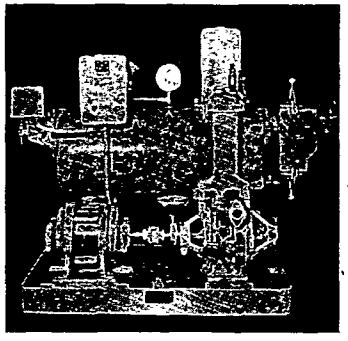
Buffalo Sump Pumps are self-contained and have unusually high efficiencies thus permitting the use of small motors. Ball bearing thrust and enclosed shaft especially adapt these pumps for their service.



The Nash Engineering Company

234 Wilson Road — **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

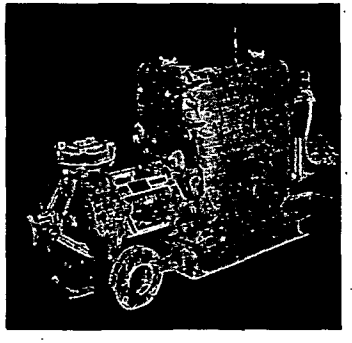


Return Line Vacuum Heating Pump

Standard with the heating industry for over seventeen years. Removes air and condensation from return lines of vacuum steam heating systems, discharging air to atmosphere and returning water to the boiler.

Two independent units are combined in a single casing—an air unit and a water unit. Impellers of both are mounted on the same shaft. Pump is bronze fitted throughout.

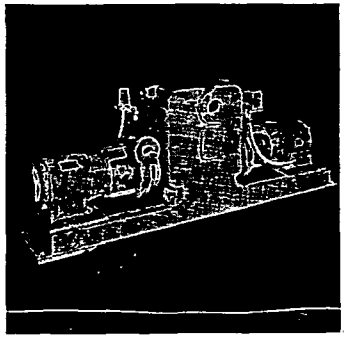
Supplied direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation. Standard in capacities up to 300,000 sq ft E.D.R. Larger units special. Bulletins Nos. 307, 308, 309, and 310 on request.



Vapor Turbine Vacuum Heating Pump

Jennings Vapor Turbine Heating Pumps combine all advantages of the standard return line heating pump with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss.

This pump affords the safety and economy which goes with continuous condensation return and steady vacuum, and at no cost for electric current. Furnished standard in capacities up to 65,000 sq ft E.D.R. Larger units special. Bulletin No. 290 on request.



Condensation Pump and Receiver

Removes the condensation from radiators in return line steam heating systems, particularly radiators set below the boiler water line level, and pumps the condensation back to the boiler. Pump is bronze fitted with enclosed centrifugal impeller of improved design. By making the pump casing a part of the return tank, and bolting the motor base to the tank, floor space is conserved. The rectangular construction permits installation in a corner against the wall.

These pumps are furnished in standard sizes with capacities ranging from 1½ to 225 gpm of water. For serving up to 150,000 sq ft of equivalent direct radiation. Bulletin No. 319 on request.

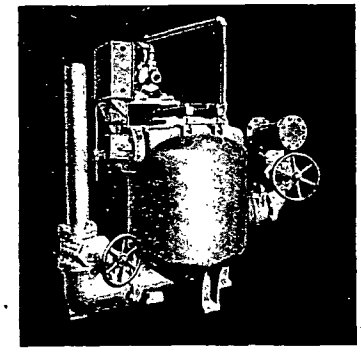
The Nash Engineering Company

234 Wilson Road — **South Norwalk, Conn., U. S. A.**

Sales and Service Offices in all Principal Cities

SEWAGE EJECTOR

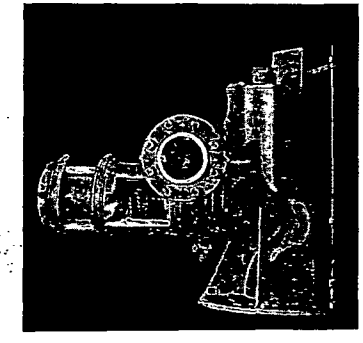
For pumping unscreened sewage or drainage from basements below the street sewer level, handling crude sewage from low level districts, pumping effluent, sludge and other heavy liquids. The Jennings Sewage Ejector is of the pneumatic type. Air, compressed only to the pressure at which it is used, by a Nash Hytor Air Compressor, is motive power to pump the accumulated sewage from a pot to the sewer. There are no air storage tanks, reciprocating air compressors or screens, no air valves. Furnished in several standard sizes up to 1500 gal. per minute against heads up to 100 ft. Bulletins on request.



Suction Sump and Sewage Pumps

Jennings Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. Sewage Pumps are equipped with non-clog type impeller for liquids containing solids. Suction piping only is submerged. Centrifugal impeller and vacuum priming rotor are mounted on same shaft that carries rotor of the driving motor, forming a single moving element, rotating without metallic contact.

Will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit, affording perfect accessibility for inspection or cleaning. Capacities to meet all requirements. Bulletins Nos: 159, 161, and 338 on request.

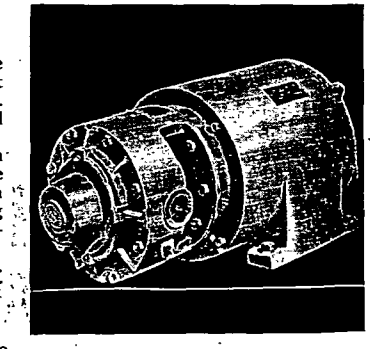


Air Compressor and Vacuum Pump

Nash Air Compressors operate on a unique and different principle. The one moving part rotates in casing without metallic contact. There is nothing to wear, and no internal lubrication.

Nash Compressors deliver absolutely clean air; ideal for agitation of liquids, pressure displacement, and handling gases. Vacuum pumps ideal for priming pumps, blood sucking pumps in hospitals, and wherever non-pulsating vacuum is required.

Pressure 75 lb or vacuum 27 in. of mercury. Furnished for any capacity; special for higher vacuums and pressures. Bulletins Nos. 282, 325, 331 and 337 on request.



Chicago Pump Company

2330 Wolfram Street BRUnswick 4110 Chicago 18

PRODUCTS—Return Line Vacuum Heating and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic and Tankless Water Supply Systems and Automatic Alternator for Duplex Sets of Pumps.

"CONDO-VAC"

Return Line Vacuum and Boiler Feed Pump for Heating Systems

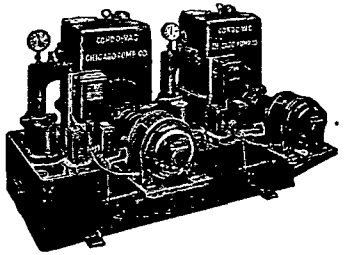


Fig. 2102—Duplex "Condo-Vacs" with Duplex Double Automatic Control

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a "Condo-Vac." "Condo-Vac" reduces corrosion in piping and boiler to minimum—because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. "Condo-Vac" is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain. *Ask for bulletin 270.*

Close-Coupled Pumps

Boiler Feed, Circulating, Tank Filling, Water Supply

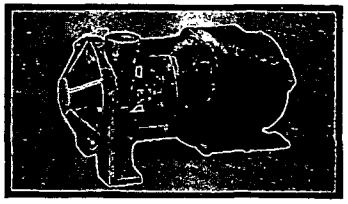


Fig. 2150—Close-Coupled, side suction pump. Capacities range from 3 to 600 Gpm against heads up to 189 ft. Motors from 1/6 to 20 Hp. Discharge 1 to 5 in. Closed and open type impellers. Bulletin 108.

"Sure-Return" Condensation Pump

for Low and Medium Pressure, and Systems up to 75,000 Sq Ft Radiation

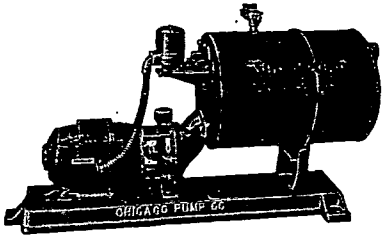


Fig. 1946

"Sure Return" Condensation Pumps and Receivers are built for systems up to 75,000 sq ft of direct radiation and for low and medium pressures. Built in either single or duplex units. Duplex units are alternated in their operation by the Automatic Alternator. *Complete data in Bulletin 250.*

Vertical Condensation Pumps

for Low and Medium Pressure for Systems from 500 to 100,000 Sq Ft Radiation

Fig. 1940 Vertical Condensation Pump

The vertical condensation pump is designed to receive returns from lowest radiation. The receiver is placed underground—an ordinary hole sufficing if necessary — and requires very little floor space. Unit is shipped complete, easy to install, assembled so as to prevent steam leaks. Special bearings will stand up under hot water for several years. A special float mechanism is guaranteed not to leak or stick in stuffing box. *Complete data and description in Bulletins 245, 253 and 255.*

Barnes & Jones

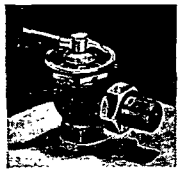
INCORPORATED

129 Brookside Avenue Boston, Mass.

New York Office: 101 Park Avenue

Barnes & Jones Vapor and Vacuum Systems of Steam Heating; Modulation Valves; Adjustable-Orifice Radiator Valves; Packless Quick-Opening Radiator Valves; Thermostatic Radiator Traps; Thermostatic Trap Replacement Units; Condensators (Boiler Return Traps); Float and Thermostatic Traps; Strainers; Damper Regulators; Gages; Systems of Zone Control for Steam Heating. Complete Catalog on Request

Modulation Valves, Series K—Packless Quick Opening Valves, Series F



Series K Valves have non-tarnishable indicating dial, non-rising stem, renewable disc seat. Tail piece extra heavy. Extra long to facilitate installation. Three models: lever handle, wheel handle, lock shield. Series F Valve furnished with wheel handle only.

Series K Valve

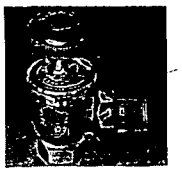
Size	½"	¾"	1"	1¼"
Cap. Sq Ft Rad.*	30	60	100	180

Series F Valve

Size	½"	¾"	1"	1¼"	1½"	2"
Cap. Sq Ft Rad.*	30	60	100	180	270	400

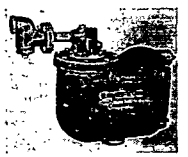
*Based on 2 oz pressure differential.

Adjustable Orifice Valves, Type H



May be adjusted for different capacities after installation. At all times provides indication of the adjustment. Operation is quiet. Unauthorized tampering with adjustment is virtually impossible.

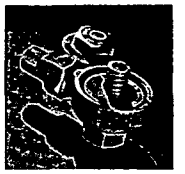
Condensators



For returning water of condensation to boiler from open return line systems independently of boiler pressure, without change in operating conditions, air binding, or admitting steam to the return side.

No.	31	32	33	34	35	36
Cap. Sq Ft Rad.*	700	1600	3500	6000	10,000	16,000

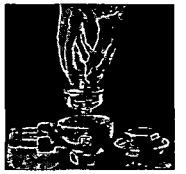
Thermostatic Radiator Traps



Sturdily made to precision standards. Sensitive in operation. Provide instant discharge of air and water, prevents passage of steam. Contains unique Cage Type Thermostatic Unit, which carries its own thermostatic element, valve piece and valve seat, factory calibrated and locked in correct adjustment.

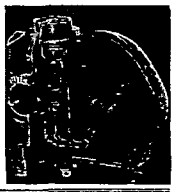
Trap Size and Number	Capacity Sq Ft. @ 1½#	Patterns
½" x ½" #122	200	Ang.-R & L Corner-St'w'y.
¾" x ¾" #134	400	Ang. & St'w'y.
1" x 1" #147	700	Angle only.

Thermostatic Radiator Cage Replacement Units



Offer complete and reliable trap renewal in practically every make of thermostatic trap. You simply (1) remove the old cover and unit, (2) insert the new Barnes & Jones Cage Unit, (3) replace the cover, and the old trap will operate with its original efficiency.

Float and Thermostatic Traps



Handle large and sudden condensation loads. Large air and water capacity. Large float assures instant opening of the discharge valve. Cage Type Thermostatic Unit assures quick elimination of air.

Trap No.	T41	T42	T43	T44	T45
Inlet & Outlet Tappings	¾"	1"	1¼"	1½"	2"
Capacity Sq. Ft. @ 2#	800	2000	4800	9600	20000

C. A. Dunham Company

Administrative and General Offices

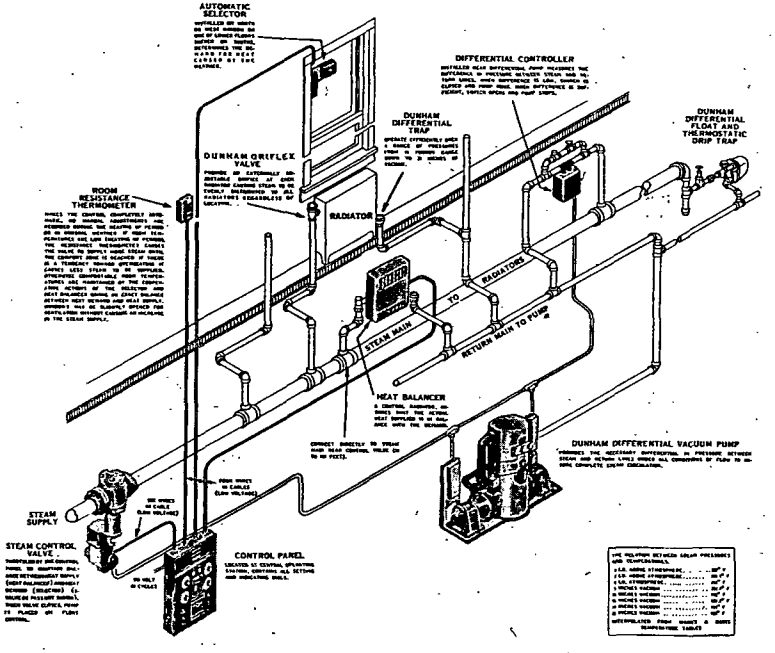
450 E. Ohio Street, **Chicago 11, Ill.**

Factories: MARSHALLTOWN, IOWA; MICHIGAN CITY, IND.; TORONTO, CANADA; LONDON, ENGLAND
TORONTO 4, 1523 DAVENPORT ROAD. **LONDON**, LOMBARD RD., MERTON, S.W. 19

THE DUNHAM DIFFERENTIAL VACUUM HEATING SYSTEM

Fully Automatic

The system is a simple two-pipe system in which all the essentials of circulation, distribution and control are co-ordinated. Control of the temperature of the steam is accomplished by controlling the pressure or vacuum of the steam in the supply piping and radiators to balance exactly the heat input with building-heat-loss.



A. Room Resistance Thermometer—Makes the control completely automatic. No manual adjustments are required during the heating up period or in unusual weather. If room temperatures are low (heating up period) the resistance thermometer causes valve to supply more steam until the comfort zone is reached. If there is a tendency toward overheating it causes less steam to be supplied. Otherwise, comfortable room temperatures are maintained by cooperative actions of the selector and heat balancer giving an exact balance between heat demand and heat supply. Windows may be slightly opened for ventilation without causing an increase in steam supply.

B. Automatic Selector—Installed on north or west window on one of lower floors (never on south), determines the demand for heat caused by the weather.

C. Dunham Orifiex Valve—Provides externally adjustable orifice at each radiator causing steam to be evenly distributed to all radiators regardless of location.

D. Dunham Differential Trap—Operates efficiently over a range of pressures from 15 lb gage down to 25 in. of vacuum.

E. Differential Controller—Installed near differential pump, measures the difference in pressure between steam and return lines. When difference is low, switch is closed and pump runs. When difference is sufficient, switch opens and pump stops.

F. Dunham Differential Vacuum Pump—Provides the necessary differential in pressure between steam and return lines under all conditions of flow to insure complete steam circulation.

G. Heat Balancer—A control radiator insures that the actual heat supplied is in balance with the demand.

H. Connect directly to steam main near control valve (50 to 100 feet).

I. Four wires in cables (low voltage).

J. Six wires in cable (low voltage).

K. Steam Supply.

L. Steam Control Valve—Throttled by the control panel to maintain balance between heat supply (heat balancer) and heat demand (selector) (3-valve by-pass not shown). When valve closes, pump is placed on float control.

M. 110 volt 60 cycles.

N. Control Panel—Located at central operating station contains all setting and indicating dials.

DUNHAM THERMOSTATIC TRAPS

For Operating Pressures up to 15 lb psi gage

The ½ in. trap is available in AP, RH, LH, SW, and vertical patterns. The ¾ in. in AP, SW only. The 1 in. in AP only.

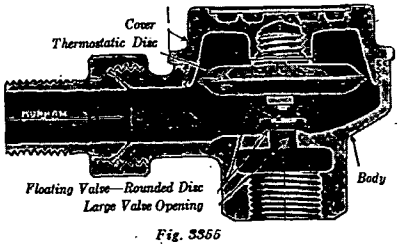


Fig. 3566

Construction—Bodies, covers, nut and nipple are of brass, valves and seat are copper alloy. The trap is non-adjustable, permanent adjustment for correct operation is built into the element. The thermostatic elements for traps of the same size are interchangeable without adjustment. The disc is made from special composition phosphor bronze sheet. The corrugations are shaped to reduce hinge action at the rim of the disc and to distribute disc motion uniformly. The filling nozzle and the valve assembly are attached to their respective halves of the disc by threaded nuts making tight screwed joints which are further reinforced, locked and sealed by sweating with solder. The valve is flat and is attached to the disc by a ball swivel joint to insure its seating squarely and tightly. Valve opening is exceptionally large and the passage of water or dirt out of the trap is not obstructed by a guide as none is necessary with the Dunham design. Valve seat is raised slightly and rounded to minimize the depositing of incrustants.

DUNHAM RADIATOR VALVES

"ORIFLEX" PACKLESS RADIATOR VALVE

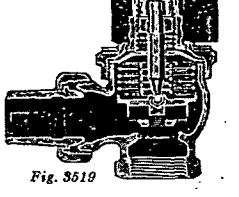


Fig. 3519

Type 175 Oriflex self-contained adjustable orifice valve combines the conveniences of both lever and wheel handle valves. Oriflex has a unique "handle movement" which eliminates graduated opening or closing. It need only be turned "on" or "off." The adjustable orifice within the valve, not the postion of the handle, controls the steam flow. Oriflex eliminates the time consuming operations of disconnecting and re-connecting the valve in order to balance a system. Merely remove the handle, insert the key on the adjustment stem thus made accessible, adjust the orifice (calibrated guide surrounds stem) to the exact setting needed for perfect balance. Instead of being limited to orifices calibrated in steps of five square feet, Oriflex provides for adjustments in increments of one square foot.

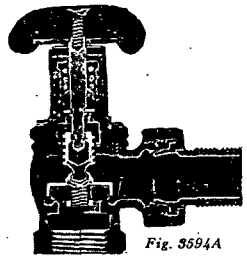


Fig. 3594A

740 SERIES DUNHAM RADIATOR VALVE
(Spring Packed)

Designed for low pressure steam heating service. Bodies are brass castings, rough finish. The valves are equipped with heavily constructed brass union nuts and nipples. All pipe threads and tappings are carefully machined and checked to standard gages. Non-rising stem, requires less than one turn of handle to open the valve fully. Dial shows direction and amount of opening. Heavy bronze spring keeps a constant pressure on a special graphited asbestos composition ring to maintain a tight seal around the valve stem.

PACKLESS RADIATOR VALVE, SERIES 1140
Wheel Handle

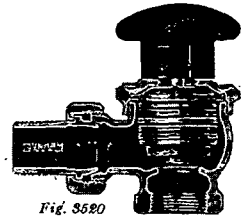


Fig. 3520

Suitable for all types of low pressure steam heating systems. The bellows construction, the non-rising stem, low bonnet, heat-resistant composition handle requiring less than one turn to open the valve fully, recommend this valve for services in which long-wearing quality, absolute tightness and attractive design are desired.

Bodies and bonnets are brass castings, rough finish. The valves are furnished with heavily constructed brass union nuts and nipples. The expansion member is the built-up type of bellows, fabricated from tinned phosphor bronze giving maximum resiliency and wear. The expansion member not only prevents leakage of steam, air and water, but also prevents steam, water and dirt from clogging and corroding the spindle nut and screw.

DUNHAM THERMOSTATIC STEAM TRAPS—TYPE TH
For Working Pressures 25 Lb to 100 Lb

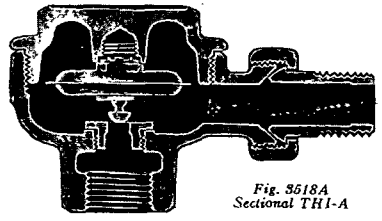
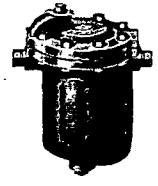


Fig. 3518A
Sectional TH1-A

Construction—Type TH traps are the thermostatic fluid expansion type.

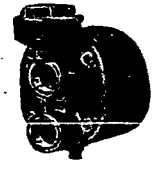
The trap consists of two principle parts, a body with renewable valve seat and a cover containing the expansion thermostat. Permanent adjustment for correct operation is built into the element. The covers may be removed from the trap body while hot without danger to the thermostatic element. The valve is swiveled to insure its seating tightly without causing localized stresses on the element. Body, cover, nut and nipple are of brass; valve and seat are of special heat treated stainless steel; thermostatic element is formed from monel metal sheet.

DUNHAM INVERTED BUCKET TRAPS
Type OB For Operating Pressures up to 250 lb.
Type OBS For Operating Pressures up to 150 lb.



Type OB sizes are ½ in. to 1¼ in. inclusive, Type OBS made only ½ in. and ¾ in. sizes, right hand female tappings. Body and cover are of high test semi-steel castings and are provided with a plugged opening at the lowest point of the body. The valve and seat, which are renewable and interchangeable, are constructed of especially hardened, corrosion resisting steel. Bucket is formed from sheet copper. Cover cap screws are steel.

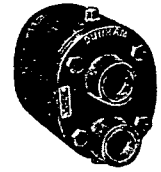
DUNHAM FLOAT AND THERMOSTATIC TRAPS
30 Series—For Operating Pressures up to 15 lb Gage



Trap is comprised of a cover and mechanism assembly and a body. Thermostatic disc and valve controls flow through a cored passage between the trap body and the discharge tapping for the release of air. The float is cuprous material. The float valve and seat are monel metal. Thermostatic elements are interchangeable. The trap body can be readily removed without disturbing piping connections

to fully expose the working parts for inspection. Covers and disc assemblied are interchangeable. Capacities, based on a rate of ¼ lb condensate per sq ft of EDR per hour, at a pressure difference between inlet and outlet of 2 lb gage, 800, 2000, 4800, 9600, 20000. Female right-hand threaded inlet and outlet. Connections ¾ in., 1 in., 1¼ in., 1½ in. and 2 in.

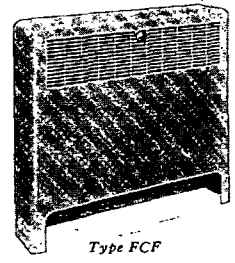
DUNHAM CLOSED FLOAT TRAPS
31 Series—For Operating Pressures up to 15 lb Gage

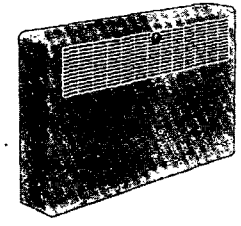


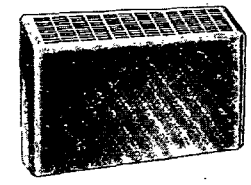
This trap is designed to release water only from low pressure steam installations. It may be used for dripping rise in steam main, and other applications where no air is to be handled. These traps are similar in construction and capacities to the Float and Thermostatic Trap except thermostatic feature is omitted.

DUNHAM CABINET CONVECTORS
With Non-Ferrous Heating Element

Dunham Convectors can be used on steam or vapor, and gravity or forced hot-water heating systems. EDR capacities with 1 lb steam, 65° F entering air range from 12 to 108 sq in. EDR. Made in 4 in., 6 in., 8 in. and 10 in. widths; 20 in., 24 in., 32 in. enclosure heights; casing lengths from 18 in. to 64 in. Casings are constructed of No. 18 gage steel with removable fronts having rounded corners for pleasing appearance. The outlet grille consists of horizontal openings punched in the casing front. Dampers can be furnished when so ordered. Heating element is constructed of non-corrosive materials, copper or aluminum fins on seamless drawn, round copper tubes brazed to bronze headers. The tubes are expanded after assembly to assure intimate contact between fin and tube for permanent heat transfer. Heavy side plates protect the fins from damage. The elements are tested at a hydrostatic pressure of 400 lb per sq in. and are suitable for operation on 150 lb steam or water working pressure. Either end or top and bottom tapped headers (¾ in. I.P.S.) can be supplied.


Type FCF

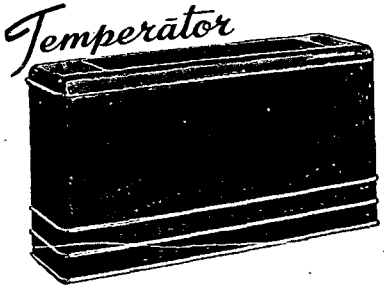

Type WCF


Type WCST

DUNHAM AIR CONDITIONING UNIT

This single unit, when correctly installed, offers complete air conditioning, including warming and humidification for the heating season and cooling and dehumidification for the summer, with filtering, ventilation and circulation of air the year 'round. These combined services are rendered with a minimum requirement of space and a high functional efficiency which guarantees genuine satisfaction and genuine economy.

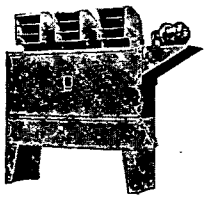
The modernized 1946 model and details of construction, etc., will be available in the Spring of 1946.



DUNHAM TYPE OTS HEATING ELEMENT

An extended surface heating element made entirely of steel. Light in weight, and has unusual heating capacity. Each element is made up of 1¼ in. steel pipe with No. 22 gage heavy fins mechanically attached, eliminating the use of a solder bond without sacrifice of heat transfer. Each fin, when pressed on the pipe, interlocks with the preceding fin—forming an exceedingly tight and permanent mechanical joint. The complete unit is painted with heat-resisting zinc chromate black enamel. Lengths from 18 in. up to 144 in. can be supplied in 6 in. increments. Standard units are threaded at each end with standard pipe threads.

DUNHAM UNIT HEATERS



Type R—Floor Type with mixing damper also furnished with elongated nozzles

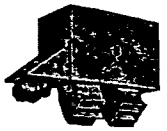
Types C and R are essentially industrial type units designed for heating large spaces. Type C, discharging large volumes of heated air downward to working levels, distributes heat evenly over large areas.

Type R is available in various types of mounting as shown. All units have belt driven centrifugal type fans using constant speed 1750 rpm motors. Complete heating element and complete fan and shaft assembly can be removed through either end of casing. Heating element is replaceable copper tube type.

Type V—Horizontal propeller fan type. Built in various sizes from 65 up to 1500 sq ft EDR.

Type R—Wall Type

Type R—Ceiling Type

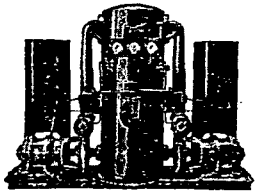


Type R—Inverted Wall Type

Type C—Built in various sizes up to 8000 sq ft EDR.

DUNHAM PUMPS

Tested and Rated with A.S.H.V.E. Code and Code of Vacuum Return Line Heating Pump Manufacturers' Section of Hydraulic Institute.

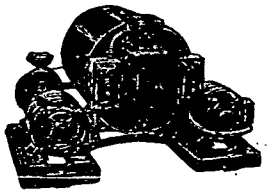


Types DV and VRD

Vacuum Pumps

Types DV and DVD—Capable of maintaining whole systems under vacuums as high as 25 in. Built in 9 sizes. Capacities 2500 to 65,000 sq ft EDR.

Types VR and VRD—Meets all code tests for air and simultaneous air and water handling capacities. No moving parts or close clearances in exhauster unit. Built in 9 sizes. Capacities 2500 to 65,000 sq ft EDR.



Condensation Pumps

Pump and motor assembled on rigid cast iron base. Bronze fitted centrifugal pump has non-corrosive shaft. Enclosed type Impeller. Liberal size ball bearings.

Type CH-Model B, Single and Duplex—Capacities 2000 to 50,000 sq ft EDR; 60 cycle d.c. or a.c. 1750 rpm; 25 or 50 cycle a.c., 1450 rpm.

Type CHH-Model B, Single and Duplex—Capacities same as type CH. Discharge pressures 20 lb for pumps 15,000 sq ft and larger; 20, 30, 40, 50 and 70 lb discharge in all sizes. 60 cycle d.c. or a.c. 3450 rpm; 25 or 50 cycle a.c., 2850 rpm.

GRINNELL COMPANY INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence 1, R. I.**

National Distributors of Thermoflex Traps and Heating Specialties

For data on other Grinnell Products, see pages 981, 1112, 1113

Thermoflex Specialties

The heart of all Thermoflex Traps is the Hydron Bellows.

The Hydron Bellows is formed under hydraulic pressure. This powerful internal pressure locates any weakness of any nature in the tubing. Such hydraulic pressure is many times more severe than any pressure the Trap will ever be called upon to control. Every Thermoflex Trap, therefore, is practically indestructible.

Thermoflex Traps have an exceptionally large orifice. This large orifice combined with high lift, insures fast action and freedom from clogging.

We supply Thermoflex Traps guaranteed for steam pressures of 25 lb, to 50 lb and to 125 lb. Complete information and details of typical installations will be gladly sent on your request.

Thermoflex Low Pressure Line

The Thermoflex line of low pressure specialties offers a wide selection of sizes, types and capacities designed for working steam pressures up to 25 lbs.

All of the specialty items which are regular components of Vacuum, Vapor, and Vented Return heating systems, are available in postwar models, including

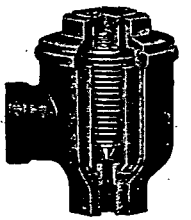
Thermostatic Traps
Modulating Packless Radiator Valves
Float and Thermostatic Traps
Vapor system Return Traps and Vents

Construction and finish are of the highest quality throughout.

Thermoflex Medium Pressure Traps

Thermostatic type traps, and Combination Float and Thermostatic type traps are furnished for working steam pressures in the range from 25 to 50 lbs.

Thermoflex High Pressure Traps



The No. 100A Thermoflex Trap is guaranteed for steam pressures from 50-125 lb. Must not be used where the steam temperature exceeds 400 F.

For use with all types of process work, Laundry Machinery, Kitchen Equipment, Hospital Sterilizers, Vulcanizers, Dry Kilns, Unit Heaters, Street Steam Service, etc., in fact any place that a trap is desired for service at the above pressures.

Small, compact and inexpensive.

Extra heavy body. Renewable nickel steel seat and disc. Bellows made from special bronze tubing and encased in brass sleeve to prevent distortion due to pressure.

Regularly furnished without unions.

Thermoflex Streamlined Strainers

Pipe line strainers of the self-cleaning Y-type are furnished for pressures up to 250 lbs, and in sizes ⅜ in. to 2 in. These are heavy duty strainers with semi-steel body and brass screen, which are suited to a wide field of use in removing harmful substances from pipe lines carrying steam, air and fluids.

William S. Haines & Company

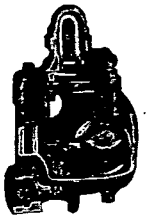
12th and Buttonwood Sts., **Philadelphia 23, Pa.**

Manufacturers of

EQUIPMENT FOR VAPOR AND VACUUM HEATING SYSTEMS

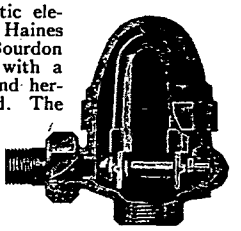
PRODUCTS—Haines Vento Radiator Traps, Medium Pressure and Blast Type Traps, Combined Float and Thermostatic Traps, Air Eliminators, High Pressure Thermostatic Traps, Boiler Return Traps, Radiator Valves.

HAINES F & T TRAPS



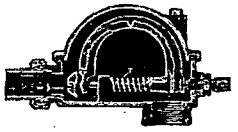
Designed to handle large quantities of condensation. For dripping steam mains, unit heaters, hot water generators, etc. Cannot become air bound as it has a thermostatically controlled air by pass. Sizes ¾ in., 1 in.. 1¼ in. For pressures to 30 pounds.

HAINES RADIATOR TRAPS



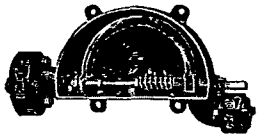
The thermostatic element in all Haines Traps is a Bourdon tube, charged with a volatile fluid and hermetically sealed. The expansion and contraction of the fluid, under varying temperatures, furnishes the operating power. The vertical seat of this trap prevents it from becoming inoperative from scale or other foreign matter.

HAINES MEDIUM PRESSURE TRAPS



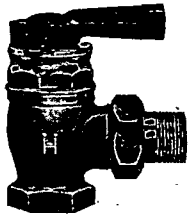
A ruggedly constructed bolted case trap. Ideal for hospital and kitchen equipment and all process work operating on pressure up to 100 pounds.

HAINES HIGH PRESSURE TRAPS



For dripping high pressure mains, laundry equipment and all process fixtures with working pressures up to 125 pounds.

HAINES MODULATING VALVES



A packless valve assuring positive and leak proof performance. Completely opens or closes on less than a full turn of handle. Can be furnished with wheel or lever handle or lockshield.

HAINES BOILER RETURN TRAPS

For vapor and atmospheric heating systems. Assures positive circulation by venting the air and returning the water of condensation to the boiler. Has no stuffing boxes or packed joints to leak air or water.

Each device is individually tested, factory adjusted and guaranteed.

Hoffman Specialty Co., Inc.

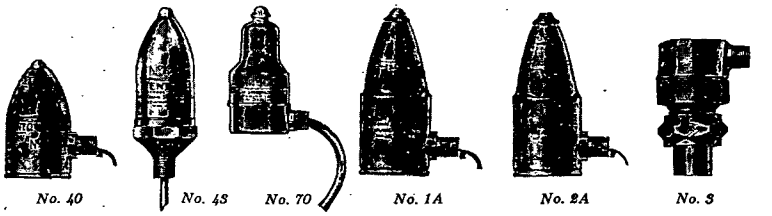
General Office and Factory

1001 York Street, **Indianapolis 7, Ind.**

Sales Representatives in Principal Cities

Manufacturers of Radiator Air Valves, Quick Vents and Air Eliminators for all types of Steam and Vacuum Heating Systems—Steam Traps of all kinds—Radiator Supply Valves—Vacuum and Condensation Pumps—and Hot Water Automatic Heat Control Systems.

RADIATOR AIR VALVES FOR STEAM AND VACUUM SYSTEMS

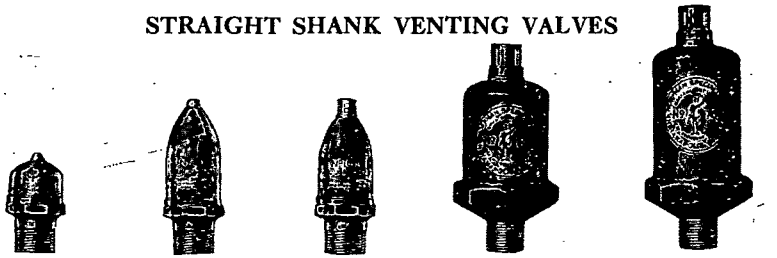


No. 40 *No. 43* *No. 70* *No. 1A* *No. 2A* *No. 3*

No. 40 Steam—Hoffman patented tongue syphon—⅛ in. connection—fixed port.
No. 41 and 43 Steam—Strait shank for convectors—telescopic syphon—⅛ and ¼ in. connections.
No. 70 Steam—Meets Federal spec. WW-V-151 Class 1—Long syphon—⅛ in. conn.
No. 1A Steam—Tongue syphon—ADJUSTABLE air opening—⅛ in. connection.
No. 2A VACUUM—Tongue syphon—ADJUSTABLE air opening—⅛ in. connection.
No. 3 Steam—For Airline or PAUL systems—⅛ x ¼ in. conn.—union tailpiece.

All radiator air valves operate on 10 lb max. press.

STRAIGHT SHANK VENTING VALVES



No. 4 *No. 4A* *No. 16A* *No. 75* *No. 76*

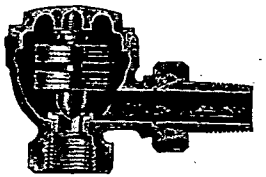
No. 4 Steam Mains—Will not close against water—25 lb press. ¾ in. connection.
No. 4A Steam Mains—Float closes against water—10 lb press. ¾ in. connection.
No. 16A VACUUM Mains—Float closes against water—10 lb press. ¾ in. connection.
No. 75 Steam Mains—Large systems—has float—10 lb press. ¾ in. connection.
No. 76 VACUUM Mains—Large systems—has float—10 lb press. ¾ in. connection.
No. 75A Steam Mains—Large systems at low pressure—has float—3 lb press. ¾ in. conn.
No. 76A VACUUM Mains—Large systems low pressure—has float—3 lb press. ¾ in. conn.

Nos. 75-75A-76-76A have cast iron bodies—others brass.

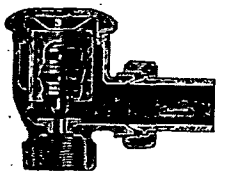
UNIT HEATER VENT VALVE

No. 74—Operates 0 to 35 lb. Vents all air at any pressure and whether rising or falling—Same appearance as No. 75—Can be used on steam mains—has cast iron body—¾ in. connection—finished in attractive Hoffman gray-green color.

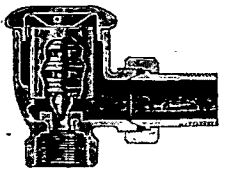
LOW, MEDIUM AND HIGH PRESSURE THERMOSTATIC TRAPS



Low Pressure



Medium Pressure



High Pressure

Low pressure traps have brass bodies, caps, and union nut and tail piece. 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.

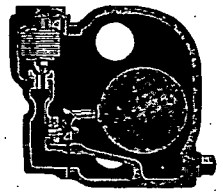
No. 17C Capacity 200 sq ft EDR 15 lb pressure ½ in. connection
No. 8C Capacity 400 sq ft EDR 25 lb pressure ¾ in. connection
No. 9C Capacity 700 sq ft EDR 25 lb pressure 1 in. connection

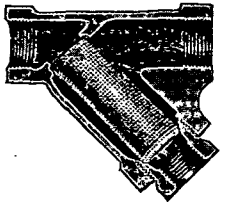
Medium Pressure Nos. 8 & 9 and High Pressure Nos. 8H & 9H have all bronze bodies and caps with union nut and tailpiece. Thermostats are 6 diaphragms of special non-corrosive metal. Thermostats and seats are renewable. ½ in. sizes are furnished in Angle, R.H., L.H., and Straightway patterns, others in Angle only. Medium Press. 50 lb limit. High Press. 125 lb.

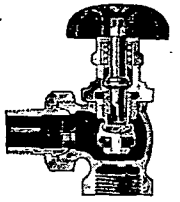
Capacities—Lb Condensate per Hour—Working Pressure—Lb per Sq. In. Gage

Traps		5	15	25	50	Traps		25	50	100	125
8	⅛"	100	180	235	400	8H	⅛"	235	400	550	590
8	½"	125	225	300	490	8H	½"	300	490	650	720
9	¾"	225	350	450	650	9H	¾"	450	650	875	950
9	1"	325	500	625	850	9H	1"	625	850	1125	1250

FLOAT TRAPS, DIRT STRAINERS AND SUPPLY VALVES





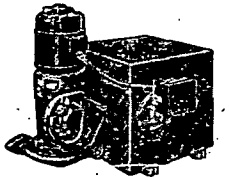


Float Traps are available in large capacities and four pressures, 15, 30, 60 and 125 lb. Used for venting and draining risers, steam mains, unit heaters, blast coils, etc. Hoffman Float Traps are made for easy servicing with all working parts mounted on cover. Remove four bolts to expose all parts. Pipe sizes are from ¾ in. to 2 in.

Radiator Supply Valves are made in sizes from ½ to 1½ in. in Angle and straightway patterns. Brass bodies, union nut and tailpiece. No. 80 has reversible cone disc and bevelled seat. No. 85 is Modulating type. Both are packless.

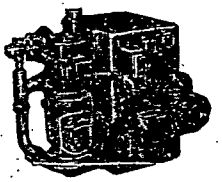
Hoffman Dirt Strainers are self-cleaning Y type. Brass strainer cylinders and cast iron body. Sizes ½ to 2 in. for 125 lb pressure. Should be used in line ahead of all float and thermostatic traps.

CONDENSATION AND VACUUM PUMPS



Condensor Pump

Hoffman-Economy pumps are available in varying capacities, D.C. and A.C. current, single, two, or three phase, and in pressures up to 200 lb. Also made in single and duplex units for different capacities and pressures.

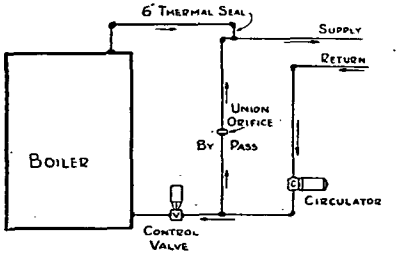


Vacuum Pump

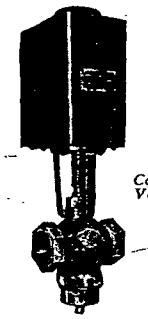
HOFFMAN HOT WATER CONTROLLED HEATING

For PANEL or RADIATOR SYSTEM

The Hoffman method vastly improves the ordinary forced hot water system by the application of Continuous Circulation. This method accomplishes 4 major improvements. 1. Avoids intermittent bursts of heat to the radiators and panels. 2. Maintains radiator and panel temperature to exactly offset the heat loss thereby eliminating "cold 70." 3. Conserves fuel by positively preventing overheating. 4. Prevents creeping noises from pipes because temperature change is so gradual.

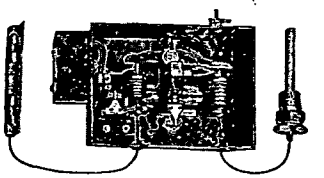


The diagram illustrates the general operation. The circulator (C) is running continuously, circulating the water through the by-pass from return back through the supply line. The water in boiler is kept at a predetermined temperature based on the Radiator Emission rate for which the system is designed. When heat is required the Temperature Controller or Thermostat (depending upon type of Hoffman system used) causes the control valve (V) to open, allowing the return water to pass through the boiler and the hotter boiler water mixes with water passing through the Hoffman orifice, in the by-pass line from the return, to raise the temperature of water entering the supply line. When the heat demand has been satisfied the control valve (V) closes and normal circulation through the by-pass pipe continues. The mixing of the water, already somewhat cooled from continuous circulation, with the high temperature boiler water eliminates the "slugs" of extremely hot water so common with forced hot water systems. There is positively no overheating and the temperature is raised so gradually that a comfortable heat is always available with apparently no change in the heating system.



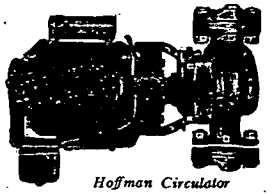
Control Valve

The two types of Hoffman Hot Water Controlled Heat differ only in the manner of control. The Circulator, Control Valve and Union Orifice are furnished with both systems.

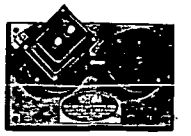


1. Outdoor Thermal Control System—This System is controlled by means of the Hoffman Patented Temperature Controller illustrated above. Capillary tubing is connected to a temperature bulb located on the exterior of the building and another capillary tubing connected to a temperature bulb located in the supply line of the System. Variations in temperatures of the outdoor weather and the circulating water are immediately transmitted through this Controller to the Control Valve. This System very completely controls the supplying of the exact amount of heat necessary to offset the actual heat loss. Made in nine regular pipe sizes from 1 in. to 6 in. with a maximum capacity of 4,460,000 Btu.

2. Indoor Thermostat Control System—Thermostat, located in that part of the building which is to determine the heat requirements for the entire System, operates the Control Valve through the Control Panel illustrated below. This allows the circulating water temperature to be increased according to the demand of the thermostat and will supply the heat required uniformly and gradually. This System is naturally less expensive than the Outdoor Control type. Available in 1 in., 1¼ in. and 1½ in. sizes with a maximum capacity of 275,000 Btu.



Hoffman Circulator



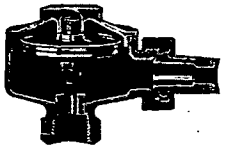
Design and Operating information is available upon request.

ILLINOIS ENGINEERING COMPANY

General Offices and Factory:
Chicago 8, Ill.

Representatives In Principal Cities

Illinois Thermo Radiator Traps



Series G

Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems. Has cone type valve. Flushes throughly and seats perfectly at all times. Valve and seat are of Nitralloy. The duplex diaphragm is of special phosphor bronze. Scientific design and rugged construction assure flexibility and long life. These diaphragms have withstood over three million strokes on a breakdown test. *Ask for Bulletin.*

Vapor Gauge

Vapair Vent Trap

Boiler Return Trap

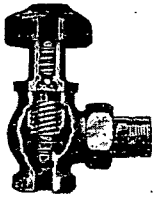
Illinois Selective Pressure Control Systems

Selective Controller

An entirely new and unique method of Steam Circulation Control . . . Heating Systems that set new standards in comfort, economy, simplicity and convenience of operation. Each system individually engineered to meet exact requirements. Recorded fuel savings, without sacrifice of comfort, warrant your investigation. *Ask for Bulletin 16.*

Illinois Radiator Supply Valve

Quick-opening, packless. Steam tight on 50 lb pressure. Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns.



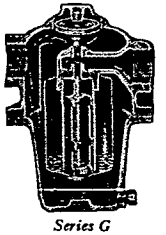
Illinois Vapor System

A two pipe low pressure steam circulating system which may be installed in any type of building, where the condensate can return to the boiler by gravity.

A sensitive damper regulator or other means of automatic control is used to control initial steam pressure above, at or below atmospheric pressure. Steam is regulated at the radiators by Illinois Modulating Supply Valves. Condensate and air are discharged from the radiator through Illinois Thermostatic Radiator Traps. In the boiler room a Vapair Vent Trap and Boiler Return Trap are installed near the boiler. The vent trap eliminates air from the system and the Return Trap insures return of condensate to the boiler.

The system and the piping arrangement are simple. No metering orifices or vacuum pumps are needed. This system will be found suitable for many installations where low first cost and low operating cost are of prime importance. May be used with unit heaters or any type of radiation.

Illinois Combination F & T Traps



Series G

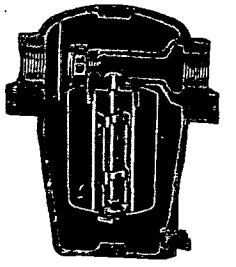
Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers—wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold.

ILLINOIS ENGINEERING COMPANY

General Offices and Factory:
Chicago 8, Ill.

Representatives In Principal Cities

Illinois Steam Trap



Series 30

Valve and stem are separate from the bucket and operated only by the bucket at the extreme top and bottom of travel—result—*valve is always either full open or tight closed.* No wire drawing or cutting of valve and seat which are of stainless steel.

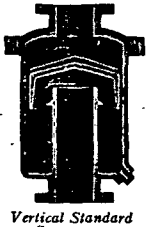
Illinois Thermostatic Traps for High Pressures

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. Also used extensively for air vents on blast type drying heaters. Multi-diaphragm of phosphor bronze. Heavy duty bronze body. Made in three sizes.

These traps are also furnished for medium pressures. *Write for literature.*

Steam and Oil Separators



Vertical Standard Separators

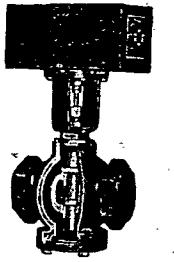
Eclipse steam separators are made in both horizontal and vertical type, and also the special receiver separators for standard or extra heavy pressures.

Eclipse oil separators are furnished in the horizontal type and have a removable baffle plate to facilitate cleaning of baffle and keeping the separator's efficiency at the highest point.

Illinois Motorized Valves (on and off)

For automatic control of steam temperatures and pressures to prevent overheating and conserve steam; to control fluid levels; and to regulate flow in hot water heating systems. May be operated by any automatic contact device or by manual switches.

Furnished in three types.



Type E3

Spring Controlled Regulating Valve

Furnished in either single seat or double seat type as service requires, for the control of steam, air or gas. Control spring is completely enclosed, protecting it from dirt and rust. Valves are furnished with proper size diaphragm and proper length spring to give satisfactory service under all operating conditions. Furnished also in weight loaded type, Fig. 71.

Write for Bulletins.

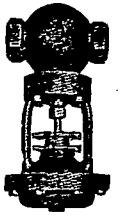


Fig. 121

Non-Return Valves

Placed between boiler and header to prevent return of steam to boiler. Sensitive in operation. Extra heavy iron bodies with bronze trim for 250 lbs steam working pressure. Bronze dash pot and water sealed pistons prevent valve sticking. Globe and angle patterns from 4 in. to 12 in.

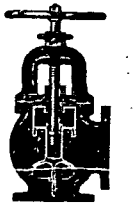


Fig. 260

Sarco Company, Inc.

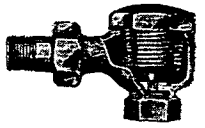
475 Fifth Ave., New York 17, N. Y.

Branches in Principal Cities

SARCO CANADA LIMITED, 85 RICHMOND ST., W., TORONTO, ONT.

PRODUCTS—A complete line of Specialities for Vapor, Vacuum and Gravity Steam Heating Systems and Control combined with a competent Engineering Service to architects and heating engineers to assist them in providing modern heating.

SARCO RADIATOR TRAPS



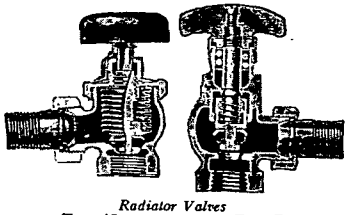
Radiator Trap, Type H

Type H is the standard radiator trap for vapor and vacuum systems. It is equipped with the well known Sarco heavy wall bellows, drawn from flat blanks and helically corrugated in our own plant. It operates noiselessly and positively at pressures from highest vacuum to 25 psi.

Body and cap are of cast brass, rough brass finish; self-aligning valve head and renewable seat of hard bronze; union connection on inlet.

Available in ½ in. and ¾ in. sizes, angle, straightway or corner offset patterns; also 1 in., angle style only. *Catalog No. HV-100.*

SARCO RADIATOR VALVES



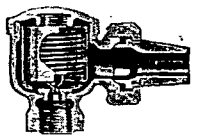
Radiator Valves
Type 45 *Type R*

Sarco offers two types of radiator valves. Where highest efficiency is desired, the bellows-packless valve, type 45, is recommended. The valve stem is sealed by a standard Sarco bellows. Leakage of air into the system under vacuum is impossible.

Modulating types with proportioning disc and indicating dial are available.

Type R is of the "spring-packless" type, quick opening, non-rising stem. The molded stem packing is spring controlled.

Bodies of all valves are cast brass, rough brass finish. Outlet fitted with union connection. Sizes ½ in. to 1½ in., with wheel handle or lock shield. *Catalog No. HV-100.*

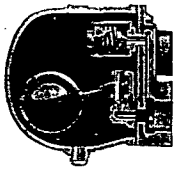


N-100 Medium Pressure Trap

SARCO N-100 TRAP

For high pressure radiators and heating coils in stationary and marine service, and for hospital and kitchen equipment. Has full length protecting shield and stainless steel valve head and seat. Sizes ⅜ in. to 1 in. pressures to 100 lb. *Catalog HV-190.*

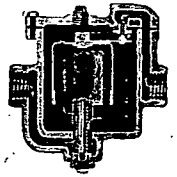
Also S-65 for pressures to 65 psi.



Float-Thermostatic Trap

SARCO FLOAT-THERMOSTATIC TRAPS

For dripping ends of mains and risers, and for stack or blast heaters, large unit heaters and hot water generators. Automatic thermostatic air vents built in. Available in six sizes with connections ¾ in. to 2 in. Pressures up to 200 psi. *Catalog HV-450.*

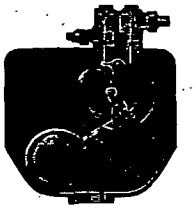


Inverted Bucket Trap

SARCO INVERTED BUCKET TRAPS

Are recommended for high pressure unit heaters and sometimes preferred for kitchen and laundry equipment. Strainers are built right into these sturdy traps. Seats and valves are stainless steel and renewable. Automatic air vents can be furnished for extra rapid removal of air. Available in sizes ½ in. to 2 in. for pressures up to 900 psi. *Catalog HV-350.*

SARCO ALTERNATING RECEIVER



Alternating Receiver

A complete line of boiler return traps for vapor systems.

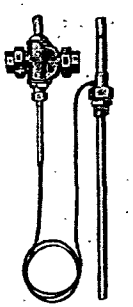
Returns water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions.

Made in four sizes for up to 14,000 sq ft of radiation. *Catalog HV-165.*

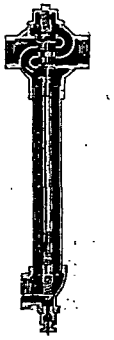
SARCO AIR ELIMINATORS

For venting air from vapor systems at one central point in the basement. Available in three sizes, No. 5 for systems up to 2000 sq ft radiation, No. 6 for 3000 sq ft, and No. 12A for 15,000 sq ft. All are equipped with float valves to stop water escaping thru the vent and with check valves to prevent ingress of air when system is under vacuum.

Also several types for hot water heating systems. *Catalog HV-170.*



Type TR-21-Standard for hot water storage tanks, fan units, etc.



Water Blender Type MB

SARCO SELF-CONTAINED TEMPERATURE REGULATORS

Sarco Temperature Regulators are simple, self-operated valves—the only self-contained units that use the irresistible force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required; all moving parts are *inside* the equipment. Here again—a type and size for every purpose—for steam, gas, oil, water or brine for temperatures ranging from 0 to 300° F. *Catalog HV-600.*

SARCO WATER BLENDERS AND TEMPERING VALVES

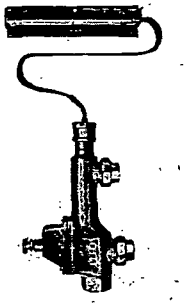
For mixing hot and cold water to deliver automatically water at any desired temperature. Two models are available, type MB for showers, wash basins, etc., and type DB, a tempering valve for use with submerged heating coils or tankless heaters. *Catalog HV-800.*

Water Blender Type DB

SARCOTHERM HOT WATER HEATING SYSTEM

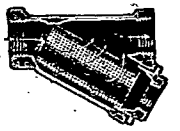
A simple, all-mechanical system for the control of radiator temperatures in direct relation to outside temperatures. Radiation is balanced by Sarcoflow fittings in the radiator outlets.

The Sarcotherm three-way valve recirculates a varying proportion of water around the boiler and back to the system as dictated by the thermostatic bulb outside the building. Write to Sarcotherm Controls, Inc., 280 Madison Ave., New York 16, N.Y. for *Catalog No. HV-1.*



Sarcotherm Weather Control Valve

SELF-CLEANING STRAINERS



For use in pipe lines carrying brine, steam, oil, gas, water, ammonia or air. Have large free screening area with minimum resistance to flow. Steam or air strainers can be cleaned by blowing through without disassembling. Made in cast iron, bronze or cast steel for pressures up to 500 psi, with brass, iron or monel screens. Available in sizes ¼ to 8 in. *Catalog No. HV-1200.*

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

Webster -since 1888 Systems of Steam Heating

Main Office and Factory:
Camden, New Jersey

Representatives in Principal Cities—
Consult Your Local Phone Directory

Webster Nesbitt UNIT HEATERS

PRODUCTS AND SERVICES

Webster Systems of Steam Heating including Vacuum and Type "R" (vapor).

Webster MODERATOR Central Control Systems.

Webster Baseboard Heating.

Modernization of Obsolete and Faulty Heating Systems.

Webster System Equipment including Light-Weight Concealed Radiation (Gravity Convection Heaters), Radiator Supply Valves, Metering Orifices, Thermostatic Traps, Drip Traps, Heavy Duty Traps, Dirt Strainers, Dirt Pockets, Boiler Return Traps, Vent Traps, Damper Regulators, Boiler Protectors, Lift Fittings, Expansion Joints, Separating Tanks, Steam and Oil Separators, Steam Vacuum Pump Governors, Air Separating and Receiving Tanks, Gages, Water Accumulators.

Webster Series "78" and Series "79" Traps for use at process pressures (10 to 150 lb per sq in.)

Webster-Nesbitt Unit Heaters.

WEBSTER SYSTEMS

Webster Systems are low pressure, two-pipe systems of steam circulation *with the addition* of accurately-sized metering orifices at radiator supply connections and, when required, intermediate metering orifices at points in branch mains. Metering orifices effect even distribution of steam to all parts of the heating system and permit the successful application of a centralized control. Webster Valves are used at supply of radiators. Webster Thermostatic Traps prevent flow of steam into return mains when radiators are filled. Webster Drip Traps and Dirt Strainers are used where needed on steam mains. Webster Systems are available for vacuum, open return or "vapor" operation. The Type "R" System corresponds to the so-called Vapor type. Fig. 1 illustrates a typical arrangement of Boiler Return Trap, Vent Trap, etc., when low pressure boiler is the source of steam.

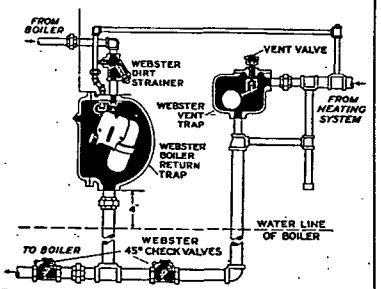


Fig. 1. Conventional arrangement of piping around Webster Basement Equipment for the Webster Type "R" System

WEBSTER MODERATOR CONTROLS

These are patented systems for varying the amount of steam to all radiators according to outside temperature. They provide *continuous heat delivery* with effective *fractional filling of radiators*. The Moderator Systems employ an automatic Outdoor Thermostat supplemented by a manual Variator.

The latter is used for quick heating-up, night load, and unusual weather or occupancy conditions. Use of Webster Moderator Control Systems results in (1) increased comfort because over-heating and underheating are minimized and (2) lower fuel or steam costs.

WEBSTER SYSTEM RADIATION

Concealed, non-ferrous type for use with steam and hot water heating systems. Is unique in that it combines *in* a *single unit*, a light-weight heating element of high efficiency with an orificed radiator supply valve, a radiator trap and supply and return piping connections. Also available in conventional models without valve and trap. Metal enclosures for installation within the wall and exposed metal cabinets are available. Webster System Radiation and enclosures are so designed that the entire heating element can be quickly removed without damage to plaster or paint. Space requirements reduced to a minimum and installation greatly simplified.

WEBSTER BASEBOARD HEATING

Combines a new and unique method of heat distribution with the well-known advantages of hot water heating. The convection type heating element is placed at the floor line behind metal baseboards and along outside walls of the room. It is installed in continuous loops, one on each floor of the home. The heating medium is forced hot water using conventional boiler, firing devices, and controls.

RADIATOR VALVES

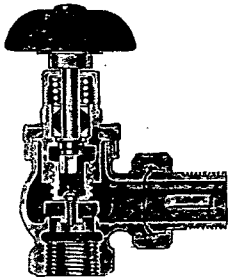


Fig. 2. Webster Type "WB-P" Brass Radiator Valve

The Type WB-P Valve meets fully specifications calling for a "spring packless" valve. A heavy spring automatically maintains pressure on die-molded metallic ring packing. Although packing seldom requires renewing, this valve is so designed that old packing ring can be removed and new installed while pressure is on the heating system.

The Type WB-P Valve opens quickly and easily in less than a turn of the handle. Has non-rising stem. All brass construction including union nut, nipple, and interior parts. Furnished with standard wheel handle as shown in Fig. 2. Lockshield type available for institutions. Angle bodies in sizes from ½ in. to 1½ in. Offset and straightway models can be had on order.

While primarily for low pressure steam heating service, the Type WB-P Valves are suitable for low pressure hot water heating. Furnished with or without leak hole as desired.

Pressures—For low pressure vapor and vacuum steam heating service.

Metering Orifices—Accurately sized and made of metal to resist erosion and corrosion, amply thick to be free from vibration and shaped for quiet operation.

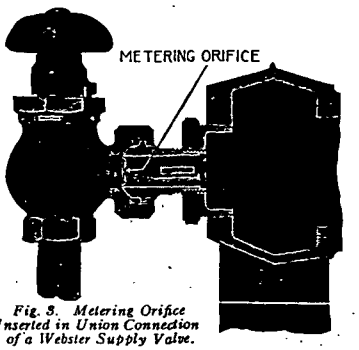


Fig. 3. Metering Orifice Inserted in Union Connection of a Webster Supply Valve.

RADIATOR TRAPS

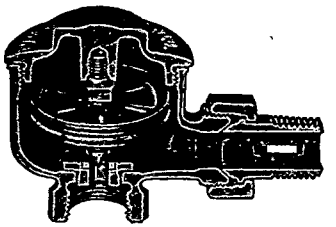


Fig. 4. Webster 702H Thermostatic Radiator Trap

The Webster Series 7 Trap is a diaphragm-type thermostatic trap designed for low pressure service. It is recommended for all types of radiators, piping drip points, steam-using equipment, and other applications within its pressure and capacity range. The Series 7 Trap is noted for its high efficiency and for giving many years of trouble-free service.

Construction Features—Body and cap are high quality brass. Double thermostatic diaphragm is phosphor bronze, individually factory adjusted and tested. Diaphragms are compensated for pressure which means that they function efficiently at *all* pressures within their operating range. They do not close too quickly at certain pressures to hold up condensate while remaining open at other pressures to pass steam.

Stainless steel valve piece is 60 deg cone type, factory adjusted. Brass seat with stainless steel insert is readily renewable.

Pressures—Webster Series 7 Traps are designed for low pressure vapor and vacuum steam heating service. Maximum pressure is 25 lb per sq in.

Table 1. Recommended Ratings in Sq Ft E. D. R.*

The ratings below are conservative and not full-flooded capacities. Applications requiring use of higher ratings should be referred to the Company or its Representatives. When writing give full details of proposed use. Select trap by rating, not by pipe size.

Symbol	Size	Pressure Difference Across Trap in Lb per Sq In.					
		1	1½	2	5	10	15
702	1"	165	200	235	370	530	640
713	¾"	330	400	465	730	1050	1300
724	1"	580	700	810	1280	1840	2300

*Based on 240 Btu per sq ft per hour.

FLOAT-AND-THERMOSTATIC TRAPS

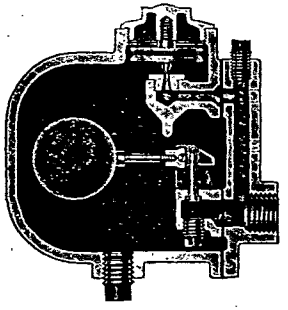


Fig. 5. Webster Size 00026-T Drip Trap has rated capacity of 200 lb Water per Hour at 2 lb Pressure Difference

Series "26"—A heavy duty trap for drips of mains, blast radiation, unit heaters, hot water generators and similar applications. A rugged float-type trap available with and without thermostatic air vent. Made in five sizes: 200, 500, 1200, 2400 and 5000 lb water per hour at 2 lb pressure difference. Maximum working pressure is 15 lb per sq in.

PROCESS STEAM TRAPS

Series "78"—thermostatic trap *built for* process steam pressures (10 to 150 lb per sq in.). Monel Metal diaphragm. Stainless Steel valve

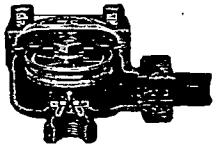


Fig. 6. Webster Size 782 Trap

piece and seat insert. Angle model only. Sizes: ⅜, ½, ¾ and 1 in. Extensively used with laundry, cooking, sterilizing and other process-steam uses.

Series "79"—For use where large volumes of very hot condensate form more quickly than can be discharged by thermostatic traps alone. Float and thermostatic traps designed for normal working pressures between 15 and 150 lb per sq in. Water of condensation is passed through a float-controlled seat opening while air is discharged into the return piping by a thermostatically controlled vent. Compact and light in weight. Can be readily mounted in a pipe line without other support. Available with 1 in. inlet and outlet. Readily bushed for smaller pipe sizes.

Cast iron body, composition gasket and cover bolted together with steel cap screws. Monel Metal valve piece and stem. Stainless steel seat. Air vent unit is Monel Metal diaphragm with Stainless Steel valve piece and brass seat with Stainless Steel insert.

DIRT STRAINERS AND POCKETS

Placed in return lines of steam heating systems to prevent dirt, rust and scale from impairing tightness of traps.

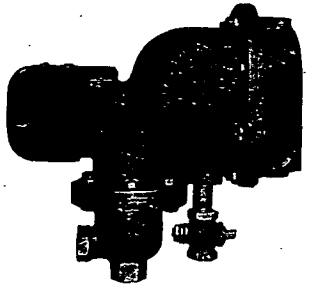


Fig. 7. Size 34C-1 Webster Boiler Protector with Low Water Electrical Cut-out Switch. Size 34 has no Cut-out Switch

BOILER PROTECTOR

Prevents breakage in low pressure heating boilers when water level becomes inadequate. Automatically supplies raw water to boiler when water level drops to 1 in. above bottom of gage glass.

For maximum boiler pressure of 15 lb per sq in. Maximum cold water main pressure should not exceed 150 lb per sq in.; minimum must not be less than 25 lb per sq in.

Made with ¾ in. connections, with or without electrical cut-out switch.

WEBSTER-NESBITT UNIT HEATERS

Are manufactured by John J. Nesbitt, Inc., Holmesburg, Philadelphia 36, Pa., and are distributed solely through Warren Webster & Company, Camden, New Jersey. Designed to circulate large volumes of air at comparatively low temperatures, assuring quick heating.

Ratings of Webster-Nesbitt Unit Heaters are based on tests made in accordance with standard test code of Industrial Unit Heater Association and A.S.H.V.E.

PROPELLER FAN UNIT HEATERS

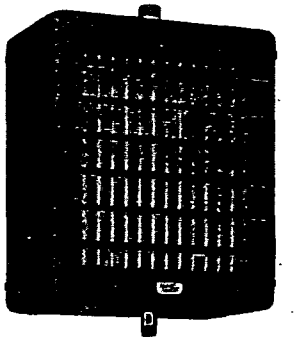


Fig. 11. Standard Propeller-Fan Type

Designed to incorporate four characteristics proved by wide experience to be essential to both proper application and satisfactory performance:

1.) *Selective range of sizes.* Manufactured in nine sizes. Heating capacities vary from 34,700 to 338,000 Btu per hour. Air deliveries from 470 to 4800 cfm.

2.) *Quiet Operation.* All fans have blades of exceptionally large areas and of a shape to impart a gradual acceleration to the air stream. Ample spacing is maintained between the fan and heating element. Motors are of sleeve bearing type and equipped with isolators.

3.) *Durable lightweight Heating Elements.* Extended fin-and-tube type, constructed of copper condensing tubes and plate-type aluminum fins.

4.) *Modern Casing Design.* Compact suspended type. Catalog W-N 126.

GIANT UNIT HEATERS

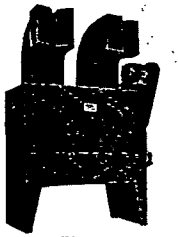


Fig. 12 Blower-Fan Type

Sturdy blower-fan units for the economical heating of large areas.

Standard (Non-Thermadjust) Type. Used principally where heating is by recirculation only, and where constant heat output is desired during period of operation.

Thermadjust Type. Employs dampers in front of casing and over face of heating element to provide mixing of unheated and heated air, producing heat output in accordance with requirements and continuous circulation of air volume.

Valve Controlled Type. Unit is of standard casing arrangement but equipped with Nesbitt Heating Surface with Steam-distributing Tubes which allows for automatic control of heat output.

Floor mounted, wall mounted, ceiling suspended, from 101,000 Btu, 3250 cfm, to 930,000 Btu, 15,200 cfm. Write for details. Catalog W-N 128.

LITTLE GIANT UNIT HEATERS

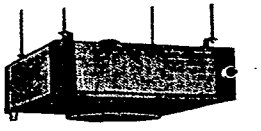


Fig. 13. "Little Giant" Down Blow Type

New, light compact draw-through high velocity down blow unit heaters. 122,000 Btu, 2370 cfm to 505,000 Btu, 10,080 cfm at basic rating of 2 lb steam and 60 deg entering air.

Down Blow Type. Generally used when the presence of cranes and other machinery requires that the unit and piping be located well above the floor level. Write for details. Catalog W-N 127.

Armstrong Machine Works

851 Maple Street, **Three Rivers, Mich.**

Representatives in All Principal Cities

Armstrong offers two types of traps for heating, air conditioning, and steam distribution service.

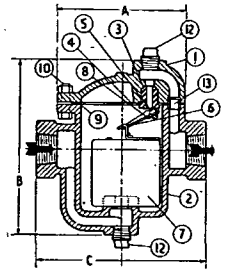
Standard Inverted Bucket Traps, the type originated by Armstrong, are *non-airbinding* and *self-scrubbing.* They are used for low, medium, and high pressure service where relatively little air must be handled along with the condensate. Their free-floating lever design makes it possible to open very large discharge orifices compared with the size of the trap itself.

Armstrong Blast Traps are used where large amounts of air must be vented quickly when steam is first turned on. They have several advantages over the conventional float and thermostatic trap.

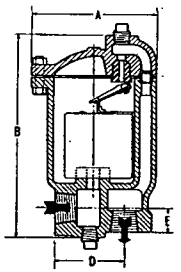
1. The Armstrong Blast Trap has but a *single orifice* to be maintained tight against the full pressure differential.

2. *Positive action.* The discharge valve in an Armstrong Blast Trap is either wide open or tight shut. Fast opening and fast closing prevent wire-drawing.

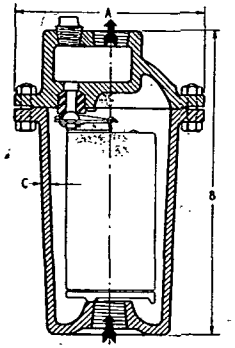
3. *Handles dirt.* There are no dead spots in an Armstrong Trap in which dirt can settle and interfere with the operation of the trap.



Cross-section of No. 800, 811, 812 and 813 traps for straight-through pipe connections.



Cross-section of No. 801 trap for standard angle pipe connections.



No. 211-216, Bottom inlet Type

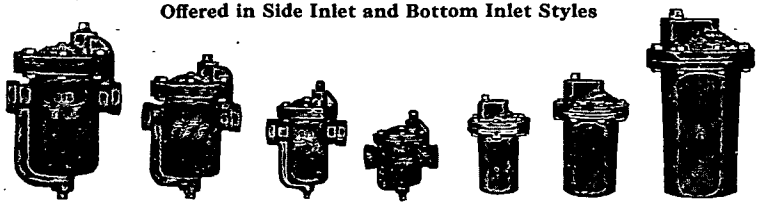
Side Inlet Traps

Trap Size		No. 800	No. 811	No. 812	No. 813	No. 801
Pipe Connections		½" or ¾"	½" or ¾"	½" or ¾"	¾" or 1"	½" or ¾"
List Price (Regular)		$7.00	$10.00	$16.00	$22.00	$7.00
List Price (Blast Trap)		$8.50	$11.50	$18.00	$24.00	$8.50
Telegraph Code (Regular)		Aloe	Brown	Cherry	Dawn	Arrow
Telegraph Code (Blast Trap)		Aloette	Brownette	Cherette	Dawnette	Arrowette
Dimension A		3¾"	3¾"	5⅝"	7"	3¾"
" B		5⅛"	6⅝"	8¹³⁄₁₆"	11¼"	6"
" C		5"	5"	6½"	7¼"	
" D		....	....	....	....	2⅛"
" E		....	....	....	....	1⅛"
Number of Bolts		6	6	6	6	6
Diameter of Bolts		¼"	¼"	⅜"	½"	¼"
Weight		4½ lbs.	5½ lbs.	13½ lbs.	25 lbs.	4½ lbs.
Maximum Pressure, lbs.		125	250	250	250	125
Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information see the Capacity Chart in Armstrong Steam Trap Book.	Lb Pressure 5	450	830	1600	2900	450
	10	560	950	1900	3500	560
	15	640	1060	2100	3900	640
	20	690	880	1800	3500	690
	30	500	1000	2050	4000	500
	50	580	840	1900	4100	580
	70	660	950	2200	3800	660
	100	640	860	1800	3600	640
	125	680	950	2000	3900	680
	150		810	1500	3500	
	200	*See Note at right.	720	1200	3200	*See Note at right.
	250		760	1300	3500	

UNIT HUMIDIFIERS Steam Type

Armstrong Steam Type Unit Humidifiers offer a low-cost, practical method of preventing the evils of winter heating-dry air. Improve worker comfort and safety; prevent drying out of materials. Write for new Bulletin No. 158.

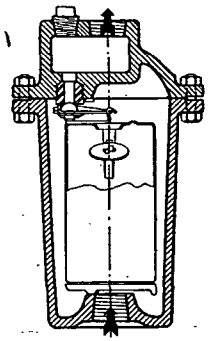
Offered in Side Inlet and Bottom Inlet Styles



No. 813 *No. 812* *No. 811* *No. 800* *No. 211* *No. 212* *No. 213*

4. The wearing parts in all Armstrong Traps are identical in design, material, and precision workmanship with parts used in Armstrong Forged Steel Traps for pressures up to 2400 psi and total temperatures of 850 F.

Armstrong Steam Trap Book. This 36 page book gives complete information on all sizes and types of Armstrong Traps. It also contains 17 pages of data on the subject of trap selection, installation, and maintenance. A free copy will be mailed on request.



No. 211-216, Blast Type

Armstrong Steam Type Humidifiers

For use during the heating season, when relative humidity is low, moisture absorbing materials dry out, and static electricity accumulations are sources of fires and explosions.

Operation is automatic and dependable. Relative humidity is maintained within close limits. Operating costs are very low, installation costs moderate, and maintenance costs are low.

Both air operated and electrically operated models available with capacities to 850 lb steam per hr. Send for Bulletin 158.

Bottom Inlet Traps

Trap Size		No. 211	No. 212	No. 213	No. 214	No. 215	No. 216
Pipe Connections		½"	½" or ¾"	½" or ¾"	1"	1" or 1¼"	1½" or 2"
List Price (Regular)		$ 9.25	$15.00	$20.75	$29.00	$38.00	$55.00
List Price (Blast Trap)		$10.75	$17.00	$22.75	$31.50	$40.50	$60.00
Telegraph Code (Regular)		Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)		Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height.....Dimension B		6⅜"	8"	10¼"	12½"	14"	16¾"
Diameter..... " A		4⅛"	5"	6⅜"	7½"	8½"	10⅝"
Wall Thickness " C		⅛"	¼"	½"	⅝"	⅝"	⅝"
Diameter of Bolts		¼"	¼"	⅜"	⅜"	½"	½"
Number of Bolts		6	8	6	8	8	12
Weight		5½ lb	10½ lb	19 lb	32 lb	47 lb	80 lb
Maximum Pressure, lb		250	250	250	250	250	250
Continuous discharge capacity in lb of water per hour at pressure indicated. For more complete information, see the Capacity Chart in the Armstrong Steam Trap Book.	Lb Pressure 5	830	1600	2900	4800	7600	14500
	10	950	1900	3500	5800	9000	17300
	15	1060	2100	3900	6500	10000	19200
	20	880	1800	3500	6000	8500	18500
	30	1000	2050	4000	6800	9800	18000
	50	840	1900	4100	6300	9000	18200
	70	950	2200	3800	6000	9200	18300
	100	860	1800	3600	6200	10400	18000
	125	950	2000	3900	6700	10900	20000
	150	810	1500	3500	5700	9500	18500
	200	860	1620	3200	5300	9200	17500
	250	760	1300	3500	5700	7000	19000

ADSCO
PRODUCTS for STEAM SERVICE

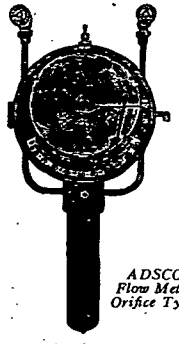
AMERICAN DISTRICT STEAM COMPANY

NORTH TONAWANDA, N.Y.

IN BUSINESS OVER 65 YEARS

Branches and Agents in Principal Cities

For Data on ADSCO Expansion Joints, refer to Insulation, Underground, page 1236.



ADSCO Flow Meter Orifice Type

ADSCO FLOW METER—ORIFICE TYPE

Exceptionally accurate at all rates of flow and will meter steam, water, gas or air. It is a compact unit for indicating, recording and integrating the flow and can be furnished in other combinations of these three devices. Easily installed and maintained by the purchaser. Frictionless meter mechanism, records on evenly-divided, direct-reading chart, giving a daily record from which to determine heating or processing costs. Write for Bulletin No. 35-83G.

Rotary Condensation Meter

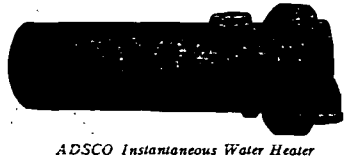
ROTARY CONDENSATION METER

Measures steam consumption by metering condensate from heating systems or industrial equipment. Accurate within 1 per cent and factory tested to 150 per cent of rated capacity. Compact, easily cleaned, tamper-proof and equipped with non-fogging counter mechanism. Counter reads directly in pounds. Suitable for vacuum or gravity service. Available in 7 sizes from 250-12,000 lb per hour capacity. Write for Bulletin No. 35-80AG.

ADSCO Vertical Steam Trap

ADSCO VERTICAL STEAM TRAP

A float type steam trap with or without thermostatic air by-pass for vacuum service to 15 lb pressure and gravity service to 125 lb pressure. The cover with all working parts can be removed without disturbing the piping connections. The trap is equipped with a reversible valve and reversible seat of stainless alloy steel. Write for Bulletin No. 35-86G.



ADSCO Instantaneous Water Heater

ADSCO HEAT EXCHANGERS

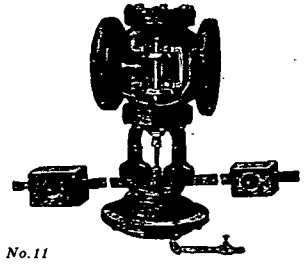
Made in various sizes and capacities to heat or cool water, oils, other liquids or gases according to expert engineering specifications. Simple in design, sturdy in construction, dependable and economical in operation. Available in U-tube or straight tube types of heaters, economizers, condensate coolers or special units. Write for Bulletin No. 35-75BG, 35-76G.

Mueller Steam Specialty Co., Inc.

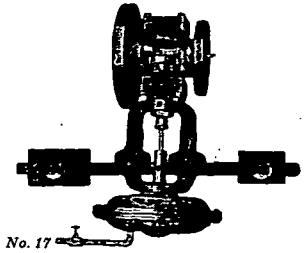
40-20 22nd Street, Long Island City, N. Y.

Steam, Water, Air, Oil and Gas Specialties for Heating and Power Plants

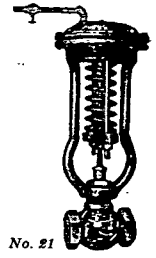
Pressure Reducing Valves—Straight Pattern and With Increased Outlet



No. 11



No. 17



No. 21

No. 11—For Vacuum, Vapor and Low Pressure Heating Systems. Initial Pressures, up to 200 lb; Reduced Pressures, 0 to 10 lb.

No. 17 and 21— For automatic control of reduced pressures on dead-end service, requiring a tight closing valve, such as tank heaters, kitchen utensils, sterilizing apparatus, laundry equipment, kettles, cookers, driers, etc. Initial Pressures up to 200 lb. Reduced Pressures 0 to 150 lb.

Constructed with full globe bodies. Center guide eliminates the wings on discs, and increases efficiency, assures minimum noise and prolongs the life of the seats and discs. Lever and weight operates on a steel roller bolt, assuring a most sensitive valve. Spring type furnished with special long springs for sensitive operation and wide ranges of reduced pressures.

No. 37

Water Pressure Reducing Valves

For controlling water pressures that require automatic and positive control of reduced pressures. Compact, and have full flow and renewable composition discs.

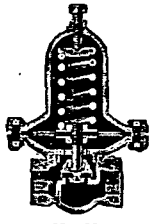
Standard and extra heavy weights suitable for all initial pressures and wide ranges of reduced pressures.

Diaphragm Operated Water Relief Valves

For relieving excessive pressure of hot or cold water, air, oil or gas.

Very sensitive, respond quickly.

Their construction is durable and compact, and accessible without disturbing pipe connections.



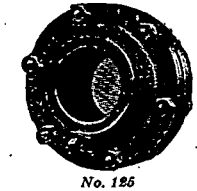
No. 63

Sediment Strainers for Steam, Water, Oil, Gas, Air, Etc.

For removing scale, cuttings, and other foreign matter from steam, water, air, oil and gas lines, or in connection with valves, pumps or other apparatus.

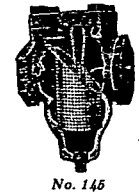
Furnished with iron bodies, plain or galvanized, with brass or copper mesh suitable for the service. Can be furnished all brass or nickel-plated iron or brass, or cast steel for extra high steam pressures, at additional cost.

Strainers with special mesh or of special metal furnished at a slight additional cost.

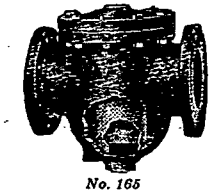


No. 126

Figures Nos. 145 and 165 Strainers have closed or open bottom baskets, so that the basket can be removed for cleaning, or all dirt and sediment can be blown out through bottom blow out connection.



No. 145



No. 165

CATALOGUE and BULLETINS covering our COMPLETE LINE gladly furnished on application.

Jas. P. Marsh Corporation

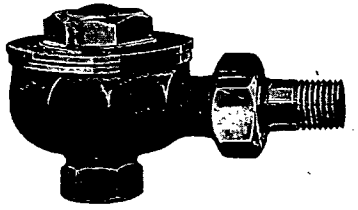
2072 Southport Ave., **Chicago 14, Ill.**

Branches in Principal Cities

Marsh products include: Pressure, Vacuum and Compound Gauges; Dial Thermometers; Steam Traps; Air Valves and Vents; Packless Radiator Valves and other heating specialties.

Thermostatic Diaphragm Radiator Traps

These efficient traps are equipped with a phosphor bronze diaphragm, consisting of two wafers of tinned phosphor bronze, drawn and spun to perfection of temper.

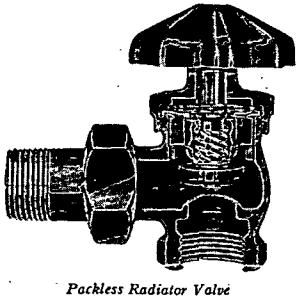


Thermostatic Diaphragm Radiator Trap

The wafers are spun together and soldered to form a seamless, sensitive, powerful expanding member—not easily fouled by dirt and foreign matter. Diaphragms are charged with a volatile fluid making them self-equalizing for use on pressures below atmosphere to 15 lbs. gauge. Traps are factory adjusted.

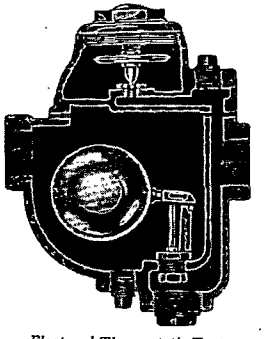
Packless Radiator Valves

Marsh all-metal packless valves are truly packless. These valves contain no packing to deteriorate, wear or crack, and are simple in design, with ample strength where strength is required. The principles upon which they are designed have been proved sound over many years of service. Valves operate easily—opening or closing tightly with less than one full turn. All valves are individually tested. Adaptable for use on hot water—forced or gravity systems—as well as all steam heating systems.



Packless Radiator Valve

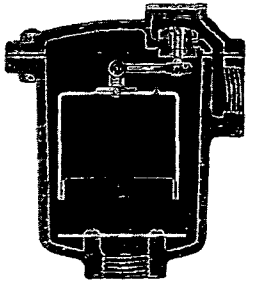
Marsh No. 12
Float and Thermostatic Traps



Float and Thermostatic Trap

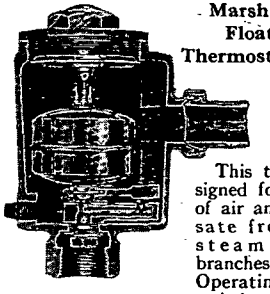
Marsh Heavy Duty Float and Thermostatic Traps are designed for removal of air and condensate from steam mains, branches, or risers, unit heaters, steam coils, etc. The size and weight of the trap permits installation in the piping without any other means of support. Condensation is discharged through a float operated valve located at the lowest point inside the trap body. Air vent is located in a by-pass in the cap or cover of the trap. Air passes through a passageway and out through the trap outlet. Construction permits removal of mechanism without disturbing the piping.

Marsh No. 500
Inverted Bucket Type Trap



This type is ideal for all types of hospital and kitchen equipment or similar service where a considerable volume of condensate is handled. The trap is self-venting, which, combined with the large water capacity assures unusually high efficiency in removing condensate, air and gases.

Marsh No. 17
Float and Thermostatic Trap

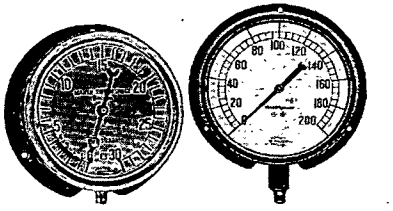


This trap is designed for removal of air and condensate from short steam mains, branches or risers. Operating characteristics adapt it for unit ventilators, unit heaters, and other equipment subjected to freezing temperatures when heating system is not in operation. Outlet discharge is water sealed at all times. Air vent is located in trap bonnet and air is normally discharged through a port directly to the outlet connection. A removable strainer protects mechanism. All working parts are made accessible by removing bonnet. The piping is all the support required for the No. 17 Trap.

Marsh Low Pressure Gauge

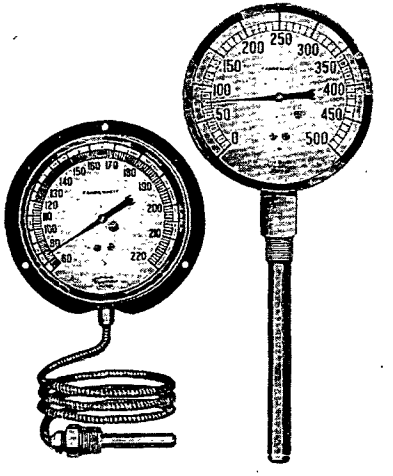
The Marsh A.S.M.E. standard, low pressure gauge will contribute to the economy and improve the operation of any type of steam boiler. It is finely built throughout and is available with the Marsh "Recalibrator" for quickly and easily resetting the hand to zero when the gauge is knocked out of adjustment.

Marsh Gauges include vacuum and compound types in a wide range of designs covering all services and pressures. Over 75 years of gauge manufacturing has reached its highest achievement in the Marsh "Mastergauge" for use where high pressures and temperatures are present and where maximum stamina and accuracy are essential.



Marsh Dial Thermometers

Have the same basic refinements found in Marsh Gauges. Typical Marsh Thermometers of bourdon tube type in self-contained and distant reading types are illustrated. They are available in either vapor tension or gas-filled types. Bimetallic types of dial thermometers are also available. Practically all temperature ranges up to 750° F. are covered. The "Recalibrator" is standard in all Marsh bourdon tube type thermometers. The Marsh line also includes recording thermometers.



Ask for complete information covering any problem involving traps, vents, gauges, dial thermometers, radiator air valves, packless valves, etc.

Products Research Co.

634 S. Western Ave., **Los Angeles 5, Calif.**

SALES OFFICES

NEW YORK · HOUSTON · SEATTLE
PORTLAND · SAN FRANCISCO · LOS ANGELES

Chrome Lock gasketing was especially developed for use in U. S. Naval Vessels. Ever since 1942, this type of gasketing has withstood the terrific vibrations of all types of ships in action, due to its tremendous cushioning effect.

Chrome Lock, this new type of gasketing of felt base, is impregnated with a chromate pigmented compound which renders it flame and corrosion-resistant. **Chrome Lock** holds up under pressure of 500 lb or more per square inch and is compressible to take up uneveness in flanges and seams. It is not affected by fuels and will retain its pliability under locked-in pressure. **Chrome Lock** is now offered for industrial uses.

ADVANTAGES

1. Flame-resistant
2. Warp-proof
3. Rust-inhibiting Qualities
4. Longer Life
5. Anti-wicking
6. Fuel-resistant
7. Adhesive to Metal
8. Anti-squeak
9. Non-oxidizing
10. Non-hygroscopic
11. Easy to Handle
12. Less Expensive

USES

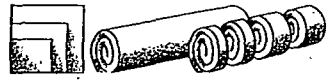
1. Air Conditioning
2. Heating
3. Refrigeration
4. Anti-squeak on Non-removable Fittings
5. All Piping
6. Joint Seals for: Diesel Oil, Fuel Oil, Gasoline, Mild Acids, Water (Fresh and Salt).
7. Water-tight and Air-tight Connections to Structures
8. Flange to Flange Joints

A FEW USERS

Bethlehem Steel Company
California Shipbuilding Corporation
Consolidated Steel Corporation
Kaiser Company, Inc.
Permanente Metals Corporation
Todd Shipyards Corporation
Western Pipe & Steel Co.
Todd Pacific Shipyards, Inc.

TYPES and FORMS

Sheets *Rolls* *Slit Rolls*



Sheets: 1/16 in. and 1/8 in. thick—36 in. x 48 in.

Rolls: 1/16 in. thick—36 in. wide—200 sq ft to roll.
1/8 in. thick—36 in. wide—100 sq ft to roll.

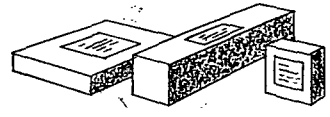
Slit Rolls: Sold by linear foot. Rolls approximately 50 linear feet in following widths: 1 in., 1¼ in., 1½ in., 2 in., 3 in., and other sizes on special order.

PACKAGING and SHIPPING

Sheets: In heavy cartons. 20 sheets of 36 in. x 48 in.—1/16 in. 10 sheets of 36 in. x 48 in.—1/8 in.

Rolls: 200 sq ft 36 in. wide—1/16 in.
100 sq ft 36 in. wide—1/8 in.

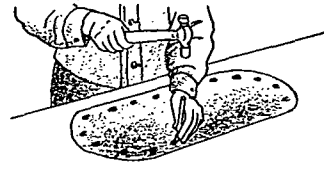
Slit Rolls: As ordered.



SPECIAL GASKETS

Special shapes or sizes can be cut economically and advantageously by any commercial gasket cutting company.

EASY TO CUT



A free sample has been reserved for you. Send for it today.

Wright-Austin Co.

309 West Woodbridge St., **Detroit 26, Mich.**

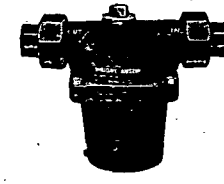
Established Over **50** Years Ago

PRODUCTS—Steam Traps, Strainers, Air Traps, Air Vents, Steam and Oil Separators, Compressed Air Purifiers, Exhaust Heads, Boiler Feeders and Controllers, Alarm Water Columns, Water Gauges, Trycocks, Illuminators and Protectors for Water Gauges.

"Airxpel" Bucket Type Steam Traps

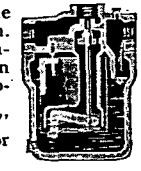
Are "double duty" traps, because they automatically discharge both air and condensate.

Union connections make them easy to connect up. Also, furnished with screw connections when desired. They save money for fittings and installation labor, by having straight through horizontal pipe connections.



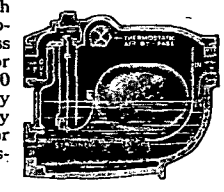
The Cub sizes are made in ½ in., ¾ in., 1 in. Especially suitable for individual unit drainage on heating and process equipment.

Also three "Master" sizes ½ in. to 2 in., for general service.

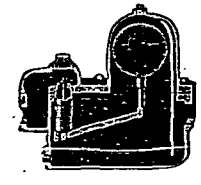


"Combination" Steam Traps

Float Type with internal thermostatic air bypass and strainer for pressures 0 to 40 lb. A modernly designed and very successful trap for vacuum and pressure heating.



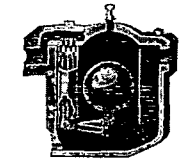
"Victor" Low Pressure Steam Traps

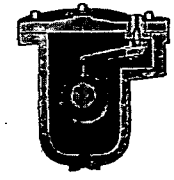


A heavy duty trap for large volumes of condensation at low pressures.

"Emergency" Float Type Steam Trap

Three valve trap with large capacity at high pressures. An exceptionally reliable trap for use in inaccessible places.

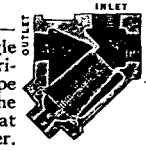




Air Relief Traps

For relieving air from forced circulation hot water heating systems, water supply lines, closed tanks, receivers, pumps, etc.

"Tuway" Strainer



May be used two ways—as a straight-way or angle strainer, in either horizontal or vertical pipe line, because it has the choice of two inlets at right angles to one another.

Separators—Steam and Oil

Type "A" Vertical Steam

Type "S" Horizontal Oil

We make separators of every type and all sizes for all pressures and conditions, to remove moisture from live steam, vibrating steam lines, oil from exhaust steam, etc.

Exhaust Heads

Designed to eliminate noise and spray. Three types to select from—the "Cyclone" Heavy Duty, and Standard Galvanized Steel—also, the cast iron type, to remedy all conditions. Sizes 1 in. to 48 in.

Send for descriptive Bulletins on any of the items listed on this page.

Yarnall-Waring Company

Manufacturers of YARWAY Steam Specialties

7600 Queen Street, **Philadelphia 18, Pa.**

YARWAY IMPULSE STEAM TRAPS

Construction—The Yarway Impulse Steam Trap is unique in that there is only one moving part, the simple valve F. This trap is made of bar stock throughout, no castings used. For pressures up to 400 lb, body and bonnet of cold rolled steel, cadmium plated; cap of tobin bronze, valve and seat of heat treated stainless steel. For pressures 400 to 600 lb, trap is all stainless steel.

Operation—Movement of valve (F) is governed by changes in pressure in control chamber (K). At lower temperatures, condensate bypassing continuously through orifice in center of valve reduces chamber pressure below inlet pressure and valve opens, allowing free discharge of air and condensate through seat. As condensate approaches steam temperature, low chamber pressure causes vaporizing and the increased volume builds up pressure in control chamber, closing valve (F).

Advantages

Light Weight—Yarway traps need no support—½ in. trap weighs only 1⅜ lb. 2 in. trap weighs 8⅝ lb.

Small Size—Can be installed in cramped quarters—½ in. trap measures 2¼ in. long—2 in. trap, 4¾ in. long.

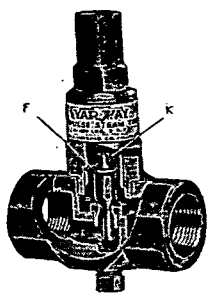
Will not air bind. Require no priming. Insure quick heating.

Operate on exclusive Impulse principle (U.S. Patents No. 2,051,732 and 2,127,649.)

Low Price—Often cheaper than repairing old traps.

Factory set to operate at all pressures up to 400 lb (or 600 lb) without change of valve seat.

Send for descriptive Bulletin T-1739.



List Prices, Weights and Dimensions

No. 60 Series—up to 400 lb. and

No. 120 Series—up to 600 lb.

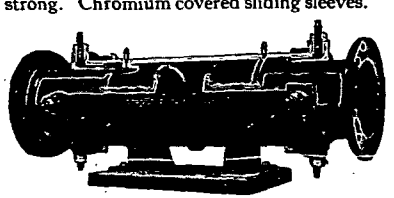
Size	Trap Complete	Weight Pounds	Length Inches
½" Nos. 60 or 120	$15.00	1¼	2⅞
¾" Nos. 61 or 121	22.00	2	3
1" Nos. 63 or 123	31.00	2½	3⅜
1¼" Nos. 64 or 124	48.00	4	3¾
1½" Nos. 66 or 126	68.00	5¾	4¼
2" Nos. 67 or 127	90.00	8½	4¾

Yarway Fine-Screen Strainers offer better protection against rust, scale and dirt for all steam equipment.

Made in six standard sizes from ¼ in. to 3 in. Cadmium plated, inside and out. High grade Monel woven-wire screens. Many thousands in use. Write for Bulletin S-201.

YARWAY EXPANSION JOINTS

All-steel welded construction; light but strong. Chromium covered sliding sleeves.



Cylinder guide and stuffing box integral, assuring perfect alignment. Internal limit stops. Gun-pakt and Gland-pakt types: Gun-pakt (illustrated) fitted with screw guns which permit addition of plastic packing while joint is under pressure. Sizes 2 in. to 24 in., single end or double end, flanged or welding ends; 150, 300 and 400 lb pressures. Also all-brass joints, ¾ in. to 2 in. For additional details send for Bulletin EJ-1911.

Anderson Products

INC.

Cambridge 39, Massachusetts

Vent-Rite Air and Vacuum Venting Valves for Radiators
Vent-Rite No. 66 Control Valves • Vent-Rite Unit Heater Valves

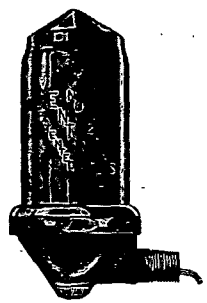
For Profitable, Efficient Venting Valve Installations . . . Specify Vent-Rite

The elimination of valve trouble caused by unavoidable accumulation of dirt can immediately be effected by use of Vent-Rite Air and Vacuum Venting Valves. Every Vent-Rite Valve can be quickly and easily taken apart, cleaned, reassembled and adjusted, and restored to its original working condition.

In rare cases where a part may have become damaged, the valve may be sent to the factory for complete reconditioning. *Only Vent-Rite Valves offer these advantages.*

It is easy to obtain the correct venting rate in each radiator in a system, regardless of size or location, because every Vent-Rite Valve has an exceptionally wide and complete range of venting rates. Adjustment of the venting rate at the individual valves is easily accomplished through a streamlined, convenient adjusting device. You can be sure of Balanced Radiation . . . of even, uniform distribution of heat.

Vent-Rite Air and Vacuum Valves are built for both one- and two-pipe systems in a wide range of types, sizes, outlets, and venting capacities. (The Vent-Rite Line includes Non-Vacuum Valves Nos. 1, 51, 3, 5A & 55, and Vacuum Valves Nos. 2, 62, 4, 6A & 66.)



No.2

REPAIRABLE

All Vent-Rite Valves are Repairable

THE VENT-VAC SYSTEM Even temperatures are provided with the Vent-Vac System by continuing the distribution of steam between firing periods. The steam is available through the use of heat left in the boiler and is distributed to the points where heat loss is greatest. To insure fast, uniform distribution of steam during the firing periods, it breaks the vacuum used between firing periods. This *breaking* of the vacuum occurs at a predetermined point, within the limits of the control, before the start of every firing cycle, provided the heat transfer from boiler to radiation has been completed. Vent-Rite Vacuum Valves are used in conjunction with a Vent-Rite Control Unit. The system is simple, economical, and amazingly effective. Vent-Rite Control Units not only create vacuum in the system between firing periods, but also limit the amount of vacuum that can be created to the point beyond which the distribution of excessively expanded vapor would be inefficient.

VENT-RITE CONTROL VALVE Vent-Rite Control Valve No. 66 is the heart of the Vent-Vac System of steam control for automatically fired one-pipe systems. It takes the place of a main line vent, limits the amount of vacuum created, and breaks the vacuum at a pre-determined point, within the limits of the control, before the start of every firing cycle, provided the heat transfer from boiler to radiation has been completed. It is entirely mechanical. With the Vent-Vac System, using a No. 66 Control Valve, a system is *vacuum* between firing periods, *non-vacuum* during the firing, combining the best of both systems and assuring *Balanced Radiation* . . . assuring even distribution of heat to give more comfortable rooms for better living. Send for Bulletins.

No. 66

Vent-Rite Pioneered

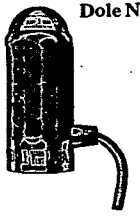
- **CONTROLLED VENTING**
- **BALANCED RADIATION**
- **REPAIRABLE VALVES**
- **VENT-VAC SYSTEM**

The Dole Valve Company

Main Offices and Factory: 1933 Carroll Avenue, **Chicago 12, Ill.**

THE ALL STAR LINE **Dole** AIR AND VACUUM VALVES

Selecting the right vent for a particular purpose is your assurance of the utmost efficiency and economy from one pipe steam heating systems. The Dole line covers every venting need and offers a complete choice for every purpose.

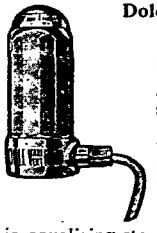


Dole No. 1A Vari-Vent Air Valve

Modern gas, oil or stoker fired one pipe steam systems require QUICK venting. This radiator valve lets air escape twice as fast as ordinary valves and balances the flow of steam at the first "breath" of boiler pressure. Adjustable vari-vent feature gets air out of those "far away" radiators as quickly as those close to the boiler.

Dole No. 3 Air Valve

Vents radiators of hand fired gravity steam heating systems. Double shell construction provides separate passages for air and condensation—extra large float defeats spitting or water leakage. Complete venting assured at pressures up to 10 lbs.



Dole No. 2B Vari-Vent Vacuum Valve

Adjustable radiator valve for "vacuumizing" and balancing gravity steam heating systems. Patented Dole bellows vacuum seal locks out air after it has been once expelled from the system. Easily adjusted vari-vent feature assists in equalizing steam flow to all radiators.

Dole No. 1933 Air Valve

Low cost valve for venting radiators of hand fired systems. Large float provides a seal against condensation to stop spitting.

Dole No. 1B Vari-Vent Air Valve

Balances the flow of steam to convectors, either cast iron or copper, of automatically fired systems.

Dole No. 1C Quick Vent Float Valve

Vents mains and speeds flow of steam to radiators of automatically fired systems. Extra large venting port.

Dole No. 5 Quick-Vent Float Valve

Vents steam mains on hand fired systems. Positive seal against water.

Dole No. 4 Quick Vent Valve

For quick venting mains that end 18 in. or more above the boiler water line.

Dole No. 103 Vacuum Valve

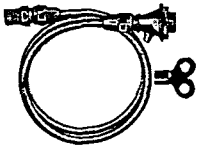
For venting convectors, ceiling radiators and pipe coils of "vacuumized" gravity steam systems.

Dole No. 6B Vacuum Valve

Vents the mains of "vacuumized" one pipe steam systems. Prevents the return of air. Closes against water.

Dole No. 14 Key Valve

Low cost venting device for concealed radiators and convectors of hot water heating systems. Protects panel fronts from rusty water stain.



Write The Dole Valve Company for complete catalog and handy selector chart which indicates the Dole Air or Vacuum Valve most suited for a particular need.

Jenkins Bros.

BRONZE - IRON - STEEL VALVES

Mechanical Rubber Goods

80 White St., New York 13, N. Y.; 524 Atlantic St., Boston 10, Mass.; 376 Spring St., N.W., Atlanta 3, Ga.; 133 N. Seventh St., Philadelphia 6, Pa.; 1514 Fulton St., Chicago 7, Ill.; 510 Main St., Bridgeport 9, Conn.; 660 Market St., San Francisco 4, Cal.

Jenkins Bros., Ltd.: Montreal; Factory, Lachine, Canada · London, Eng.

IN VALVES **Jenkins** GIVES YOU EVERYTHING

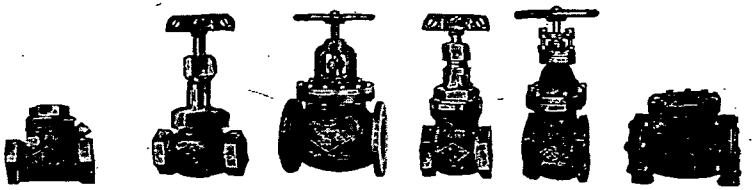


Fig. 762 Bronze Regrinding Swing Check | *Fig. 106A Bronze Globe, Renewable Comp. Disc* | *Fig. 613 Iron Body Regrinding Globe* | *Fig. 370 Bronze Gate* | *Fig. 325 Iron Body Gate* | *Fig. 624 Iron Body Regrinding Swing Check*

OVER 500 DIFFERENT JENKINS VALVES COVER EVERY HEATING AND AIR CONDITIONING NEED

To adequately describe the complete Jenkins line of valves requires a Catalog of more than 400 pages. There are over 500 different types and patterns of valves that bear the trusted "Diamond" trade mark. Practically speaking, Jenkins can furnish any valve that you may require for plumbing, heating, air conditioning, general industrial or engineering service.

General Classifications of Jenkins Valves Include—Bronze Valves fitted with Jenkins renewable composition disc. Bronze Regrind-Renew Valves with bevel and plug type seats. Bronze Gate Valves. Iron Body Valves fitted with Jenkins renewable composition disc. Iron Body Regrinding Valves. Iron Body Gate Valves with solid wedge and double disc parallel seats. All-Iron Valves.

Air Furnace Malleable Iron Valves. Cast Steel Gate, Globe and Swing Check Valves. Electrically and Hydraulically Operated Valves. Radiator Valves. Fire Line Valves. Quick-opening and Self-closing Valves, Needle Valves, Y Valves, Solder-End Valves, Stainless Steel Valves.

Other Jenkins Products Are—Composition Valves Discs exactly suited to service conditions. Friction Tape, Splicing Compound, Mechanical Rubber Goods.

JENKINS VALVES ARE SOLD BY GOOD SUPPLY HOUSES EVERYWHERE

Consult A-Z Listing of Thomas Register For the One Nearest You.

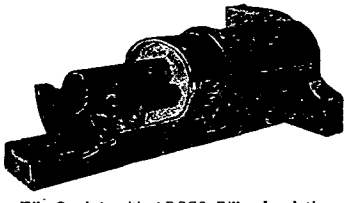
ADSCO
PRODUCTS for STEAM SERVICE

AMERICAN DISTRICT STEAM COMPANY

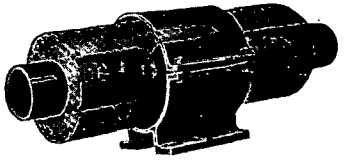
NORTH TONAWANDA, N.Y.

IN BUSINESS OVER 65 YEARS

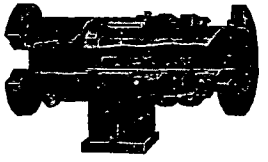
Branches and Agents in Principal Cities



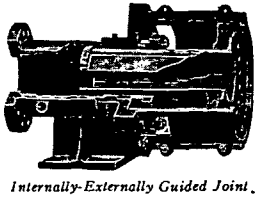
Tile Conduit with ADSCO Filler Insulation—a "Fiberglas" Product



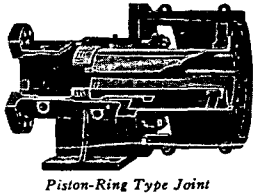
Alignment Guide



Internally Guided Joint



Internally-Externally Guided Joint



Piston-Ring Type Joint

OVER 65 YEARS EXPERIENCE IS BUILT INTO THE DESIGN AND MANUFACTURE OF DEPENDABLE ADSCO PRODUCTS FOR PIPE LINES

For over 65 years ADSCO engineers have specialized in the design and application of pipe fittings and accessory equipment for underground and surface steam, water, oil and other piping systems. An extensive, modern plant including foundry, machine shop, casing mill, shipping and storage facilities enable ADSCO to produce high grade products by skilled workmen under expert supervision.

LEADING MAKERS OF EXPANSION JOINTS

As pioneer manufacturers of expansion joints for pipe lines, ADSCO is the largest single producer of such equipment in the world. We offer the most extensive line of packless and slip type joints in various types to meet the requirements of any pipe line expansion and contraction problem. In addition, ADSCO produces all of the related equipment necessary to the permanent installation of efficient pipe lines, including tile conduit for underground lines, pipe supports, saddle plates, alignment guides, steam traps, condensation and flow meters, storage and instantaneous water heaters, strainers, separators, manhole frames, and vapor heating specialties.

ENGINEERING ASSISTANCE

ADSCO engineers welcome the opportunity of working with you in the solution of your pipe line expansion and contraction problems and correspondence is invited giving the details of any proposed piping installation.

WRITE FOR ADSCO CATALOG No. 35

All ADSCO products are illustrated and described in the latest ADSCO Catalog No. 35 containing informative data for the specification and purchase of dependable products for underground or surface pipe line distribution systems. Write for your copy today to the American District Steam Company, 65 Bryant St., North Tonawanda, New York.

H. W. Porter & Co.

INCORPORATED

817-G Frelinghuysen Ave., **Newark 5, New Jersey**

Permanent Protection and Insulation for Underground Pipe Lines

BALTIMORE, MD. CHARLOTTE, N. C. RICHMOND, VA.

Also sold and installed by Johns-Manville Construction Units in all principal cities.

Therm-O-Tile

REG. U.S. PAT. OFF.

STEAM CONDUIT SYSTEMS

"Permanent" Protection is Important—Therm-O-Tile has been on the market over a dozen years. Its design is well known to all leading heating and ventilating engineers but we would again like to point to the great importance of "Permanent protection."

In previous issues of this book we stated that Therm-O-Tile is "A complete conduit system for the permanent support, protection, and insulation of underground mains for steam distribution."

Beware "Temporary" Protection

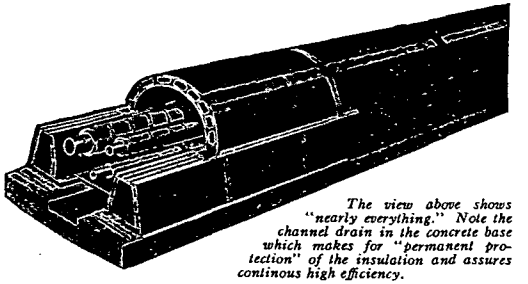
We consistently emphasize the word "permanent." It is easy to provide "temporary" protection and insulation for underground pipe lines. Threads and joints don't fail immediately, and foundations don't sag immediately, but unless the job is *properly* done it won't be long before water seeps in and ruins the insulation. With wet and spoiled insulation, efficiency drops drastically. And, unless the conduit is built on a solid foundation there will be sagging and collection of water in pockets. Be sure of a "permanent" job.

Therm-O-Tile Drainage System—In the Therm-O-Tile concrete base there is a drainage channel—clearly visible in this photo—which carries off all water that may enter the conduit from any source, thereby keeping the insulation *permanently* dry. Drainage is entirely internal. The channel is accurately and *permanently* sloped so that condensate or other pockets cannot form. Open to thorough inspection at any time at manholes. Amply large to keep the pipe space dry at all times.

"Spread Footing" Foundation—So that there will be no settling or sagging the Therm-O-Tile "Spread Footing" foundation base is a thick concrete slab poured directly in the trench bottom as shown in the photograph. It is steel reinforced or placed on driven piles when installed over filled or boggy ground—constructions that are necessary for *permanent* protection.

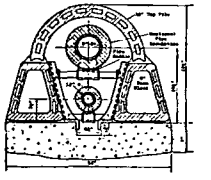
See Previous Issues—In issues of this GUIDE previous to 1944 we gave details regarding the Tile Envelope which produces 27 different conduit cross sections. We told about the ideal accessibility of this conduit, its great strength, how it is water-proofed, and how the insulation is applied. For complete information ask for a copy of Bulletin 381.

Competitive in Cost—Despite the high efficiency, greater strength, dependability, and other outstanding features that are obtained in Therm-O-Tile, it is nevertheless competitive in total first cost. Final cost is much less, thanks to its permanence. Thorough investigation is invited.



The view above shows "nearly everything." Note the channel drain in the concrete base which makes for "permanent protection" of the insulation and assures continous high efficiency.

We Co-operate—Our engineers have had unusual and broad experience in the design and installation of steam conveying equipment for an exceptional range of purposes. They will gladly cooperate with you in the solution of your own individual problems of this nature.



Single or Multiple Pipe Lines Using Sectional Pipe Insulation.

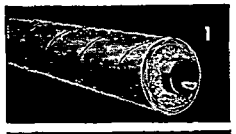
Showing a typical Therm-O-Tile piping arrangement when there are two pipe lines. Note the channel drain which "permanently protects" the insulation.

The Ric-wiL Company

INSULATED PIPE CONDUIT SYSTEMS

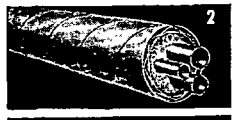
Union Commerce Bldg., Cleveland, Ohio • Agents in Principal Cities

There is a Ric-wiL insulated conduit system engineered to your specific needs—the transmission of steam, hot water, oil, hot or refrigerated process liquids—providing heat transfer with the lowest possible loss.



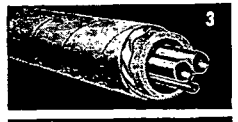
1. RIC-WIL INSULATED PIPE UNIT—SINGLE PIPE SYSTEM

Prefabricated complete units—pipe as specified, thoroughly insulated, in helical corrugated conduit, coated and wrapped with asphalt saturated asbestos felt. 21-ft lengths for speedy installation. For underground or overhead systems.



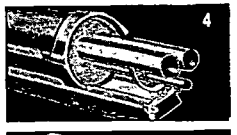
2. RIC-WIL INSULATED PIPE UNIT—MULTIPLE PIPE SYSTEM

Any specified combination of pipes in prefabricated conduit—insulated and protected the same as the single pipe system. Any or all of the pipe lines may be specially insulated to meet job requirements.



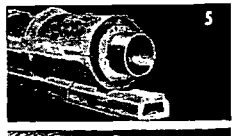
3. RIC-WIL INSULATED PIPE UNIT—FOR PROCESS LIQUIDS

An adaptation of the multiple system used where a steam or hot water line heats fluids in other lines. Pipes are insulated from the exterior but not from each other. Sizes and specifications as required—conduit same as for other insulated pipe units.



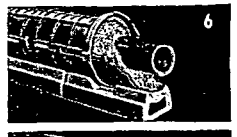
4. RIC-WIL STANDARD TILE CONDUIT—TYPE F

Vitrified glazed A.S.T.M. Standard Tile Housing—acid and waterproof—with foundation type base drain supporting weight of piping through correctly engineered pipe supports. Positive locked-in-place cement seals on sides and ends. For single or multiple pipes.



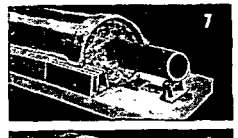
5. RIC-WIL SUPER TILE CONDUIT—TYPE F

Same advantages as Standard Tile but with walls approximately double thick for strength under heavy traffic or where overhead load is above normal. Will support concentrated static load of 6 tons per wheel under actual installation conditions. Base drain of extra-heavy tile.



6. RIC-WIL CAST IRON CONDUIT—TYPE F

Heavy reinforced cast iron conduit for use where underground pipe lines run close to or under railroad tracks. Durable, water-tight and vibration-proof. Positive locked-in-place cement seals on sides and ends with metal clamps for extra tightness.



7. RIC-WIL TILE CONDUIT—UNIVERSAL TYPE

Where installation conditions dictate the use of a concrete pad Ric-wiL Universal Tile is recommended. Side walls are double-cell vitrified trapezoidal block design. Arch may be Standard Tile, Super-Tile, or Cast Iron.

8. RIC-WIL TILE CONDUIT—TYPE DA

For oil or process liquids where conduit must be insulated but individual lines are not insulated from one another. Insulation is a diatomaceous earth lining, moulded and keyed to inside of tile. May also be used (Type DF) with fibre insulation for steam heat, power and superheated steam. Applicable to Standard, Super-Tile and Cast Iron.

Ric-wiL accessories are available in all type systems; standard and special fittings, factory fabricated or field fabricated expansion devices, alignment guides, and anchors. Descriptive bulletins on request.

GET THE ORIGINAL—SPECIFY RIC-WIL

American 3 Way-Luxfer Prism Co.

24 N. Pulaski Rd., **Chicago-24, Ill.**

3-WAY Daylighting Products

103 Park Ave., **New York-17, N. Y.**

AMERICAN GLASS BLOCK SKYLIGHTS

Light-Up—the AMERICAN *Way!*

American 3-Way Rooflights make use of Glass Blocks of special design and strength, manufactured by the proven process, incorporating four way design lenses on inner surfaces of plates, also resulting in uniform even light distribution over wide areas, leaving top and bottom surfaces smooth for easy cleaning.

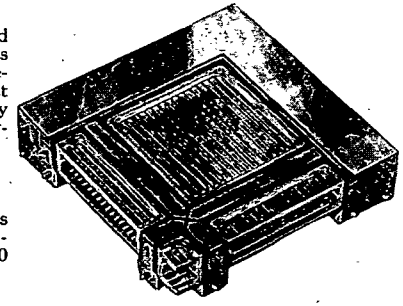
Glass Blocks are 9 in. x 9 in. x 2½ in. and spaced approximately 10⅝ in. on centers.

Low Heat Transfer

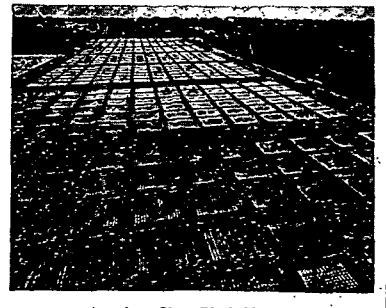
Tests conducted by methods suggested by the A.S.H.V.E. Code show that Glass Block Rooflights have about two and one-half times the insulating value of sheet metal skylights with no heat losses by "escape," since the construction is air-tight.

Solar Heat Transmission

Reduction in total solar heat gain as compared with ordinary windows is indicated by relative values given in Table 10 and Table 12 in Chapter 15.



Section of American Glass Block Skylight Showing Method of Block Application



American Glass Block Skylight

Triple Plates of Glass

Magnalite Diffusing Glass units may be attached to under side of glass block construction thus making for effective uniform light diffusion and even distribution; also very effective in condensation problems.

Insulated Construction

Construction of rigid reinforced concrete grids can be arranged with insulation materials sufficient to approximately equal the performance of the glass blocks.

Condensation

Due to the nature of the grid construction where insulation materials are employed with semi-vacuum glass blocks assemblies there is little or no tendency for condensation to form on the under side. Should relative high humidities or abnormal conditions exist, further insulation treatment can be provided.

Glass Block Assemblies for all off-vertical arrangements are available. Details will be furnished on request.

3-Way Glass Block Skylights may also be furnished without special insulation treatment in reinforced concrete grid construction.

Write for Complete Information

Ingersoll Steel Division

Borg-Warner Corporation

310 So. Michigan Ave., **Chicago, Illinois** **Distributors in the Principal Cities**

KOOLSHADE

YOU CAN ELIMINATE SWELTERING SUMMER HEAT THAT CAUSES DISCOMFORT . . . CUTS WORKERS EFFICIENCY . . . ADDS TO REFRIGERATION COSTS—*BY INSTALLING*

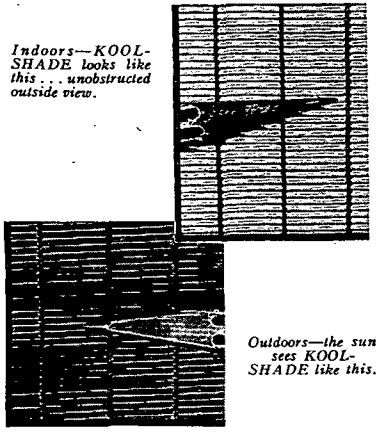
KoolShade Sun Screen

THE AUTOMATIC "window" insulation that controls sun heat and still provides ample light and visibility. KOOLSHADE Sun Screen is—in effect—a bronze *outside* miniature Venetian blind with the louvres permanently slanted outward and down at a 17 deg angle to block, reflect, absorb and radiate as much as 90% of the sun's heat rays—*outside the window!*

Air Conditioning Engineers, during the past 7 years, have proved that KOOLSHADE:

- Frequently eliminates the need for zoning.
- Maintains lower temperatures in non-cooled rooms.
- Reduces size of Refrigeration installations. (*100 sq ft of KOOLSHADE on sun-exposed glazed areas is equal to approximately one ton of equipment.*)
- Cuts equipment operating costs.

Because KOOLSHADE needs no adjusting it works most efficiently during peak solar loads. (See table on opposite page.)



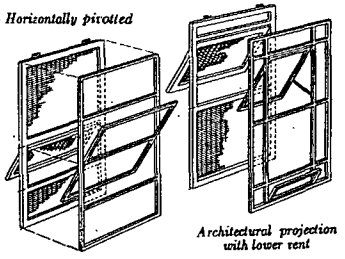
Indoors—KOOLSHADE looks like this . . . unobstructed outside view.

Outdoors—the sun sees KOOLSHADE like this.

KOOLSHADE can be adapted to any type window

No matter what your particular problem, authorized KOOLSHADE distributors have the answer. Remember that KOOLSHADE also gives insect protection as an added feature.

In comparison with these methods . . . which, at best do only part of the job, KOOLSHADE stops heat rays *outside* the room and does not obstruct ventilation or outside view.



Horizontally pivoted

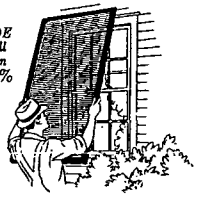
Architectural projection with lower vent

Half-Drawn Shade Sun Load 68% or more

Inside Venetian Blind Sun Load 58% or more

Awnings in Use Sun Load 22 Per Cent to 28 Per Cent

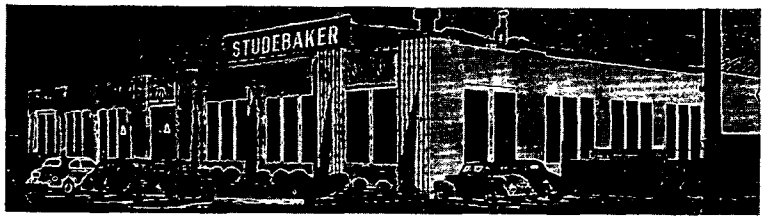
Outside Venetian Blind or Shutter Sun Load 22 Per Cent



KOOLSHADE easy to install and maintain Sun Load 10%

KOOLSHADE has been proved in actual use

Below are listed a few leading Industrial users of KOOLSHADE—the scientific sun heat control.



KOOLSHADE NEVER MARS ARCHITECTURAL HARMONY

The Hoover Company	No. Canton, Ohio
American Tobacco Company	Durham, N. C.
Easy Washing Machine Company	Syracuse, N. Y.
Ford Motor Company	River Rouge, Mich.
General Electric Company	Chicago, Ill.
Inland Steel Company	East Chicago, Ill.
Pullman-Standard Car Mfg. Company,	Chicago, Ill.
Radio Corporation of America	Bloomington, Ind.
Wright Aeronautical Corporation	Cincinnati, Ohio
Pacific Gas & Electric Company,	Sacramento, Calif.

The organizations listed above are typical of hundreds of satisfied KOOLSHADE users.

FOR INFORMATION regarding technical proof of performance and descriptive literature covering complete installation methods—write Ingersoll Steel Division.

STOPS HEAT OUTSIDE—KILLS GLARE

DATA FROM PITTSBURGH TESTING LABORATORY

FROM CALCULATIONS BASED ON ACTUAL TESTS

Solar Radiation Transmitted Through Windows Equipped with KOOLSHADE Sun Screen

For 40 deg. latitude, on July 21st All figures given represent B.t.u. per sq. ft per hour.

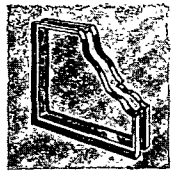
TIME	SOLAR RADIATION	NE	EAST	SE	SOUTH	
6 AM	Intensity Incident to Vertical Surface (1)	72	80	40		6 PM
	Transmitted thru Window with KOOLSHADE (2)	33	38.5	13		
7 AM	Intensity Incident to Vertical Surface	143	180	112		5 PM
	Transmitted thru Window with KOOLSHADE	38.5	60.5	22		
8 AM	Intensity Incident to Vertical Surface	143	211	155	8	4 PM
	Transmitted thru Window with KOOLSHADE	17	42.5	20.5	0	
9 AM	Intensity Incident to Vertical Surface	104	192	168	46	3 PM
	Transmitted thru Window with KOOLSHADE	8	22	17	2	
10 AM	Intensity Incident to Vertical Surface	46	143	156	77	2 PM
	Transmitted thru Window with KOOLSHADE	1.5	11.5	13	4	
11 AM	Intensity Incident to Vertical Surface		75	121	95	1 PM
	Transmitted thru Window with KOOLSHADE		3.5	8.5	5.5	
12 M	Intensity Incident to Vertical Surface			73	103	12 M
	Transmitted thru Window with KOOLSHADE			3.5	6	
TIME		NW	WEST	SW	SOUTH	TIME

(1) Data from the A.S.H.V.E. Guide 1940.

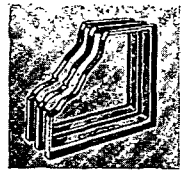
(2) Figures represent Solar Heat Gain in EXCESS of heat gain by conduction through window glass.

Libbey·Owens·Ford Glass Company

Nicholas Building, **Toledo 3, Ohio**



THERMOPANE
Insulating Glass



L-O-F Thermopane is a transparent factory-fabricated insulating glass unit composed of two or more panes of glass separated by ¼ in. or ½ in. of dehydrated captive air, hermetically sealed at the edges in the factory with a metal-to-glass bond.

Thermopane is Fabricated to ordered sizes at Libbey-Owens-Ford's factory. The glass, before being assembled into Thermopane units, is specially cleaned before the patented Bondermetic seal is applied. The layer of air inside the Thermopane units is scientifically cleaned, dried and hermetically sealed. The patented, metal-to-glass seal bonds the two or more panes of glass into one unit to prevent dirt and moisture from entering the air space.

Thermopane Reduces the coefficient of heat transmission; increases the room-side surface temperature, thus promoting radiant comfort and lowering the dew point; furnishes a control of light quantity and quality through combinations of various types of glass, and deadens sound transmission to some degree.

Double-glass Thermopane: Glazing of wood or metal windows or doors for practically any purpose in structures requiring heating or air conditioning.

Triple-glass Thermopane: Large stationary units such as insulated glass walls in homes, apartments, public and commercial buildings and in show windows for refrigerated display where temperature differential must be considered.

Quadruple-glass Thermopane: Engineered to meet low temperature and high humidity conditions.

Thermopane Uses are many, but may be briefly summarized as below:

Thermopane Units provide a high resistance to heat flow, varying with the number of panes and the thickness of the air space used. In summer the low heat transmission coefficient reduces the load on air conditioning systems. In winter it saves heat. The greater efficiency of Thermopane makes it possible to incorporate larger windows in houses and keep the cost of fuel constant. For example, a house could have 107 sq ft of Thermopane instead of 35 sq ft of single glazing and not lose any more heat. Thermopane permits the influx of solar heat in exterior glazing of buildings without a prohibitive compensating loss from conduction.

The over-all heat transmission coefficient *U* varies with the ranges of temperature at which the coefficient is determined. For most practical heat loss calculations coefficient *U* can be the value determined at 0 deg outside temperature, 70 deg inside temperature, 51 mph outside air velocity, 0.25 mph average inside air velocity. The following table gives such values:

Number of Panes	Thickness of Panes	*U* Values	
		¼ In. Air Space	½ In. Air Space
2	⅛"	0.62	0.57
	¼"	0.57	0.53
3	⅛"	0.42	0.37
	¼"	0.39	0.35

Note: *U* for single ¼ in. thick glass = 1.14, *R* = 0.88.
U for single ¼ in. thick glass = 1.07, *R* = 0.94.

The Room Side Surface Temperature of Thermopane is considerably higher than that of single glazing. Usually radiators or registers are near glass areas in buildings to offset conducted heat loss from a room and radiant loss from the bodies of persons near cold glass areas. The higher glass surface temperature of Thermopane greatly reduces the amount of heat which must be supplied, permitting more flexibility in room design.

Another Important Benefit from Thermopane is the prevention of frost or condensation from forming on the room-side surface of a single pane of glass in winter due to higher room humidity. The absence of condensation on the room-side surface of glass is of considerable importance where clear visibility is a factor as in residences, all types of commercial or industrial buildings and refrigerated display spaces.

More Complete Information on Thermopane is available by writing to Libbey-Owens-Ford, or its district office nearest to you, and requesting technical data sheets prepared by Don Graf, a general booklet about the product, and a brochure which discusses Solar Housing.

LIBBEY-OWENS-FORD GLASS COMPANY, Toledo 3, Ohio

Owens-Illinois Glass Company

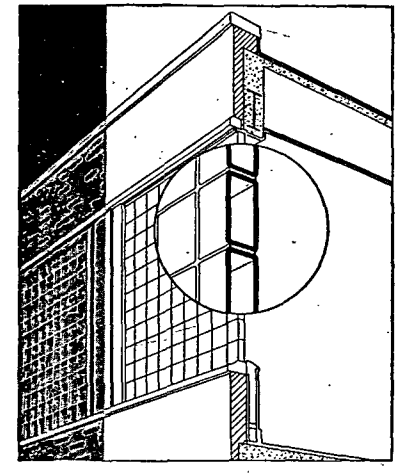
INSULUX PRODUCTS DIVISION, Toledo 1, Ohio

GLASS **INSULUX** BLOCK

Dealers in all principal cities

Insulux Glass Block Give Better Control of Interior Conditions

Insulux Glass Block are hollow, partially evacuated units, 3⅞ in. thick. Faces are smooth or ribbed. Solid panels of these block, laid in mortar, make a light-transmitting wall of high insulating value. Their proper use aids control of interior conditions to a point where initial and operating cost of heating or cooling equipment is reduced.



Conductivity

The *U* factor for ribbed glass block is 0.46; smooth face, 0.49. For design purposes, these factors may be used as constant for either 6 in., 8 in. or 12 in. block. See Chapter 6 of this volume for additional data.

This *U* factor is only 43 per cent of that of ordinary light-transmitting materials. The reason lies in the two heavy glass surfaces separated by partially-evacuated and hermetically sealed dead air space.

Surface Condensation

Because of the low overall air-to-air heat transfer, the room condensation point of a glass block panel is much lower than that of ordinary windows. This will permit higher humidities for both comfort air-conditioning and in industries where high humidity is part of the process. Glass block are not subject to deterioration caused by moisture.

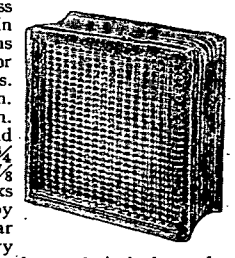
Infiltration

Insulux Glass Block are sealed in the building. They form a barrier against infiltration of dust, air and vapor. Winter drafts and summer vapor leakage are cut. Natural ventilation requirements can be met by installation of windows either inset in the panels or installed directly below the panels.

Solar Heat Gain

A comparative test showed 94 per cent more solar heat through steel sash than through glass block. However, as with sash, glass block transmit less solar heat when properly oriented and shaded. Data in Table 12, Chapter 15 of this GUIDE are for standard block. Other designs, such as the No-glare and Directional block afford further reduction. Data will be sent on request.

Design, Sizes, Erection



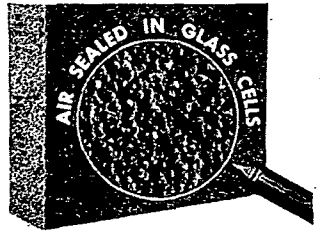
Insulux Glass Block is made in 10 face patterns for residential or industrial uses. Sizes are: 5¾ in. x 5¾ in., 7¾ in. x 7¾ in. and 11¾ in. x 11¾ in. All are 3⅞ in. thick. Blocks are erected by laying in mortar like any masonry material. *Complete technical data, description, etc., sent to you on request.*

Pittsburgh Corning Corporation

Room 616, 632 Duquesne Way, Pittsburgh 22, Pa.

PC FOAMGLAS—waterproof, fireproof—Insulation

Also manufacturers of PC Glass Blocks



PC FOAMGLAS is different from all other insulating materials, being composed of millions of minute cells of inert air sealed in glass. Thus PC Foamglas embodies not only remarkably efficient insulating qualities, it is also—*vaporproof, fireproof, waterproof and permanent.*

Protecting delicate high-speed machinery and goods in process, insuring continuous, efficient operation—demands rigid control of temperature and humidity in all sorts of mills and factories.

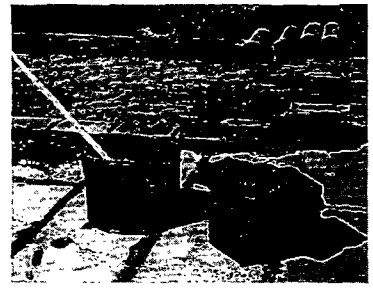
PC Foamglas, the new cellular glass insulation, has proved its ability to meet the most exacting requirements—efficiently and economically—in a wide range of industries.

On roofs and in core walls, floors, ductwork and processing equipment, PC Foamglas retains its full insulating efficiency throughout the life of the building. It is an efficient vapor-seal and water-stop, requires no repairs, maintenance or replacement of the material during ordinary use.

Check these features of PC Foamglas against other insulating materials.

1. ***Permanent insulation.***
2. ***Vermin- and vaporproof.***
3. ***Fireproof.***
4. ***Waterproof.***
5. ***Light weight.***
6. ***Easy installation.***
7. ***Rigid structure.***
8. ***Economical.***

For technical data on PC Foamglas, send for special bulletins.



Under severe service conditions such as the roof pictured here, PC Foamglas retains its insulating efficiency because it is waterproof and vaporproof.

When you install PC Foamglas Insulation in core walls, it becomes an integral part of the structure, assists in efficient, trouble-free control of temperature and humidity levels in the enclosed area—permanently.

SEND FOR FREE BOOKLETS—To find out how and why PC Foamglas will save time and money on all sorts of heating, ventilating and air-conditioning jobs, send for free copies of our booklets which describe its advantages when used on roofs, in core walls and in floors and on tanks, towers, ducts and other processing equipment.

s-07

Pittsburgh Corning Corporation

632 Duquesne Way, Pittsburgh 22, Pa.

Distribution through Pittsburgh Plate Glass Company warehouses in principal cities and by the W. P. Fuller Company on the West Coast. Also Manufacturers of PC Foamglas Insulation.

Glass Blocks allow the economical use of large glass areas, reduce heat loss in cold weather and materially aid air-conditioning. This is because each PC Glass Block contains a sealed-in dead-air space that is an effective retardant to heat transfer.

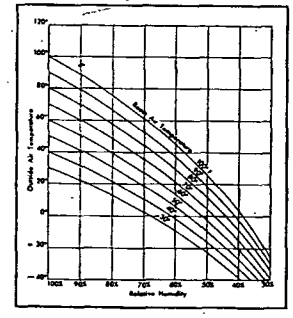
Thermal Insulation

Tests run by nationally recognized laboratories have established the value of glass blocks for insulation of light-transmitting areas. These tests have proved that with glass block panels, heat loss is slightly less than half that experienced with single-glazed windows. In computing heat losses through panels for most design purposes, it is recommended that a "*U*" value of 0.46 to 0.49 be used for all block sizes and face patterns. For complete data on heat transfer values see the section on heat transfer elsewhere in this Guide—page 138.

Surface Condensation

Due to high insulating value, condensation will not start forming on the room side of glass block panels until outside air has reached a temperature much lower than that necessary to produce condensation on single-glazed windows. The accompanying chart shows at what temperatures condensation will form.

Outdoor temperature required to produce condensation on the room side surface of PC Glass Block panels.



For example, with inside air at 70°F and relative humidity at 40 per cent, condensation will not begin to form on the interior surfaces of a glass block panel until an outdoor temperature of −14 deg is reached. Under similar conditions with single-glazed sash, moisture will begin to form when the outdoor temperature reaches +33°F.

Solar Heat Gain

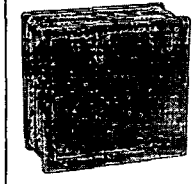
The use of glass blocks for light-transmitting areas results in a marked reduction in total solar heat gain as compared with ordinary windows. This factor is of considerable advantage in buildings that are properly air-conditioned, but does not eliminate the need for adequate ventilation or shading in non-air-conditioned rooms.

For data on solar heat gain through glass blocks see table 12 in the solar radiation section of this Guide—chapter 15. This table is for standard pattern glass blocks.

PC Glass Blocks Aid Air-Conditioning

The three chief aims of air-conditioning—temperature control, humidity control and cleansing of air are all aided by the use of PC Glass Blocks. Heat loss is less in winter—heat gain is less in summer. Ideal humidity conditions are much more easily maintained without undue condensation. Solar heat transmission and radiation are reduced. Dirt can't filter in, for each panel is a tightly sealed unit.

Sizes and Shapes Available



PC Glass Blocks are available in ten attractive patterns, some of the patterns being designed for special control and direction of transmitted daylight. For complete information on the sizes and shapes of PC Glass Blocks, and for illustrations of the many patterns available, write the Pittsburgh Corning Corporation, Pittsburgh, Pa., or call the nearest Pittsburgh Plate Glass Company warehouse.

• • •

Additional technical data, including detailed figures on thermal insulation, solar heat gain, surface condensation, light transmission and construction data, will be furnished on request.

Alfol Insulation Company

Incorporated

155 East 44th St., New York 17, N. Y.

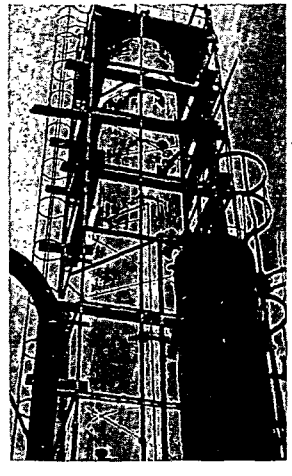
Agents in Principal Cities

HEAT INSULATION for ALL PURPOSES

ALFOL

ALFOL PRE-FABRICATED INSULATION PANELS FOR TANKS, TOWERS, AND ALL TYPES OF HEATED EQUIPMENT

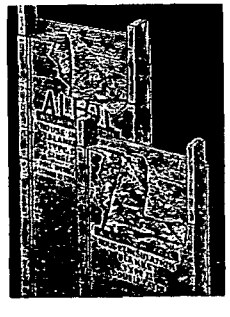
Prefabricated Panel. At Right—Applied to Tower

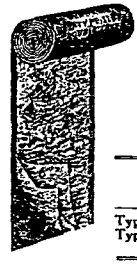


- **Metal Jacketed Panels** containing insulation best suited for each particular condition.
- **Removable and Replaceable** by means of Lock-Joint construction.
- **Shop Fabricated** with Cut-outs for manholes and pipe connections.
- **Easily and Rapidly Applied** by any type of labor.
- **Trim Appearance** with minimum of up-keep.

FOR MORE DETAILED INFORMATION WRITE FOR ALFOL PANEL DATA BOOK

ALFOL BUILDING INSULATION BLANKET





Pure Aluminum Foil spaced on three-ply thick paper vapor barrier sheet. Single and Double Layers insure spaced sheet to reduce conduction and convection. Applied between structural members or furring, Alfol Blankets give high insulation value at low cost.

Specifications

Description	Widths	Net Area per Roll	Net Weight per Roll
Type I.—1 Layer ALFOL	16"-24"	250 sq. ft.	17 lbs.
Type II.—2 Layers ALFOL	16"-20"-24"	250 sq. ft.	19 lbs.

See technical data in Table 1, Section C, Page 116, this volume.

ALFOL RADIATOR REFLECTORS

ALFOL REFLECTORS behind radiators reduce heat loss through walls, save fuel. Temperature gradient to outside reduced 50 per cent.

THIRTEEN YEARS' SERVICE PROVES LASTING VALUE OF ALFOL

American Flange & Manufacturing Co. Inc.

30 Rockefeller Plaza, New York 20, N. Y.

Circle 7-3741

Ferro-Therm

Reg. U. S. Pat. Off.

STEEL INSULATION

Ferro-Therm Steel Insulation, made from rigid steel sheets with a special alloy coating, reflects 95 per cent of all radiated heat. This high reflectivity, combined with extremely low heat storage capacity, provides maximum insulating efficiency in a minimum overall thickness.

Saves Pay Space and Weight

In building construction, the general practice is to install one sheet of Ferro-Therm between roof rafters or over ceiling joists and between studs of exterior walls. In cold storage construction, the number of sheets of Ferro-Therm depends on the temperature to be maintained and the U value required. The k value of Ferro-Therm, based on tests, is listed in the Data Book of the *American Society of Refrigerating Engineers* as 0.226 Btu per (hr) (sq ft) (°F temperature difference). Laboratory tests and thousands of applications have demonstrated that a wall of Ferro-Therm will provide insulating efficiency equivalent to a wall of mass insulation approximately twice as thick.

Assures Rapid Pull Down of Temperature

The low heat storage capacity of Ferro-Therm is extremely important in achieving rapid pull down of temperature, and in saving refrigeration costs for the initial and each subsequent cooling of space. Specifically, the heat storage capacity of a single sheet of No. 38 gauge is 0.029 Btu per (hr) (sq ft) (°F temperature difference). This is approximately 1/16 of the heat storage capacity of 1 sq ft of 1 in. thickness corkboard.

Permanent, Fire-proof Insulation

Ferro-Therm construction eliminates trapping of moisture condensate, with subsequent deterioration of the construction. As it is all-metal, Ferro-Therm cannot be penetrated by rodents, vermin or termites, and is absolutely non-combustible. The value of Ferro-Therm for fire protection is apparent.

125° Below Zero Maintained in Altitude Test Chambers

Ferro-Therm has proved its superiority in buildings, cold storage rooms, refrigerated cabinets, locker rooms, dry ice containers, refrigerated railway car construction, ovens, high-temperature storage tanks—in fact, practically every type of application where high insulating efficiency with economies in space and weight are a requisite. The most notable demonstration of Ferro-Therm performance has been its selection for the insulation of altitude chambers for the testing of Army and Navy aviation equipment and personnel. In these chambers, temperatures as low as −125° F were maintained, with a temperature drop of +70° F to −100° F in 10 to 12 min.

Our catalog, giving data and installation details on Ferro-Therm, will be sent upon request.

Thinner Wall Construction for Lower Temperature Requirements

Thickness of Ferro-Therm Construction	3 sheets 1 in.	4 sheets 1½ in.	5 sheets 2 in.	6 sheets 2½ in.	7 sheets 3 in.	10 sheets 4½ in.	12 sheets 5½ in.	14 sheets 6½ in.
U Factor.	.13	.097	.077	.064	.055	.04	.03	.025

The chart above shows the wall thickness and the number of sheets of Ferro-Therm for various U factors. For any particular temperature requirement the installation with the correct number of Ferro-Therm sheets will require only about 50 per cent of the wall thickness required by equivalent mass insulation, resulting in a proportionate saving in storage space, and a great saving in weight.

Armstrong Cork Company

Building Materials Division

Lancaster Pennsylvania

Offices

Albany	Chicago	Houston	Memphis	Providence
Atlanta	Cincinnati	Indianapolis	Milwaukee	Richmond
Baltimore	Cleveland	Jacksonville	Minneapolis	Rochester
Birmingham	Columbus	Kansas City	New Orleans	St. Louis
Boston	Dallas	Los Angeles	New York	San Francisco
Buffalo	Detroit	Louisville	Philadelphia	Tulsa
Charlotte	Hartford		Pittsburgh	Washington, D.C.

In Canada

Montreal, Que. Toronto, Ont. Winnipeg, Man.

Distributors

Charleston 23, W. Va.	Capital City Supply Co.
Denver, Colo.	Stearns-Roger Manufacturing Co.
Eau Claire, Wis.	Horel-George Co.
El Paso, Texas	Neff-Stiles Co.
Fort Wayne, Ind.	Asbestos & Asphalt Products Co.
Glendale, N. Y.	Charles M. Trojahn
Grand Rapids, Mich.	Tony Batenburg Insulation Co.
Joplin, Mo.	Joplin Cement Co.
Little Rock, Ark.	Fischer Cement & Roofing Co.
Manitowoc, Wis.	Northwestern Asbestos and Cork Insulation Co.
Portland 4, Ore.	Asbestos Supply Co. of Oregon
San Antonio, Tex.	General Supply Co., Inc.
Seattle 4, Wash.	Asbestos Supply Co. of Seattle
South Bend 23, Ind.	Asbestos & Asphalt Products Co.
Spokane 8, Wash.	Asbestos Supply Co. of Spokane
Springfield, Mass.	Johnson Asbestos Co.
Springfield, Mo.	Southwestern Insulation Co.
Tacoma 2, Wash.	Asbestos Supply Co. of Tacoma
Terre Haute, Ind.	The Hartmann Co.

For detailed technical information, samples, and descriptive literature, ask any office or distributor. Specifications appear in Sweet's Catalogs for Architects and for Engineers and Contractors.

PRODUCTS—Armstrong's Corkboard, Cork Covering, Mineral Wool Board—Foamglas,* Heat Insulations, Armstrong's Insulating Refractories, Vibracork, Cushiontone,** Temlok,** Insulation Sundries.**

Corkboard

The thermal conductivity of Armstrong's Corkboard is 0.27 Btu per hour, per degree temperature difference, per inch thickness at 60°F mean temperature. It is furnished in rigid boards 12 in. x 36 in., 18 in. x 36 in., 24 in. x 36 in. and 36 in. x 36 in., in 1 in., 1½ in., 2 in., 3 in., 4 in., and 6 in. thicknesses.

Armstrong's Corkboard conforms in all details to Federal Specification HH-C-561b, July 26, 1941

Cork Covering

Armstrong's Cork Covering is made of pure cork in sizes to fit all standard pipe sizes. The inside surfaces of each piece are machined to assure an accurate fit, free from moisture-catching air pockets. Cork covering is rigid and will not sag. Thicknesses are: Ice Water (1.20 in. to 1.93 in.); Brine (1.70 in. to 3.00 in.); and Special Thick Brine (2.63 in. to 4.00 in.).

Armstrong's Fitting Covers are rigid and are designed to fit accurately all types of standard ammonia and extra heavy fittings, screwed, flanged, and welded.

Mineral Wool Board

Armstrong's Mineral Wool Board equals or exceeds Federal Specification HH-M-371a for board or block form insulation; has low thermal conductivity; is moisture resistant, odorless; is easily handled and erected; possesses structural strength. Standard size 12 in. x 36 in.; thicknesses 1 in., 1½ in., 2 in., 3 in., 4 in.

Foamglas

Foamglas has a closed cellular structure which will not permit passage of air or moisture. It is efficient, moistureproof, fireproof, and offers effective, lasting insulation. This new type of insulation is made in standard 12 in. x 18 in. blocks; thicknesses 2 in., 3 in., 4 in., 4½ in., 6 in. It may be used to insulate refrigerated storage rooms and equipment.

Heat Insulation

The Armstrong Cork Co. distributes and offers contract service on a complete line of high temperature insulations. Included are: 85 per cent magnesia blocks and pipe covering; high temperature blocks and pipe covering; air cell blocks, sheets, and pipe covering; wool felt; hair felt; etc.

Engineering and Contract Service

All Armstrong offices and distributors maintain skilled erection crews. For aid in the solution of any technical problems involving insulation, isolation, or acoustical treatment, and for literature and prices, get in touch with an Armstrong district office or distributor or the Armstrong Cork Co., Building Materials Division, Lancaster, Pa.

*Reg. U. S. Patent Office. Product Mfg. by Pittsburgh Corning Corp.
**Reg. U. S. Patent Office.

Baldwin-Hill Co.

549 Klagg Avenue, **Trenton 2, N. J.**

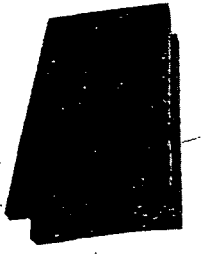
Plants in

TRENTON, N. J. KALAMAZOO, MICH. HUNTINGTON, IND.

B-H No. 1 INSULATING CEMENT

A plastic insulation produced from high-temperature-resisting, nodulated B-H *black* Rockwool combined with high-grade, long-fibre asbestos and colloidal clay. Effective up to 1800 F; reclaimable up to 1200 F. Suitable for insulating large or small irregular surfaces, including those not suited to molded types of insulation. Contains a special *rust inhibitor* which prevents corrosion taking place between insulated surface and cement. Makes a secure bond on either hot or cold surfaces. Instantly adhesive, easy and economical to apply. When mixed with water to trowelable consistency, the nodules of B-H *black* Rockwool retain their physical properties; when dried out, the dead air cells in these nodules provide maximum insulating efficiency. Packed in strong 50-lb bags; can be stocked without breakage or loss.

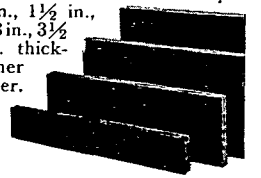
B-H KOLDBOARD



Effective from −150 to 300 F. Made from 100 per cent chemically stable B-H Rockwool fibres, felted and bonded together to form flat, semi-rigid blocks which do not disintegrate or break down structurally under severe service conditions. Koldboard will not support combustion nor smolder when flame is applied. Low moisture absorption: 0.68 per cent at relative humidity of 65 per cent at temperature of 75 F. Thermal conductivity: 0.29 Btu/sq ft/hr/F at a mean temperature of 70 F. Standard sizes: 18 in. and 36 in. long; 12 in., 18 in., 24 in. wide; ½ in. to 4 in. thick; packed in high test fibre cartons.

B-H MONO-BLOCK

A one-block insulation effective over the full temperature range up to 1700 F. Fabricated of high-temperature-resisting B-H *black* Rockwool, felted by a special patented process. A strong, light-weight block, easily cut and fitted on irregular or flat surfaces. Low alkalinity factor insures stability under severe temperature conditions. Density—approx. 20 lb per cu ft. Standard sizes: 18 in. and 36 in. long; 6 in., 12 in. and 18 in. wide; 1 in., 1½ in., 2 in., 2½ in., 3 in., 3½ in. and 4 in. thicknesses. Other sizes to order. Packed in high-test cardboard cartons.

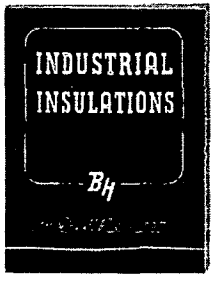


B-H DUCT SOUNDLINER

A rigid material felted of chemically stable Rockwool fibres. Provides an effective means of minimizing sound transmitted through the ducts, by installing it inside the ducts. Does not disintegrate nor support combustion. Easily cut and fitted around bends in the duct—attached with either bolts or specially prepared acoustical cement. Standard sizes: 22 in. x 33 in. and 24 in. x 36 in.; both sizes in ½ in. or 1 in. thicknesses.

CATALOG ON REQUEST

We invite you to write for fully illustrated catalog giving more detailed specifications on these and other B-H Industrial Insulation Products of many types.



The Celotex Corporation

General Offices

120 South LaSalle Street, **Chicago 3**

CELOTEX

REG. U. S. PAT. OFF.

Celotex Cane Fibre Insulation products are made by felting the long, tough fibres of bagasse into strong, rigid boards. They are manufactured under the Ferox Process (patented) which effectively protects them from destruction by termites, fungus growth, and dry rot. They are integrally water-proofed which insures a non-hygroscopic insulation of low capillarity and enduring insulating efficiency.

Celotex Insulating Sheathing

An insulating, weather-resisting sheathing for use under any type of exterior. Surfaces and edges are moisture-proofed with a surface impregnation of asphalt.

Sizes: $^{25}/_{32}$ in. thick: 4 ft wide: 8 ft, 9 ft, 10 ft and 12 ft long.

Big Board—4 ft wide x 8, 9, 10 and 12 ft long x $^{25}/_{32}$ in. thick.

Center Matched—Available in the same thickness, in 2 ft x 8 ft T & G units for horizontal application.

Celotex Insulating Lath

Regular Insulating Lath—A cane fibre plaster base of high insulating efficiency. Surface provides a strong bond for plaster and the bevelled edges and shiplap joint provide additional reinforcement.

Size: 18 in. x 48 in.; thickness: ½ in.

Celotex Roof Insulation

Regular Roof Insulation—A cane fibre product possessing superior insulating properties. It reduces roof heat transmission as shown by coefficients established in THE GUIDE; reduces roof movement due to contraction and expansion.

Size: 23 in. x 47 in.; thickness: ½ in., 1 in., 1½ in. and 2 in.

Vapor-seal Roof Insulation—Same as above except coated on all edges and surfaces with waterproof asphalt and made with an offset on all bottom edges to provide a network of channels which equalize air pressure to reduce roof blisters and buckling.

Size: 23 in. x 47 in.; thickness: 1 in., 1½ in. and 2 in.

Cemesto

A completely fabricated fire and moisture resistant structural insulating wall unit. Consists of a Celotex cane fibre core surfaced on both sides with a ⅛ in. layer of asbestos-cement board. The established low thermal conductivity of the Celotex core is maintained in the manufacture of Cemesto.

Sizes: 4 ft x 4 ft, 4 ft x 6 ft, 4 ft x 8 ft, 4 ft x 10 ft, 4 ft x 12 ft; thicknesses: 1⅛ in., $1^{9}/_{16}$ in. and 2 in.

Celo-Siding

A weather-resistant, insulating, structural siding. Replaces wood or other sheathing materials and provides the exterior finish as well. Made of a Celotex cane fibre core that has been asphalt coated on all sides and edges. The weather side is additionally coated with a high grade asphalt into which mineral granules are firmly embedded.

Sizes: Thickness, ⅝ in. and ⅞ in.; ⅝ in.—4 ft x 8 ft (square edges only); ⅞ in.—2 ft x 8 ft tongue and groove (long edges only), and 4 ft x 8 ft, 4 ft x 10 ft (square edges only).

Celo-Block

A cane fibre cold storage insulation. Made from ½ in. low density Celotex Cane Fibre Boards laminated with waterproof asphalt mastic. The finished block is surfaced front and back with vapor proofing asphalt. Its efficient low thermal conductivity and protection against moisture makes Celo-Block the ideal answer for all low temperature requirements.

Size: 18 in. x 36 in.; thickness: 2 in.

Celotex Rock Wool Products

Available in the following forms—Loose, Granulated, and Paper-backed Batts. Celotex Rock Wool is made from the clean fibres of molten rock. It is incombustible and integrally waterproofed.

Q-T Ductliner

An acoustical material designed especially for duct lining in air conditioning systems. Absorbs duct noises. Made of rock wool and a special binder. Designed to withstand air duct humidity conditions. Is fire resistant and will not smoulder or support combustion. Thermal conductivity of 0.30.

Cork Insulation Company, Inc.

155 East 44th Street, **New York, N. Y.**

Factory: WILMINGTON, DELAWARE

Branches

BOSTON, MASS.
CHICAGO, ILL.
LOS ANGELES, CALIF.
PHILADELPHIA, PA.
SAN FRANCISCO, CALIF.
*MILWAUKEE, WIS.
SEATTLE, WASH.
ST. LOUIS, MO.
WASHINGTON, D. C.
PORTLAND, ORE.

*Represented by CORINCO INSULATION CO., INC.

CORINCO PRODUCTS

Insulation Corkboard
Cork Pipe Covering
Cork Tile Flooring
Acousticator
Cork Lagging and Discs
Marine Corkboard
Isolation Corkboard

INSULATION

Proper and efficient insulation for industrial or domestic purposes demands a material with a low conductivity factor. Cold storage equipment is vitally dependent on the efficiency of the insulating material used. Corinco Corkboard and Corinco Cork Pipe Covering afford positive protection from refrigeration loss for Freezers, Cold Rooms, and for Cold Pipe Lines.

CORINCO CORKBOARD

Corinco Corkboard is manufactured in sheets 12 x 36 in. and 24 x 36 in., in thicknesses of 1, 1½, 2, 3, and 4 in. Its low conductivity factor, and its high resistance to moisture and fire make it an ideal insulation for cold storage and other low temperature installations.

CORINCO CORK PIPE COVERING

Corinco Cork Pipe Covering and Fittings are available in three standard thicknesses —Ice Water, Brine, and Special Thick Brine. They are made to fit snugly around the pipe and fittings, thus precluding the possibility of moisture and frost forming within the covering. This assures long life and efficient operation.

For the insulation of large pipe lines and cylindrical tanks, Corinco Cork Lagging is used.

AIR CONDITIONING

With the rapid increase in the use of Air Conditioning equipment, it has become more and more apparent that a proper insulating medium is necessary for the ducts, pumps, and other cold equipment. Corinco Corkboard is ideally suited for efficient and economic covering of this equipment, greatly reducing heat loss and preventing the formation of condensation.

CORINCO ISOLATION CORKBOARD

For effective control of machinery noises and vibrations, Corinco Isolation Corkboard has proved successful. It is available in boards 12 in. wide by 36 in. long by 1, 1½, 2, 3, or 4 in. thick. To meet specific requirements the boards are manufactured in several densities classified according to the weight per board foot.

CORINCO ACOUSTICATOR

The adoption of noise control is becoming more widespread with the growing realization that the workers' efficiency is greatly increased in quiet surroundings. Theatres, auditoriums, and broadcasting studios, as well as industrial plants, have turned to Corinco Acousticator for definite noise control.

For other information and engineering services, write to Cork Insulation Co., Inc., 155 East 44th Street, New York City.

Carey PRODUCTS

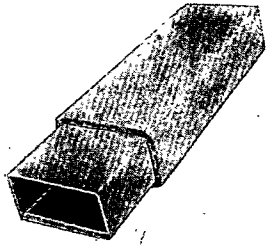
The Philip Carey Mfg. Company

Lockland, Cincinnati 15, Ohio

Branches In All Principal Cities

CAREYDUCT is recommended wherever quietness, ease of installation, fire safety, fume resistance and good appearance are desirable or essential. Used in air conditioning systems widely before the war, Careyduct has proven itself on some of the largest governmental industrial and commercial installations in the country.

Write for engineering performance and installation data.



ACOUSTICAL. Careyduct is a natural sound absorber and non-conductor of sound. Quiets fan noise; won't pick up and "telegraph" other outside noises.

INSULATED. High-efficiency insulation assures delivery of hot or cold conditioned air to outlets with minimum change in temperature.

AIRTIGHT. Won't "breathe" or vibrate at high velocities. Slipjoint construction prevents leakage.

SAVES SPACE. Being 40% to 50% quieter than ordinary duct, Careyduct handles higher velocities, permitting the use of smaller sized ducts.

EASY TO INSTALL. Prefabricated Careyduct units are easy to install—particularly in tight places. Simple low cost fittings can be made in the shop or on the job.

FIREPROOF. Being 100% asbestos construction Careyduct won't smoulder or burn.

GOOD LOOKING. Surfaces are smooth and free from unsightly raised seams or joints. No stiffeners or braces. Blends well with modern interiors.

5 TYPES OF CAREYDUCT

Insulated and Acoustical Type. 100% asbestos construction—combines duct and insulation.

Key-lock Type. For high temperature applications up to 500 F. Impervious to water.

Single-wall Type. For heating and ventilating systems.

Reinforced Corner Type. Fabricated of Carey Firefoil or insulated sheathing. An ideal duct for large industrial installations.

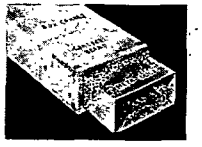
Asbestos-cement Type. Made of asbestos-cement wallboard in different thicknesses. Sizes: 23½ in. and up.

The Philip Carey Mfg. Company

Lockland, Cincinnati 15, Ohio

Branches In All Principal Cities

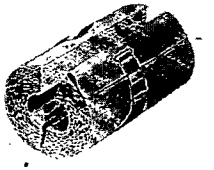
Carey PRODUCTS



CAREYCEL FOR AIR DUCTS

Uses: A fireproof, low cost, high efficiency asbestos board for insulating ducts and all types of air conditioning equipment.

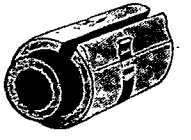
Description: Composed of 12 to 14 laminations of indented (not corrugated) asbestos felt per inch of thickness. Weight: approximately 1½ lb per board foot. Sheet Size: 36 in. x 36 in., or cut to order. Blocks: 6 in. x 36 in. Thickness: ½ in. up.



CAREYCEL FOR HEATING SYSTEMS

Uses: Pipe coverings and blocks for pipes, boilers, ovens and other apparatus where the temperature doesn't exceed 300 F.

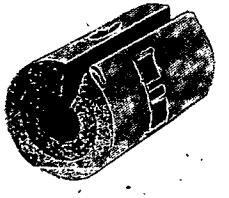
Description: Pipe covering sections 36 in. long by 1 in. thick, finished with cotton duck jacket and bands. Blocks: 6 in. x 36 in. Sheets: 36 in. x 36 in., or cut to order. Thickness: ½ in. up.



CAREY IMPERVO FOR COLD PIPES

Uses: A high efficiency insulation for cold or ice water pipes—keeps the water cold and prevents sweating.

Description: Laminated insulating felt with waterproof liner and jacket. 36 in. long in ½ in., ¾ in., double ½ in. and double ¾ in. thick sections, finished with cotton duck jackets and bands.



CAREY PROTECTO TO PREVENT FREEZING

Uses: Designed especially to reduce the danger of freezing of exposed water pipes.

Description: Consists of an inner layer of hair felt, a waterproof felt liner and an outer layer of insulating felt. For severe conditions use two thicknesses. 36 in. long sections with cotton duck jacket and bands. One thickness only—approximately 1 in.

The Eagle-Picher Company

General Offices: American Building, **Cincinnati 1, Ohio**

Offices in Principal Cities

A Remarkable Insulating Wool Made From Minerals

Years ago Eagle-Picher pioneered a method of fusing and fiberizing carefully selected minerals into a dark gray insulating wool. This mineral wool is chemically inert. Fibers are mechanically strong, extremely resilient and flexible. They withstand expansion and contraction without loss of efficiency even at elevated temperatures.

From this mineral wool, Eagle-Picher has fabricated a long list of insulating products to meet a wide range of temperatures and operating requirements.

H-2 Loose Wool

A clean fill insulation that is highly efficient for temperatures to 1200° F. Averages considerably lighter in weight than many rock and slag wools—goes farther. Fibers are soft and flexible. Approved by Underwriters Laboratories as fireproof and a non-conductor of electricity. Retains physical and chemical stability in presence of water. Packed in 40-lb. bags.

7-B Granulated Wool

Another grade of fill insulation that has all the advantageous properties of Eagle H-2 Loose Wool. It consists of small pellets averaging ⅛ to ½ in. in size. For all fill jobs in irregular spaces. May be poured. Packed in 40-lb. bags.

Low Temperature Felt

A highly efficient insulating material for subzero and low temperatures (to 400° F). Available in densities 6-lb to 8-lb per cu ft. Recommended for refrigerator rooms, trucks, refrigerators, stoves, etc. Sheds water. Extensively used in marine field.

Paper Encased Batts and Blankets

These light-weight, sturdily constructed batts and blankets are easy to apply. Enclosed on four sides with paper, one side of which is an approved vapor barrier. Strong tacking flanges. Quickly cut with knife or shears. Three thicknesses—Ful-Thik, Semi-Thik and 1-in. For home use.

Super "66" Cement

A high-temperature insulating cement. Easy to apply and trowels to a smooth finish. Actively inhibits rust. Will stick on any clean, heated surface. Dry coverage 50-55 sq ft per 100 lb. 100 per cent reclaimable up to 1200° F. Packed in 50-lb bags.

Supertemp Blocks

An all-purpose high-temperature block insulation which will withstand elevated temperatures up to 1700° F without loss of efficiency or structural strength. Fibers are water-repellent. Light weight. Easily cut to fit irregularly shaped surfaces. Blocks withstand all normal vibration and abrasion encountered in use for which they are recommended. All standard sizes.

Insulseal

A protective coating for Industrial Insulation Blankets, Supertemp, "66" Cement and other kinds of heat insulation. Provides a permanent seal that safeguards insulation against air infiltration, moisture, water, fumes; also against vibration and abrasion. Does not support combustion.

For more complete specifications and technical data on these and other Eagle Insulating Products, see Sweet's Engineering or Power Plant catalogs.

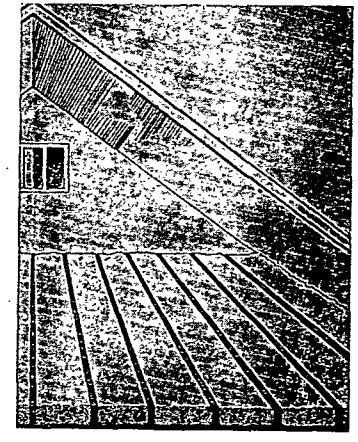
Insul-Wool Insulation Corp.

General Offices, **Wichita, Kansas**

Branches in Principal Cities

Manufacturers and Distributors of Insul-Wool

Insul-Wool is a fibre insulation of the "fill" type—made of wood pulp, a natural insulating material. By the exclusive "Insul-Wool" method the wood pulp is converted into a loose fluffy substance which, when installed in a building, forms a soft heat-resisting blanket having millions of tiny air cells capable of resisting passage of either heat or cold.



Insul-Wool Applied over Ceiling

UNIFORMITY OF PRODUCT

Only one grade of Insul-Wool is made and every "run" is tested at the factory to insure uniformity of product and unvarying high quality. It is free from grit, silicon particles, or "shot."

RESISTANT TO FIRE AND VERMIN

A special "Insul-Wool" method of chemical treatment makes Insul-Wool thoroughly and permanently vermin and fire resistant.

"INSUL-WOOL" SERVICE

Insul-Wool is distributed and installed only by specially trained men—direct factory representatives or men in the organizations of the largest insulation material dealers throughout the United States.

ADVANTAGES OF INSUL-WOOL

1. It is made from wood pulp, a natural insulating material.
2. Chemical treatment makes Insul-Wool safe under all conditions and hazards.
3. Its light weight ⅝ lb per sq ft 4 in. thick adds very little load to the ceiling rafters.
4. A permanent type isulation which does not pack or settle and outlasts the building in which it is installed.
5. Does not attract moisture.
6. Cuts fuel costs up to 50 per cent and reduces Summer temperatures up to 18 F in the home.
7. Meets U. S. Government requirements of Federal Construction with a thermal conductivity of 0.24 Btu per hour, per square foot, per degree Fahrenheit, per inch thickness.

Analysis of Insul-Wool in Terms of Commercial Thickness

Material	Commercial Form	Comm'l. Thickness Inches	D. Wt. Per Cu. Ft.	C. Conductivity
INSUL-WOOL	Wood Fiber-Loose Type, permanently treated for vermin and fire.	1 4	2.5	0.24* 0.067**

*Kansas City Testing Laboratory, Inc., February 25, 1938.
**J. C. Peebles, Armour Institute of Technology, April 8, 1937.

Complete data on Insul-Wool Insulating Product will be sent upon request.

INSULITE

General Offices
500 Baker Arcade Bldg., **Minneapolis 2, Minnesota**

THIRTY-TWO YEARS PROVEN DURABILITY

For 32 years engineers and architects have specified Insulite materials for structural uses, interior finish, low temperature duct lining, and for other thermal insulation and sound control work. Insulite materials have proved themselves practical through their performance on the job.

STRUCTURAL MATERIALS

Lok-Joint Lath—An insulating plaster base, fabricated from Ins-Lite or from Graylite. Patented "Lok" firmly locks the sheets between supporting members. Thickness: ½ in. Size: 18 x 48 in.

Sealed Graylite Lok-Joint Lath—An insulating plaster base of Graylite, sealed on stud space side with an effective *moisture vapor barrier*. Has patented "Lok" on long edges. Same thickness and size as Ins-Lite and Graylite Lok-Joint Lath.

Bildrite Sheathing is an asphalt-containing wood fiber insulating board manufactured under an exclusive process which provides increased strength and moisture resistance. It is $^{25}/_{32}$ in. thick and has a gray-brown color. Thermal conductivity maximum: 0.36 Btu per inch thickness. Each sheet is marked to indicate proper nail spacing. Available in sizes 4 x 8 ft up to 4 x 12 ft with all edges square. Also available in 2 x 8 ft size with interlocking joint on long edges. Used as a structural sheathing board and as roof boarding.

Condensation Control—Where low outside temperatures and high inside humidities may occur, authorities recommend "sealing the warm side and venting the cold side" of the wall to prevent condensation. An adequate vapor barrier, Sealed Graylite Lok-Joint Lath, should be used on the warm (room) side of the wall thereby effectively reducing vapor transmission into the stud space. Bildrite Sheathing is designed to allow any surplus vapor in the stud space to "breathe" or be vented to the exterior air. If vapor is trapped within the stud space and cannot escape through the sheathing, destructive condensation may occur.

THE APPROVED INSULITE WALL OF PROTECTION

This construction consists of Bildrite Sheathing on the exterior of the frame work and Sealed Graylite Lok-Joint Lath on the interior. Transmission coefficient (U) is shown below.

	Interior Finish
Exterior Finish and Sheathing	No Insulation Between Studding
	Plaster (½ in.) on Sealed Graylite Lok-Joint Lath (½ in.)
Wood Siding, $^{25}/_{32}$ in. Bildrite Sheathing	0.15 Btu/sq ft./hr/ °F

The above value is typical of results which can be obtained by utilizing Insulite materials in frame construction. For further (U) values refer to Chapter 6 pages 127 and 128.

Applying Bildrite Sheathing

Applying Lok-Joint Lath

INTERIOR FINISH MATERIALS

Ins-Lite Building Board—A wood fiber board with the light color of natural wood—burlap and linen textured surfaces. Thermal conductivity: Nominal 0.33 Btu/hr/sq ft/ in./F; density: 16 lb/cu ft. Furnished in thicknesses of ½ and ¾ inch and sizes of 4 x 7 ft to 4 x 12 ft. Also available in 6 x 8 ft, 6 x 12 ft and 8 x 12 ft sizes.

Graylite Building Board—An integrally treated asphalt containing wood fiber board of grayish brown color—burlap and linen textured surfaces. Thermal conductivity nominal 0.35 Btu per inch thickness. Furnished in same thicknesses and sizes as Ins-Lite Building Board.

Smoothcote Interior Board—Factory Coated Insulating Board with smooth, finished surface one side, having 68 per cent light reflection. Furnished in ½ inch thickness only and in sizes of 4 x 7 ft to 4 x 12 ft.

Satincote Interior Board—Factory finished Insulating Board in color light ivory, oyster white, coral and green. Light reflection from 64 per cent for green to 80 per cent for the light ivory color. Requires no further decoration. Resistant to abrasion and washable. In ½ inch thickness and in sizes of 4 x 7 ft to 4 x 12 ft.

TileBoard—Available in Smoothcote and Satincote. TileBoard is furnished with the Lok-Grip Joint that permits concealed nailing and which together with the Lok-Pin (a flat diamond shaped metal dowel) definitely and mechanically safeguards against any falling units even though no face nailing is used.

Smoothcote and Satincote TileBoard available in ½ inch thickness and sizes of 12 x 12 inches to 16 x 32 inches.

Plank—Available in Smoothcote and Satincote. Plank has the Lok-Grip joint which permits concealed nailing and is beveled and beaded both long edges. Smoothcote and Satincote Plank furnished in ½ inch thickness, widths of 8 to 16 inches and lengths of 8 to 12 ft.

Acoustilite—A high efficiency acoustical material for sound control. Coefficient of sound absorption, at 512 cycles, is 0.79 when mounted on solid background and 0.80 when on furring strips. Noise reduction coefficient is 0.65 when mounted on solid background and 0.75 when on furring strips. Factory painted in buff, (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ¾ in.; sizes, 12x12 in. to 16x32 in.

Fiberlite—An efficient sound absorptive and decorative material. Coefficient of sound absorption, at 512 cycles, is 0.53 when mounted on a solid background and 0.72 when on furring strips. Noise reduction coefficient is 0.55 when mounted on solid background and 0.65 when on furring strips. Factory painted in buff (light reflection 77 per cent) and in white (light reflection 80 per cent). Units have a butt joint and are beveled on four edges. Thickness, ½ in.; sizes, 12x12 in. to 16x32 in.

Acoustilite or Fiberlite effectively quiet and control sound

HardBoard Products

HardBoard materials are tough, durable, grainless, pressed wood fiber boards with a hard, smooth surface. Available in a range of densities from 55 to 68 lb/cu ft. Thicknesses are from $^{1}/_{10}$ to $^{5}/_{16}$ in. and sizes of 4 x 2 ft to 4 x 12 ft.

Industrial Insulation

Industrial Insulation is a wood fiber board for use in all types of manufacturing industries producing items such as refrigerators, coolers, showcases, brooders, partitions and cabinets.

It can be cut-to-size and fabricated to customer's specifications. Three types of industrial board are available.

Lowdensite Industrial Board—A 10 to 14 lb density board with an average tensile strength of 100 lb/sq in. and an average conductivity of 0.30 Btu/hour /sq ft/F/inch thickness.

Ins-Lite Industrial Board—A 14 to 18 lb density board with an average tensile strength of 250 lb/sq in. and an average conductivity of 0.33 Btu/hour/sq ft/F /inch thickness.

Graylite Industrial Board—Differs from two above products in that it has an integral asphalt treatment which provides increased strength and moisture resistance as well as minimum thickness and linear expansion. A 16 to 20 lb density board with an average tensile strength of 350 lb/sq in. and an average conductivity of 0.35 Btu/hour/sq ft/F/inch thickness.

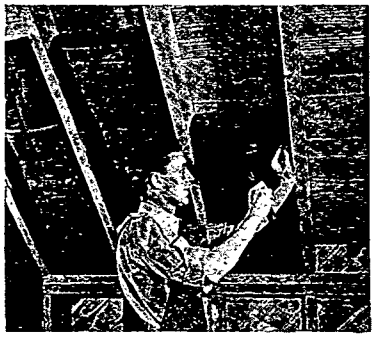
Johns-Manville

Executive Offices: 22 East 40th Street, New York 16, N. Y.

Offices in All Large Cities

Johns-Manville Home Insulation

Johns-Manville Rock Wool Home Insulation is a light, fluffy mineral wool, highly efficient in heat-proofing practically any building, old or new. It is durable, rot-proof, fire-proof and odorless, and will not corrode or settle. Full stud thickness of this material will cut fuel costs up to 30 per cent in winter and help keep rooms up to 15 deg cooler in hottest weather. J-M Rock Wool Home Insulation is furnished in two forms: for new construction,



Applying J-M Super-Felt Type B batts in new home

in easily handled batts; for existing buildings, in nodulated form to be installed pneumatically.

For New Construction
J-M Super-Felt Type B Batts

Super-Felt Type B Home Insulation is furnished in pre-fabricated batts of uniform thickness and density, in both full stud thickness and semi-thick, in sizes 15 x 23 in. and 15 x 48 in., designed to fill completely the space between studs, joists and rafters on the usual 16 in. centers. The sturdy felted "wool" is strong enough to be handled rapidly without damage. The batts are backed with waterproof, vapor-resistant paper, extending on both the long sides in 1½ in. wide flanges, by which the batt is fastened in place and which also aid in sealing the joints. This backing protects against penetration of moisture from wet plaster and also resists infiltration of moisture vapor from the house into the wall.

As a further protection against moisture, the felted wool is also waterproofed.

Super-Felt may also be obtained in blanket form, in Thick, Medium and 1 in. thicknesses. The blankets have a waterproof vapor barrier paper on one side and a permeable kraft paper on the opposite side, cemented together along the long edges to form a strong nailing flange.

For Existing Homes and Buildings
Type A "Blown" Rock Wool

Type A Rock Wool is blown pneumatically into the spaces between studs in outer walls and between rafters or joists in roofs or attic floors. Insulation thickness in walls corresponds to stud depth, approximately 3⅝ in.; the density, approximately 5 to 8 lb per cu ft, assures maximum thermal efficiency. This type of insulation is installed only by Johns-Manville or by Approved J-M Home Insulation Contractors, who are equipped with the necessary apparatus and trained crews.

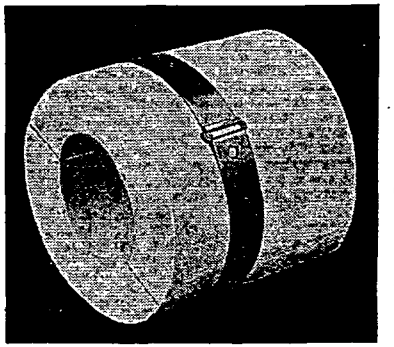
Write for Details

Complete information on all types of J-M Rock Wool Home Insulation will be furnished on request.

J-M Airacoustic Sheets for lining Air-Conditioning Ducts

J-M Airacoustic Sheets, for duct linings of air conditioning systems, are flame-proof, highly sound-absorbent and moisture-resistant, with a surface which will not materially increase friction losses in the duct system. Write for Bulletin AC-23A.

Johns-Manville Pipe and Boiler Insulation



J-M 85% Magnesia Pipe Insulation

J-M Pre-Shrunk Asbestocel Pipe Insulation

Cellular type of insulation for pipes carrying low pressure steam or hot water.

Made up of alternate layers of plain and corrugated, specially-treated, moisture-resistant, asbestos paper. Sections surfaced with asbestos paper for high speed work or with a canvas cover.

Furnished in 3-foot sections, in thicknesses of 2 to 8 plies, each ply approximately ¼ inch thick.*

J-M 85% Magnesia

Recommended as the most widely used insulation of the molded type for temperatures up to 600 F. Pipe insulation is furnished in sectional or segmental form for all commercial pipe sizes,* in thicknesses up to 3 in. Blocks are 3 in. by 18 in. and 6 in. by 36 in., flat or curved, from 1 in. to 4 in. thick. Minimum thickness for curved blocks, 1¼ in.

J-M Pre-Shrunk Wool Felt Pipe Insulation

Due to its Dual-Service Liner—an asphalt-saturated felt—J-M Pre-Shrunk Wool Felt is equally effective and durable on either hot or cold water service piping. By the use of moisture-resistant felts, shrinkage troubles have been minimized.

Supplied in the regular canvas finish, it is furnished in 3-ft sections in thicknesses of ½ in., ¾ in., 1 in., Double ½ in., and Double ¾ in., for all commercial pipe sizes.*

J-M Asbesto-Sponge Felted Pipe Insulation

Recommended on all high pressure steam piping at temperatures up to 700 F where insulation may be subjected to rough usage or where maximum efficiency and durability are desired. Furnished in 3-ft sections up to 3 in. thick, for all commercial pipe sizes.*

J-M Superex Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 F. Superex and Magnesia are both furnished in sectional and segmental pipe covering, and in block forms.

J-M Asbestocel Sheets and Blocks

Asbestocel Sheets and Blocks are used for insulating warm-air ducts, flues, heater casings and fan housings in the ventilating systems. Temperature limit 300 F. Furnished 18 and 36 in. wide by 36 in. long, from ¼ in. to 4 in. thick.

J-M Rock Cork Sheets and Pipe Insulation

J-M Rock Cork is made of mineral wool and a moisture-proof binding ingredient molded into sheets for insulating refrigerated rooms, air conditioning equipment and other low temperature requirements; and into sectional pipe insulation with an integral waterproof jacket, for all low temperature service. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance, its high insulating efficiency is maintained in service.

Furnished in sheets 18 in. by 36 in., in 1, 1½, 2, 3 and 4 in. thicknesses; also 18 in. by 18 in. by 1 in. thick. In lagging form, for curved surfaces, supplied 18 in. long by 1½, 2, 3 and 4 in. thick, 2 to 6 in. wide, depending on diameter. In pipe covering form, in ice water, brine and heavy brine thicknesses, for all commercial pipe sizes.*

Details on Request

Write for complete information on any Johns-Manville insulating material.

*Can also be supplied in sections to fit straight runs of copper pipe or tubing with outside diameter ⅝ in. and larger.

Kimberly-Clark Corporation

Neenah, Wisconsin

KIMSUL INSULATION

. . . to increase efficiency in Heating, Ventilating, Air Conditioning

A PRODUCT OF Kimberly Clark RESEARCH

THERMALLY EFFICIENT!

KIMSUL* blanket insulation in an attic will usually make a house up to 15 degrees cooler in summer, comfortably warm in winter. Fuel cost is often cut 30 per cent by KIMSUL. Thermal efficiency: 0.27 Btu/hr/sq ft/deg F/in. (J. C. Peebles, Armour Institute).

RESISTS FIRE, MOISTURE, VERMIN, INSECTS, FUNGUS!

PERMANENT!

Strong stitching prevents shifting, sagging, settling—Hence, no *transoms* through which heat can leak. Made of wood-fiber, impregnated with asphalt, KIMSUL lasts indefinitely.

ECONOMICAL!

Because it is compact (stretches to 5 times packaged length), KIMSUL is less expensive to store, ship, handle. Installation cost is small also, because it is exceptionally easy to apply . . can be shaped swiftly around obstacles, as it is flexible as a towel.

3 THICKNESSES—4 STANDARD WIDTHS!

KIMSUL blanket comes in Commercial Thick (nominally ½ in.), Standard Thick (nominally 1 in.) and Double Thick (nominally 2 in.) . . . giving you choice of thicknesses to fit specific needs. Each thickness is available in four standard widths: 16 in., 20 in., 24 in., and 48 in.

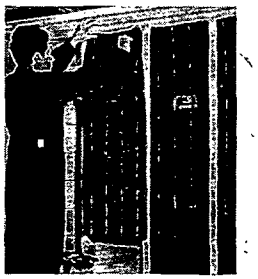
Consult Our Engineers

Heating and ventilating engineers, architects, contractors and builders are invited to discuss their insulation problems with our engineers. These insulation experts are equipped to render a valuable advisory service . . . and to give accurate information regarding the application of KIMSUL to all types of construction.

Send for free KIMSUL book giving full technical data

*Kimsul (trade mark) means Kimberly-Clark insulation.

To cut the compressed KIMSUL blanket to length required, simply count off trade marks—placed approximately 24 in. apart when blanket is expanded—allow about 6 in. for end fastening, and cut.



To install KIMSUL in walls, partially expand blanket and attach to under-side of top plate with fiber strip provided. Expand blanket to base of wall and attach similarly. Then staple KIMSUL to sides of studs.

Ordinarily a difficult job, insulation of sloping roofs is usually a one man job with KIMSUL. Simply fasten blanket to sides of rafters and collar beams.

HOW KIMSUL INCREASES WINTER COMFORT BY RAISING WALL AND CEILING SURFACE TEMPERATURES

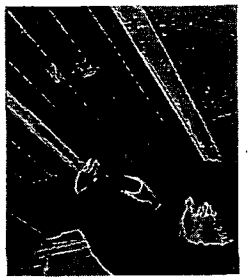
		Wall Surface Temperature, Deg	Ceiling Surface Temperature, Deg
No Insulation		60.0	52.0
KIMSUL in Walls and Ceilings	Commercial Thick	64.4	60.0
	Standard Thick	65.3	63.0
	Double Thick	66.5	65.4
	Commercial Thick in Walls Standard Thick in Ceilings	64.4	63.0
	Commercial Thick in Walls Double Thick in Ceilings	64.4	65.4
	Standard Thick in Walls Double Thick in Ceilings	65.3	65.4
KIMSUL in Ceilings only	Commercial Thick	60.0	60.0
	Standard Thick	60.0	63.0
	Double Thick	60.0	65.4

HOW KIMSUL LOWERS FUEL BILLS

		Tons of Coal per Season	Tons Saved per Season	Per Cent of Fuel Saved
No Insulation		12.1	....	....
KIMSUL in Walls and Ceilings	Commercial Thick	8.0	4.1	33.9
	Standard Thick	7.1	5.0	41.3
	Double Thick	6.4	5.7	47.1
	Commercial Thick in Walls Standard Thick in Ceilings	7.5	4.6	38.0
	Commercial Thick in Walls Double Thick in Ceilings	7.0	5.1	42.1
	Standard Thick in Walls Double Thick in Ceilings	6.7	5.4	44.6
KIMSUL in Ceilings only	Commercial Thick	9.1	3.0	24.8
	Standard Thick	8.5	3.6	29.7
	Double Thick	8.1	4.0	33.1

Calculations apply to a one-story home in the Chicago area.

As shown in the table, almost any desired performance can be produced by insulating with KIMSUL. The *economic thickness* provides the best balance between insulation costs and resultant fuel savings.



KIMSUL installs easily in flat roofs and ceilings. After cutting blanket to proper length, expand it and fasten one end to top wall plate. Pull across, and fasten other end to opposite wall plate. Fasten edges if there is no supporting finishing material.

To install KIMSUL in attic floor, place blanket directly on plaster base, with paper side down. Fasten one end, expand blanket, and fasten other end. That's all!

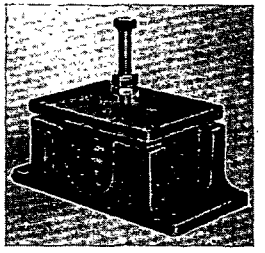
Left-overs and odd pieces of KIMSUL are ideal for caulking spaces around window frames and doorways. Stops drafts and wasteful heat leaks.

The Korfund Company, Inc.

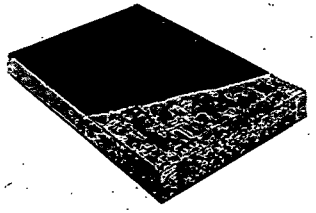
48-50 32nd Place

Long Island City 1, N. Y.

Representatives in KORFUND VIBRATION CONTROL **Principal Cities**



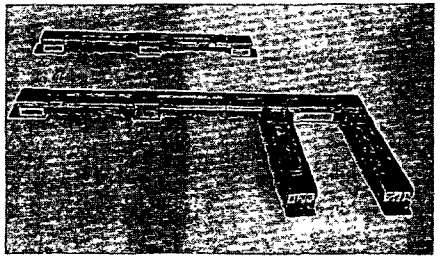
TYPE LK UNIVERSAL VIBRO-ISOLATORS are recommended for isolation of compressor and condensing units where almost perfect absorption of noise and vibration is required, as for instance on upper floors of office buildings, hospitals and hotels. These units absorb vibration in all directions and provide for both horizontal and vertical adjustment. Loading range 75-12,000 lb. Catalog LK-550.



KORFUND CORK PLATES provide a high degree of noise and vibration isolation when applied to compressors mounted on a slab of concrete. Since Cork Plates should be loaded to a minimum of 1,500 lb per sq ft, it is seldom necessary to cover the entire area of concrete. Instead, a number of Cork Plates of dimensions 8 in. x 8 in., or smaller, may be spaced under the concrete so that the load will be evenly distributed over the several plates. Bulletins Bc-13; Bc-22.

The Korfund Co., Inc., is exclusive national distributor of ARMSTRONG STANDARD DENSITY VIBRACORK in 2 in., 3 in., and 4 in. thicknesses, loading capacity 1500 to 4000 lb per sq ft.

The TWIN RAIL FAN BASE is a development in fan and blower isolation which incorporates many new and exclusive features. Interchangeable prefabricated parts are used throughout. It is light in weight, yet has great rigidity to vertical and lateral stresses. The Twin Rail Base is easily assembled and installed, and is fully adjustable. Rubber-in-Shear, Cork, or Steel Spring Isolators are supplied, depending upon requirements. Catalog TR-800.



Lockport Cotton Batting Co.

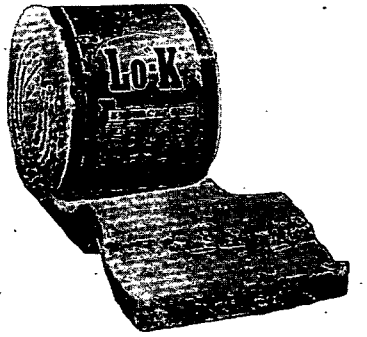
Lockport, New York

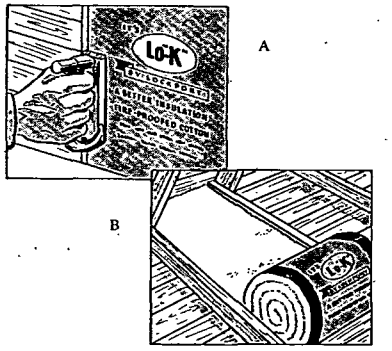
Lo-"K"

FLAME-PROOFED

COTTON INSULATION

Fire proofed and manufactured under Department of Agriculture Specifications





Lo-"K" is made of processed American cotton of varying thicknesses, packaged in rolled form, *with* or *without* Kraft paper backing—an added moisture barrier.

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 7/8 lb. (*See table.*)

Flame-Proofed—To comply with Department of Agriculture specifications to withstand a temperature of 1800° F.

Moisture-Resistant—Chemical treatment, combined with natural protective coating on cotton fibres, enables cotton to effectively resist moisture. Prevents rot and mildew.

Smooth Texture—Cotton contains none of the sharp particles that can harm workmen's hands during process of installation.

Flexible—Cotton batt may be expanded or contracted to fit any enclosure. It easily passes over and under pipes and wires.

In walls or roof, the nailing flange is folded out from paper backing and attached to edge of studs or rafters. See illustration "A" above.

In attic with open floors, Lo-"K" is simply tucked between ceiling rafters, cotton side up. See illustration "B" above.

Manufactured in Standard Sizes.

Packaged in Rolled Form.

Thicknesses—inches: 1, 1½, 2, 3, 3⅝. Width—inches: 16, 24 on centers. Lengths: Standard from 12 ft up.

Available in Quantity—Lockport is establishing dealers in principal distributing centers. If your dealer doesn't happen to have Lo-"K" in stock, we invite your inquiry for prices and samples.

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	[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
Rigid fibre board made from shredded wool and cement	24.20	0.46

[a] Compiled from Chapter 6.
[b] "*k*" indicates temperature conductivity.

Mundet Cork Corporation

65 S. Eleventh St. INSULATION DIVISION **Brooklyn 11, N. Y.**

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork, Natural Cork Isolation Mats, and all kinds and varieties of Cork Specialties.

Authorized contractors for high temperature insulation.

Mundet Branches

Atlanta, Ga.	Chicago 21, Ill.	Houston 1, Tex.	New Orleans 16, La.
Brooklyn, N. Y.	Cincinnati 2, Ohio	Jacksonville 7, Fla.	Philadelphia 39, Pa.
Boston (No. Cambridge) 40, Mass.	Dallas 1, Tex.	Kansas City 7, Mo.	St. Louis 4, Mo.
	Detroit 26, Mich.	Los Angeles 31, Calif.	San Francisco 7, Calif.

Mundet Distributors are Located in the Following Cities—Names and Addresses on Request

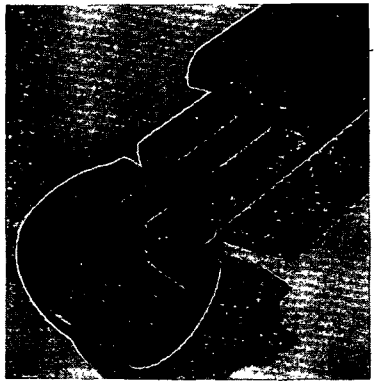
Amana, Ia.	Hartford, Conn.	Norfolk, Va.	Salt Lake City, Utah
Baltimore, Md.	Johnson City, Tenn.	Oklahoma City, Okla.	Seattle, Wash.
Buffalo, N. Y.	Memphis, Tenn.	Portland, Oregon	Tucson, Ariz.
Denver, Colo.	Minneapolis, Minn.	Richmond, Va.	Tulsa, Okla.
El Paso, Tex.	Nashville, Tenn.	Rochester, N. Y.	Utica, N. Y.

Mundet "Jointite" Corkboard

—for all low temperature insulation and for acoustical correction. 100 per cent pure cork, fabricated in accordance with U. S. Government Master Specifications and unsurpassed in its field. Sold in standard 12 in. x 36 in. sheet. Standard thicknesses, ½ in., 1 in., 1½ in., 2 in., 3 in., 4 in., 6 in.

Mundet "Jointite" Cork Pipe Covering

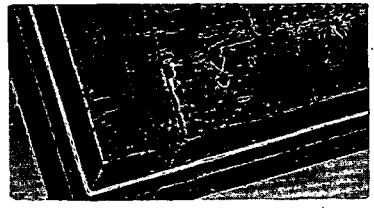
Shown below, with fitting cover. Protects all types of low temperature lines. Made in 3 thicknesses, with complete line of standard covers, suitable for pipes carrying sub-zero to 50 F temperature.



Section of Mundet Moulded Cork Pipe Covering, with Fitting. The pipe covering is made in sections 36 in. long, to fit all sizes of pipes.

Mundet Cork Vibration Isolation

Machinery vibration encountered in heating and ventilating work is effectively controlled by the use of Mundet Natural Cork Isolation Mats. These consist of blocks of pure cork, held together within a rigid steel frame or bound with asphalt paper applied with hot asphalt top and bottom. Mundet steel bound mats are usually used under exposed mounts; asphalt paper bound mats under concrete foundations of the envelope type. Mats are made to fit under any type of machine foundation. For loads exceeding 2000 lb per square foot, we manufacture Mundet Machinery Isolation Cork, which is a board form of compressed granulated cork, available in 3 densities. All types of isolation are furnished in 1 in., 1½ in., 2 in., 3 in., 4 in., and 6 in. thicknesses, depending on class of service.



Above close-up of Mundet Natural Cork Isolation Mat shows how the blocks of cork are held together within a rigid steel frame.

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers, to assist and advise in the preparation of specifications pertaining to cork. This service is available without obligation to any one who has a low temperature insulation or a vibration isolation problem. Our latest catalogue will be sent on request. It is replete with information and data of value to every specification writer whose field touches our products.

Mundet Contract Service

Covers the complete installation of our products, in accordance with best established practice. Divided responsibility is avoided. Materials and workmanship are guaranteed.

The Pacific Lumber Company

PALCO WOOL INSULATION

100 Bush Street San Francisco	35 E. Wacker Drive Chicago	5225 Wilshire Blvd. Los Angeles	122 East 42nd St. New York

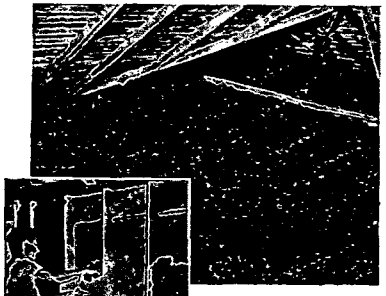
HOUSE INSULATION

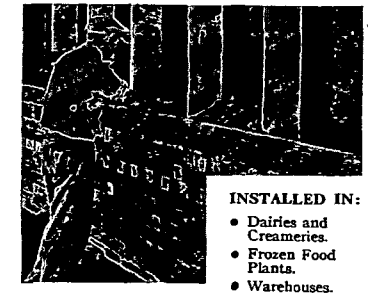
INSTALLED IN CEILINGS AND WALLS

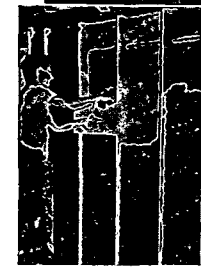
PALCO INSULATION WOOL

TRADE MARK REG U. S. PAT. OFFICE

COLD STORAGE INSULATION







PALCO WOOL INSULATION

Provides maximum protection against the intrusion of heat or cold. Readily installed in upper ceilings of uninsulated homes.

Can be installed by blower or hand-pack methods.

—

FLAME PROOF SAFERIZED

INSTALLED IN:

- Dairies and Creameries.
- Frozen Food Plants.
- Warehouses.
- Refrigerators.
- Pre-cooling Plants
- Ice Plants.
- Fur Vaults.
- Fruit and Produce Storage, etc.

EIGHT POINTS OF PALCO WOOL SUPERIORITY

1. Thermal Efficiency: The established conductivity of PALCO WOOL is .26 Btu per hour per sq ft per inch of thickness per degree F difference in temperature by the Flat Plate Method.

2. Non-Settling: The fibres of PALCO WOOL possess such resilience that no settlement in a wall can occur under the most severe conditions of vibration.

3. Moisture Resistant: The fibres of PALCO WOOL are entirely lacking in capillarity and have little attraction for moisture, enabling it to remain dry and efficient when in use.

4. Permanent: The inherent antiseptic qualities of PALCO WOOL make the existence of fungus impossible. The fibres retain their resilience indefinitely.

5. Vermin Repellent: PALCO WOOL is distasteful and repellent to rodents and insects.

6. Fire Resistant: PALCO WOOL, like the Redwood bark it comes from, is inherently fire resistant. As an additional protection it is *Saferized* to make it flameproof.

7. Odor Proof: PALCO WOOL is odorless itself and does not absorb or give off odors.

8. Economical: PALCO WOOL is light in weight and low in density, offering exceptional thermal efficiency per dollar invested.

WRITE FOR INSULATION MANUALS

- House Insulation Manual.
- Cold Storage Manual.
- Frozen Food Locker Plant Manual.
- How to Build a Plant Manual.

Get your copies today.

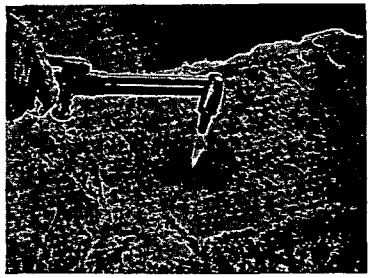
Reynolds Metals Company

Reynolds Metals Building

Richmond, Virginia

NEW YORK CHICAGO BOSTON MINNEAPOLIS SAN FRANCISCO

REYNOLDS PROCESS COTTON INSULATION



The Fire Test: Blowtorch Flame on Reynolds Cotton Insulation Will Not Cause Combustion

Reynolds Process Cotton Insulation stops up to 73 per cent of heat flow and permits complete air circulation around framing. Thermal conductivity is 0.24 Btu per hour, per square foot, per 1°F, per inch thickness. (Authority—Prof. J. C. Peebles, Armour Institute of Technology).

Reynolds Process Cotton Insulation is one of the most efficient barriers to the passage of heat that is commercially available today. It consists of heat retarding dead air cells. There are myriads of minute and hollow cellulose fibres, entwined and interlocked into a flexible, clean, resilient and light-weight mass.

Reynolds Process Cotton Insulation is not affected by heat or vibration. It will not settle, sag or pack. Age does not impair its lifetime efficiency.

AIR CIRCULATES FREELY

Reynolds Process Cotton Insulation permits free circulation of air on both sides of the insulation, thus allowing rapid evaporation of any moisture which may occur. Possible damp-rot, decay or other damage to structural materials is thereby minimized.

INSECT AND VERMIN REPELLENT

Reynolds Process Cotton Insulation insures utmost cleanliness, not only during installation, but during the lifetime of the structure. This blanket-type insulation does not harbor vermin or other insects. It is odorless, and will not decay.

MOISTURE RESISTANT AND FLAME RETARDENT

Reynolds Process Cotton Insulation will not absorb water or moisture. It has successfully withstood flame tests up to 1800°F.

Reynolds Process Cotton Insulation is one of the most effective sound absorption materials.

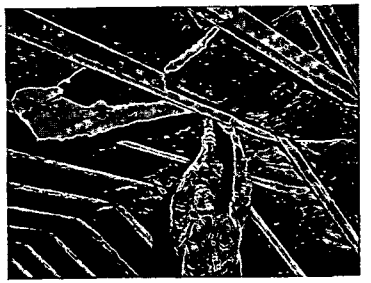
APPROVED AND ACCEPTED

Reynolds Process Cotton Insulation is manufactured under constant United States Government inspection and in strict accordance with Department of Agriculture specifications. It is approved for home and industrial insulating purposes by Federal, State and Municipal bureaus, builders, architects, and heating engineers throughout the United States.

Reynolds Process Cotton Insulation is ideally suited for equipment insulation. It can be furnished cut to size and in special widths up to 60 in.

COSTS LITTLE TO INSTALL

Furnished in convenient blankets, or rolls, Reynolds Process Cotton Insulation is adaptable to all constructions without expensive cutting or waste. For existing homes, as well as for walls, ceilings or roofs of new structures, it provides maximum insulating efficiency. Labor costs for installation are exceptionally low.



Reynolds Process Cotton Insulation is Easily Installed

Send for literature describing its many advantages in existing houses as well as in new construction.

Reynolds Metals Company

Reynolds Metals Building

Richmond, Virginia

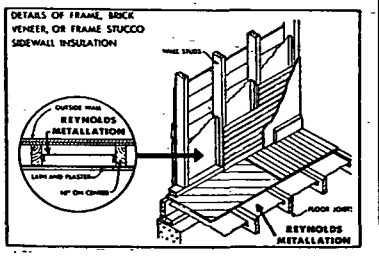
NEW YORK CHICAGO BOSTON MINNEAPOLIS SAN FRANCISCO

REYNOLDS METALLATION

REFLECTIVE ALUMINUM FOIL

Reynolds Metallation is produced by mounting bright aluminum foil to one or both sides of 80-lb tough Kraft paper—or on heavy Sisalkraft when greater tensile strength is desired.

Reynolds Metallation serves as a permanently efficient vapor barrier and reflector of radiant heat. It appreciably reduces heat loss in air-filled areas it faces or divides. It has extremely low heat-storage capacity.



HOW IT INSULATES

Heat is transferred by radiation, convection and conduction. Up to 85 per cent of all heat transmitted through air spaces is by radiation. Reynolds Metallation reflects 95 per cent of the radiant heat imposed upon it without regard to direction of heat flow. This high reflectivity and resultant low emissivity permits the passage of not more than 5 per cent of radiant heat into adjoining air spaces. The above is accomplished through use of Reynolds Metallation, installed to divide the normal space in a hollow wall into two separate reflective air spaces.

CHARACTERISTICS

Reynolds Metallation is so light in weight it places no strain on structural members to which it is affixed. It is flexible yet sufficiently rigid to conform to angles or curves in a structure when not under tension.

Reynolds Metallation is soundproof, vermin and water proof; dust will not impair its insulating value.

The aluminum surface of Reynolds Metallation retains its reflectivity indefinitely under all normal conditions. Recent tests by Prof. Gordon B. Wilkes, Massachusetts Institute of Technology, proved that Reynolds Metallation suffers only negligible loss of its reflective insulating quality after years of exposure.

Table below gives values for the rate of moisture transmission through the more common building materials encountered.

The values given are expressed in terms of Btu/Hr/100 sq ft/Grains per lb difference.

Moisture Transmission Values	Latent Heat Btu/Hr/100 Sq Ft/Gr/Lb Difference
Double aluminum foil on paper	0.1
50-lb asphalt sheathing paper	0.45
Coated surface aluminum foil (single)	1.0
2 coats asphalt paint	1.1
2 coats aluminum paint	1.2
Duplex building papers	1.3
Wood per inch thickness	4.0
Concrete—per inch thickness	5.5
Common brick—per inch thickness	6.0
Cinder concrete—per inch thickness	9.0
Plaster—(⅜ in.)	23.0
Fiber Board—(½ in.)	38.0
Mineral Wool—per inch thickness	40.0

Table prepared by Carrier Corporation, Syracuse, N. Y.

EASY TO INSTALL

Reynolds Metallation is recommended for all types of building construction involving hollow walls; for use between floors and in roof sections. It is available in rolls of convenient lengths and correct widths for standard stud and rafter spacings.

Reynolds Metallation is easily and quickly installed, thereby adding a low installation cost to a low purchase price and assuring users of maximum efficiency with a minimum investment.

Send for literature describing REYNOLDS *REFLECTIVE ALUMINUM* METALLATION.

The RUBEROID Co.

INSULATING PRODUCTS

500 Fifth Avenue, **New York 18, N. Y.**

307 N. Michigan Ave., **Chicago 1, Ill.**

Divisional Offices

NEW YORK CHICAGO BOSTON (Millis) ERIE BALTIMORE MINNEAPOLIS MOBILE

Today, Ruberoid materials for heating and power equipment are safeguarding insulation efficiency in hundreds of plants, factories and buildings—giving maximum results with minimum cost.

Ruberoid products are of proved merit, high efficiency and of a type to meet every need economically. They include pipe coverings and blocks for temperatures from 350 F to 1900 F; Woolfelt pipe covering for hot or cold water conduits; asbestos papers for wrapping furnace pipes, protecting air conditioning; high temperature cements, millboard, rollboard, rock wool bats and blankets.

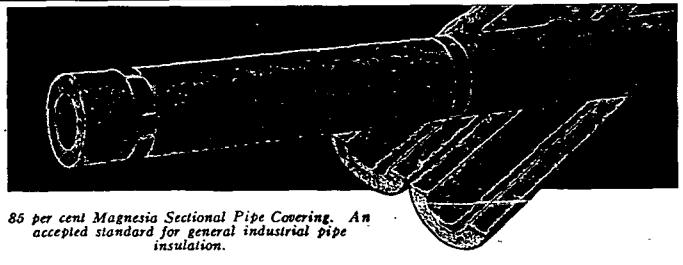
A complete Insulation Guide, will be gladly forwarded upon request.

Product	Temp. Limit	Suggested Use
High Temp.	to 1900 F	Protective inner layer for high temperature insulations.
Calsilite	to 1250 F	In pipe covering and block form for high pressure steam.
Sponge felt	to 700 F	For vibrating pipes and underground insulation—excellent efficiency.
85 per cent Magnesia	to 600 F	Combines efficiency and reasonable cost—General use in industrial work.
Imperial	to 600 F	Rugged, efficient—wide range of applications.
Air Cell	to 300 F	Standard insulation for residential pipes.
Woolfelt	to 200 F	For cold and hot water lines. Recommended especially for air conditioning work.
Anti-Sweat	to 120 F	For cold water lines to prevent condensation.
Frost-proof	30 F to 100 F	To assist in the prevention of freezing in circulating water pipes exposed to cold.

Above products are also made in sheets and blocks for insulating tanks, breechings, furnaces, etc.

Characteristics and Insulating Values of Ruberoid Products

Product	Temperature Limit	Density	Modulus of Rupture	Shrinkage		Thermot Conductivity "K"	
	Of soaking Heat	Lbs. Cu. Ft.	Lbs. Sq. In.	Temp. F	Per Cent	Mean Temp. F	Btu
High Temp.	1900 F	24	70	1800	2.0	200° 400° 600°	0.594 0.630 0.666
Calsilite	1250 F	12	50	1000	0.5	100° 200° 400° 500°	0.330 0.392 0.510 0.570
85 per cent Magnesia	600 F	16	50	500	1.0	100° 200° 300° 400°	0.410 0.477 0.500 0.605
Sponge Felt	750 F	30	85	500	1.0	200° 300° 400° 500°	0.470 0.501 0.531 0.562
Imperial	600 F	20	73	500	1.0	100° 200° 300° 400°	0.405 0.455 0.500 0.550



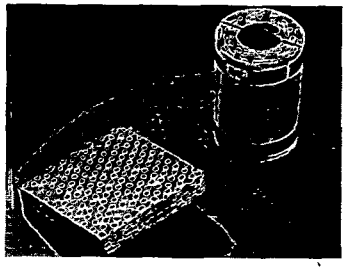
85 per cent Magnesia Sectional Pipe Covering. An accepted standard for general industrial pipe insulation.

Ruberoid Calsilite

Calsilite is a molded insulation that provides the physical characteristics required by engineers designing modern high pressure steam equipment. It is a combination of raw materials, ingeniously controlled, to form a product free from active, harmful ingredients. It is not only unique in its extremely light weight for such material, but it will stand soaking heat of 1250 degrees Fahrenheit indefinitely without changing its usefulness to any appreciable degree. This permits its use on pipe lines and equipment operating at high temperature without the usual protective inner layer of less efficient insulation. The use of single layer insulation also speeds up its application.

Calsilite is highly efficient in heat saving; it has a high modulus of rupture; it resists abrasion excellently. There is little breakage of Calsilite in shipment or when handled on the job. It has low moisture absorption when subject to high humidity atmospheres.

Calsilite is furnished in both pipe covering and block form. Pipe Covering is made in sections 36 in. long, canvas covered and in such thicknesses that combinations of layers can provide whatever thickness may be required to cope with the conditions. Blocks are made in standard sizes of 6 in. x 36 in. in thickness up to 3½ in. All are packed in cartons suitable for easy handling on the job.



Imperial Pipe Covering

This is a laminated asbestos paper insulation that has been indented to use 22 laminations of asbestos paper per inch thickness. Its efficiency makes it satisfactory for most medium pressure steam work in industrial plants. Its construction makes it ideal for vibrating conditions. It is recommended for temperatures to 600 F. Being an asbestos felt laminated material it is used on vibrating pipes or where hard service is expected. Will withstand water conditions for underground piping. Excellent as an industrial, oil refinery and synthetic rubber plant insulation.

Ruberoid Insulating Cements

For the finishing of sheet and block insulation and the insulating of irregular surfaces, such as valves, unions, flanges, etc., the Ruberoid line of insulating cements is complete. This group of cements not only uses asbestos as its base, but also takes advantage of such excellent natural products as magnesia, mineral wool and Vermiculite.

Asbestos Cements—Factory Prepared—Grades AA, A, HF.

Asbestos Cements—Mine Run—Grades 115, 214.

Magnesia Cement—85 per cent Magnesia.

High Temperature Cement—Grade H.T.

Mineral Wool Cement—Good Insulation in plastic form for temperatures to 1500 F.

Vermiculite Cement—Grade A-11.

Ruberoid Asbestos Insulating Papers and Millboard

Asbestos Paper

Made of pure asbestos, fire-resisting. May be obtained in 6, 8, 10, 12, 14, 16 and 32 lb weights. Also thicknesses ⅛ in. and 3/32 in. known as asbestos rollboard.

Asbestos Corrugated Paper

Efficient for insulating warm air pipes and ducts. 36 in. wide. Rolls contain 250 sq ft.

Asbestos Millboard

A rigid board of exceptional strength and whiteness. Cuts and drills easily. Withstands temperatures to 1000 F. Sheets 42 x 48 in.

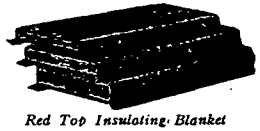
United States Gypsum Company

General Offices: 300 W. Adams Street, **Chicago, Ill.**

INSULATION PRODUCTS

Blanket **Decorative** **Structural**

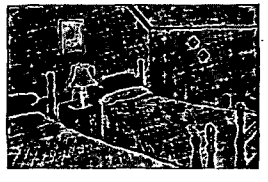
Blanket



Red Top Insulating Blanket
A Fiberglas Product

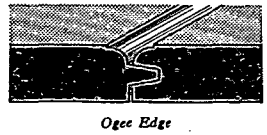
RED TOP INSULATING BLANKETS—Made for standard framing in three thicknesses: one inch, medium and thick, in rolls of 125, 75 and 50 sq ft respectively. Also available in bats 3 ft long in same thicknesses. Light-weight *RED TOP INSULATING WOOL blanket is enclosed with an asphalt-type vapor barrier front side and a vapor permeable paper on the back side. This is to resist accumulation of moisture within blanket.

Decorative



Decorative

WEATHERWOOD* PLANK—Manufactured in widths of 8, 10, 12 and 16 inches and in lengths 6, 8, 10 and 12 feet—½ inch thick. The "Ogee" tongue and groove on the long edges (see cut) conceals nails and seals against dust and air infiltration. Weatherwood Plank is made in Blendtex (gray and tan blends) and Hilite (ivory) colors. When combined in variations of shades and width, Weatherwood Plank produces maximum values in both insulation and decoration.



Ogee Edge

*Red Top Registered Trademark.

WEATHERWOOD TILE—Available in 12 x 12, 12 x 24, 16 x 16 and 16 x 32 inches in ½ and 1 inch thicknesses. Colors are Blendtex (gray and tan blends) and Hilite (ivory).

WEATHERWOOD PANELTILE—Hilite color available in 12 x 24, 16 x 32, 12 x 48, 24 x 48, 48 x 48, 48 x 96 and 24 x 96 inches in ¾ inch thickness. Blendtex colors available in 12 x 24, 16 x 32, 12 x 48 and 24 x 48 inches. Tile sizes 12 x 24, 16 x 32 and 24 x 48 inches can be mill cross scored to represent ½ inch Tile dimensions. Dimensions (see above). All tile have "Ogee" tongue and groove edges.

Structural

WEATHERWOOD SHEATHING—Asphalt coated. 2 feet x 8 feet x 25/32 inches thick, with tongue and grooved long edges for horizontal application. Also available in 4 x 8, 4 x 9, 4 x 10 and 4 x 12 feet in either ½ inch or 25/32 inches thickness with square edges for vertical application.

WEATHERWOOD BUILDING BOARD—4 feet wide, made in lengths 6, 7, 8, 9, 10 and 12 feet, ½, ¾ and 1 inch thick in either ivory or gray tan colors. Applied by nailing; effectively insulates, strengthens and decorates.

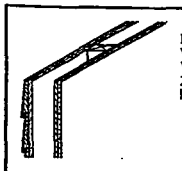
WEATHERWOOD INSULATING LATH—18 x 48 inches x ½ thick with V joint on long edges. Gives an excellent plaster bond and also acts as a cushion for plaster with sound deadening qualities.

ROOF INSULATION—In sheets 22 x 47 inches—½, 1, 1½ and 2 inches thick. All but the ½ inch size are supplied laminated with either square or "ship-lapped" edges.

Heat Loss Factors

The heat loss factors shown on the opposite page indicate the comparative insulation value of various insulating treatments included in common construction systems.

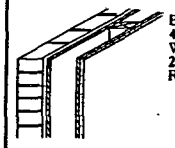
NOTE: These figures apply to 1 story buildings. To get figures for 2 story homes add 20 per cent to the values below for the wall constructions and divide by one-half for floor and ceiling constructions. It is important to use correct factor due to variations in the ratio of wall and window areas.



WALLS

Basic Construction—Frame
Wood Siding
Wood Sheathing
2 x 4's
Rocklath & Plaster

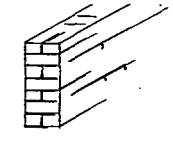
	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	.248	.120	.083	.064
Substituting in Above Basic Const.				
a. 25/32" WW Sheathing	.186	.105	.076	.059
b. ½" WW Plaster Base	.183	.103	.076	.059
c. ¼" WW Plaster Base	.157	.094	.069	.055
d. 1" WW Plaster Base	.143	.088	.067	.054
e. 25/32" WW Sheathing and ½" WW Plaster Base	.147	.090	.068	.054
f. ½" WW Bldg. Bd., Tile or Plank	.187	.104	.077	.059
g. ¾" Bldg. Bd., or Tile	.160	.095	.070	.056
h. Gyplap Sheathing	.310	.135	.090	.068
i. ½" Sheetrock	.260	.123	.085	.064
j. Gyplap Sheathing and ½" Sheetrock	.330	.136	.091	.069



WALLS

Basic Construction—Brick Veneer
4" Brick
Wood Sheathing
2 x 4's
Rocklath & Plaster

	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	.270	.125	.085	.065
Substituting in Above Basic Const.				
a. 25/32" WW Sheathing	.202	.111	.078	.061
b. ½" WW Plaster Base	.200	.107	.077	.060
c. ¼" WW Plaster Base	.178	.102	.073	.058
d. 1" WW Plaster Base	.157	.094	.070	.055
e. 25/32" WW Sheathing and ½" WW Plaster Base	.162	.095	.070	.055
f. ½" WW Bldg. Bd., Tile or Plank	.215	.112	.079	.061
g. ¾" WW Bldg. Bd. or Tile	.187	.104	.075	.059
h. Gyplap Sheathing	.350	.128	.087	.066
i. ½" Sheetrock	.288	.130	.088	.066
j. Gyplap Sheathing and ½" Sheetrock	.368	.142	.093	.069

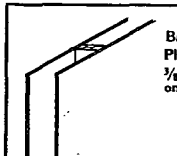


WALLS

Basic Construction
8" Brick Wall—4" Face Brick and 4" Common Brick—No interior finish

	No Wool Between Studs*	Adding Red Top Wool** 1"	2"	3"
Basic Construction	.500	..	..	..
Adding to Above Basic Const.				
a. ½" Plaster	.480			
b. Rocklath and Plaster (Furred)	.300	.143	.093	.069
c. ½" WW Plaster Base and Pl. (Furred)	.220	.121	.084	.064
d. ¾" WW Plaster Base and Pl. (Furred)	.190	.112	.079	.061
e. 1" WW Plaster Base and Pl. (Furred)	.160	.101	.073	.058
f. ½" WW Bldg. Bd., Tile or Plank	.230	.124	.085	.065
g. ¾" WW Bldg. Bd., Tile or Plank	.200	.115	.081	.062
h. 1" WW Bd., Plank or Tile	.170	.104	.075	.059
i. ½" Sheetrock Furred	.320	.146	.095	.070

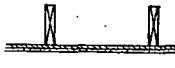
*Based on ⅛" Furring Strip **Based on Full Dimension



WALLS

Basic Construction—Plywood
Plywood on Wood Studs
⅜" Outside—¼" Inside with one Air Space Over ¾"

	No Wool Between Studs	Adding Red Top Wool 1"	2"	3"
Basic Construction	.431	.151	.095	.074
Substituting in Above Basic Const.				
a. ½" WW Bldg. Bd., Tile or Plank	.275	.126	.087	.065
b. ¾" WW Bldg. Bd. or Tile	.230	.115	.081	.062
c. 1" WW Bldg. Bd. or Tile	.196	.106	.076	.060
d. ⅜" Sheetrock	.430	.151	.095	.074
e. ½" Sheetrock	.413	.148	.095	.074
f. Adding to basic construction 25/32" WW Sheathing	.218	.113	.080	.061



CEILINGS

Basic Construction
⅜" Rocklath and ½" Plaster

	No Wool Between Joists	Adding Red Top Wool 1"	2"	3"
Basic Construction	.610	.169	.116	.080
Substituting in Above Basic Const.				
a. ½" WW Plaster Base & Plaster	.329	.136	.091	.068
b. ⅜" WW Plaster Base & Plaster	.290	.128	.087	.066
c. 1" WW Plaster Base & Plaster	.213	.110	.079	.061
d. ½" WW Bldg. Bd., Tile or Plank. No Plaster	.356	.139	.092	.067
e. ¾" WW Bldg. Bd. or Tile	.268	.124	.086	.065
f. ½" WW Bldg. Bd. or Tile	.220	.113	.080	.062
g. ⅜" Sheetrock	.670	.174	.118	.089
h. ½" Sheetrock	.635	.170	.115	.079

FLOORS

Basic Construction
Maple or Oak Flooring on Yellow Pine Sub-Flooring

	No Wool Between Joists	Adding Red Top Wool 1"	2"	3"
Basic Construction	.340	.138	.091	.068
Adding to Above Basic Const.				
a. ½" WW Bd. on bottom of joists	.180	.102	.075	.059
b. ¾" WW Bd. on bottom of joists	.158	.094	.070	.055
c. 1" WW Bd. on bottom of joists	.141	.088	.066	.053

Above calculations based on data from A.S.H.V.E. GUIDE—1942.

Wood Conversion Company

First National Bank Building, **St. Paul 1, Minn.**

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INSULATION FOR EVERY PURPOSE

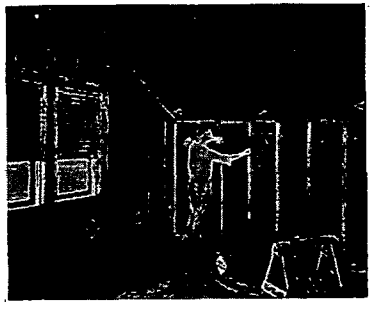
Wood Conversion Company's insulation line includes flexible fibre and rigid insulation for industrial and domestic purpose. Every product in this line is backed by the name and reputation of Weyerhaeuser, assuring high quality and satisfactory service. Following are the products included:

Balsam-Wool Sealed Blanket Insulation—a building insulation that is windproof, moisture-proofed, vermin-proof and fire-resistant. Spacer flanges on the sealed blanket permit easy, positive application—no settling or sagging.

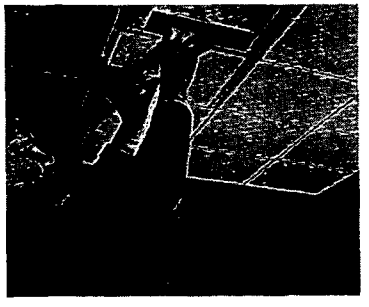
Balsam-Wool K-25 Fibre—an industrial fibre shipped in bales which, when mechanically fluffed and processed, provide an ideal insulation for use in locker plants, cold storage plants, etc. K-25 is low in conductivity—has an amazingly low "K" factor—and retains its efficiency permanently.

Balsam-Wool Fibre Pneumatic System—provides an effective means of insulating refrigerator cabinets and doors pneumatically, under pressure. The processed fibre fills every void—eliminates the need for slabs or filler strips.

Plastic Filler Fibre—Designed to fill special needs of the plastics industry. Provides additional tensile strength. Available in a special group for various needs.



Balsam-Wool Sealed Blanket Insulation



Nu-Wood Interior Finish Tile

NU-WOOD STRUCTURAL INSULATION

Nu-Wood Interior Finish—insulates, decorates and quiets noise. Fade-proof colors. Available in Plank, Tile, Board and Wainscot.

Nu-Wood Sta-Lite—an insulating interior finish with more than 70% light reflection.

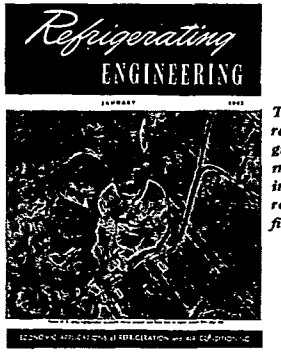
Nu-Wood Insulating Lath—Used as a plaster base, assures strong, smooth, true walls and ceilings, free from cracks.

Nu-Wood Roof Insulation—Fulfills all requirements of Federal Specification LLL-F-321A. Furnished in any practical thickness.

American Society of Refrigerating Engineers

40 West 40th Street, **New York 18, N. Y.**

REFRIGERATING ENGINEERING



The most rapidly growing magazine in the refrigeration field

REFRIGERATING ENGINEERING

LONG acknowledged the most authoritative periodical in the field, *Refrigerating Engineering* has added steadily to the practical value of its contents, and its number of readers has grown in proportion. A wide variety of material is presented, all from the viewpoint of its usefulness to the reader in his own business. This magazine is a must for men who keep in touch with all that is new and important in refrigeration and air conditioning. The annual subscription price is **$3.**

THE REFRIGERATING DATA BOOK

IS an essential tool in the refrigeration and air conditioning industries. It has been published biennially since 1932 and appears in two volumes, published alternately—the basic volume, now in its fifth edition, a standard reference work which deals with refrigeration cycles, fundamental data, industrial, domestic, and commercial systems, and air conditioning; the refrigeration applications volume, consisting wholly of practical how-it-is-done chapters on all the known applications of refrigeration and air conditioning. Either volume sells for **$4** in the U. S.

APPLICATION DATA BULLETINS

SOME 33 bulletins are available separately at reasonable prices for single copies or quantity orders and can also be had bound with a paper cover, the complete set for **$5.50.**

The APPLICATION DATA Bulletins tell precisely how refrigeration is used in various fields, giving examples and specific information on the best practice up to date. Some of the subjects covered to date are: refrigeration of locker plants, of fur storage, of restaurants, of liquids, of apples and pears, humidity in refrigeration, refrigeration service charts, refrigeration for skating rinks, butter and cheese making, milk plants, citrus fruits, beer dispensing, retail stores, wine making, load calculations, operation of ammonia machines, how to figure air conditioning, refrigeration of ships' stores, etc.

CODES AND STANDARDS

THE A.S.R.E. further contributes to refrigeration progress by its participation in establishing codes and standards in the industry. Among the recent codes made available are: No. 21—Testing and Rating Milk Coolers; No. 22—Rating and Testing Water-cooled Refrigerant Condensers (Revised—1942); No. 23—Rating and Testing Refrigerant Compressors (Revised—1942); No. 24—Rating and Testing Water and Brine Coolers (Revised—1944); No. 25—Rating and Testing Forced-circulation Air Coolers for Commercial and Industrial Refrign. (Revised 1944), (Supplement to Cir. No. **13**, not sold separately.)

MEMBERSHIP ACTIVITIES

IT is the policy of the A.S.R.E. to treat in its meetings current subjects touching upon all phases of the art of refrigeration. Membership is in two grades with dues from **$10.00** to **$15.00.** Sections hold meetings in the following cities: Baltimore-Washington, Boston, Central N. Y., Chicago, Cincinnati, Cleveland, Detroit, Erie, Kansas City, Los Angeles, Milwaukee, New Orleans, New York, Northern New Jersey, Pacific Northwest, Philadelphia, Pittsburgh, Richmond, San Francisco, St. Louis, Twin Cities.

To keep apace with progress in refrigeration and air conditioning, read the publications and follow the activities of THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS, 40 West 40th St., New York 18, N. Y.

Coal-Heat

Published at

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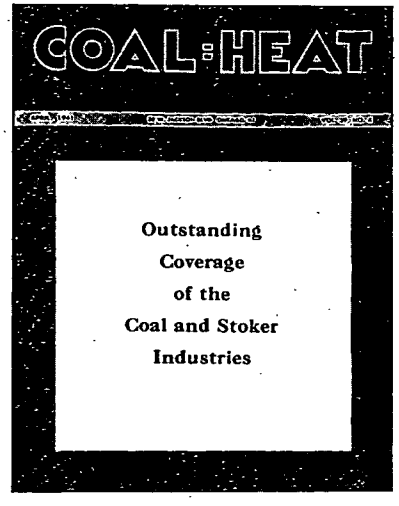

FOR information on the use and sale of stokers, coal and coal heating equipment, you can turn to COAL-HEAT with complete confidence.

Here is a magazine that appeals to every man concerned with the market, use and sale of solid fuel and modern heating equipment. Having long since recognized the importance of properly designed efficiently operated, properly maintained equipment to the successful use of coal, and therefore to the welfare of the coal industry, COAL-HEAT constantly emphasizes the significance of the "equipment factor" in heating merchandising.

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COAL-HEAT has at its disposal an almost unlimited number of sources of authentic information on the topics it covers; its articles are written by the best informed men in the coal, stoker and heating industries. It enjoys quite a following, not only among the most progressive merchants in these industries, but among the industry's leading combustion and heating engineers. For years it has championed the importance of the fuel engineer to the coal and stoker industries, and each year prints many articles *for* and *by* fuel engineers.

COAL-HEAT's fundamental editorial policy is "to further the more satisfactory use and sale of coal and modern coal-burning equipment." It actively supports the application of scientific and engineering knowledge to the use of coal and coal-burning equipment. It covers both the merchandising and utilization of the coal, stokers and modern heating equipment.

With over a million stokers in use today, the importance of COAL-HEAT's field is clearly evident. It has been and is COAL-HEAT's job to supply coal and stoker men with the information they need to insure satisfaction for stoker users. The same is true with hand-fired heating plants and all kinds of household and commercial coal heating equipment.

In addition to providing its readers with a basic and diversified editorial program, COAL-HEAT also publishes a number of books and booklets, manuals and reprints covering a wide range of subjects of interest to coal, stoker and heating men. Its series of heating guides for the consumer have proved particularly popular. These are available at small cost.

Subscription rates—$2.00 a year; $3.00 for two years. Rates apply for both United States and Canada. Foreign rates —$2.00 a year; $4.00 for three years.

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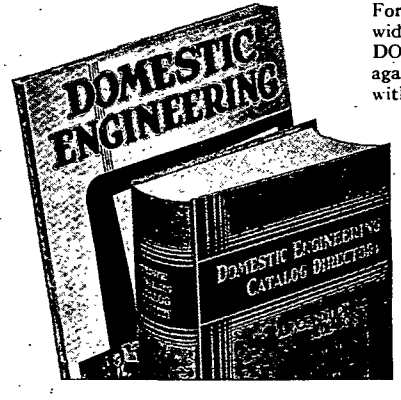

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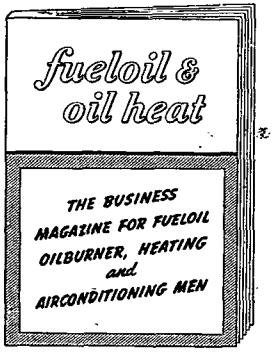

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ROBERT GRAY, *Business Manager*
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OILHEATING is a vertical, integrated industry, selling through the same retail outlets oilburners, oilfired heating and air conditioning equipment, and fueloil.

Starting some 15 years ago, this integration of fuel and the equipment to burn it has grown until in 1945 two-thirds of all oilheating equipment (boilers, furnaces and accessories) sales were made by companies that also supply the fuel, and about three-fifths of all heating oil was sold by companies also selling oilheating equipment.

FUELOIL & OIL HEAT, the only publication actively covering both the equipment and oil activities of dealers, was organized in 1922. It is accepted as the editorial and advertising authority for all heating contractors, fueloil dealers, furnacemen and others who really participate in oilheating; also for the many manufacturers who cultivate this vast market for their products.

Manufacturers

Oilburner manufacturers in the late months of 1945 and the winter of 1946 were rapidly getting their production facilities back into high gear on civilian output. With an immediate flush demand for about a half million domestic burners and oilfired heating units, 1946 can be expected to reach new sales peaks. Approximately 200 oilburner manufacturers are now in production. Before the war ten of these were supplying about half of the industry's total output, and when reconversion is completed these companies will probably again sell that share of the total.

The industry installed 145,596 domestic oilburners and oilfired units in 1945, more than half of these in fall months after the war's close. Of these, 58% replaced hand fired coal, 30% replaced old oilburners, 4% went to new homes, 7% replaced coal stokers and 1% replaced gas heating.

Dealers

After a wartime mortality of around 40% in number of dealers, the war's end has brought many of them back to the industry. Present outlook is that within two years there will be considerably more oilheating retailers than the previous peak of 13,000 in 1941. Dealers with oilburner or fueloil background are making 80% of total burner sales; plumbing and heating contractors the other 20%.

Oilheating Tomorrow

According to our projections of post-war activities, there will be a 100 per cent expansion of oilheating markets during the first 5 post-war years, with sales and installations of 2½ million oilheating jobs in all types and sizes of heating plants. They will be divided: 1,544,000 conversions from coal; 561,000 in new homes; and 388,000 replacements of existing jobs.

The equipment will be divided into: 1,443,000 conversion jobs; 406,000 boiler-burner units; 644,000 furnace-burner units.

For a statistical picture of all phases of oilheating—wartime and postwar, send $1.00 for the January, 1946, Yearbook Issue and for a free copy of "Oilheating Tomorrow"—an 8-page statistical study.

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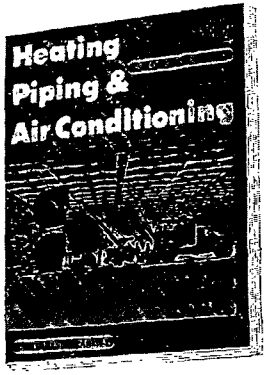

This is the publication which carries the Journal of the A.S.H.V.E. in addition to its own regular editorial section. Its field is that of industry and large buildings. It is devoted to the design, installation, operation, and maintenance of heating, piping and air conditioning systems in such plants and 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 established buying and specifying guide of the industry.

H. P. & A. C. 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. All A.S.H.V.E. 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 H. P. & A. C. those from whom he is seeking the necessary engineering acceptance. Write for our new booklet "A Quick Picture."

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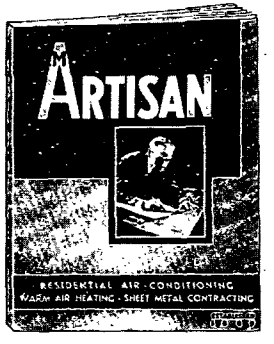

AMERICAN ARTISAN covers the field of warm air heating, residential air conditioning, and sheet metal contracting. Its readers are warm air heating and sheet metal contractors, dealers, jobbers, manufacturers, and public utility companies.

Each January issue includes a complete directory of warm air heating, air conditioning, and sheet metal products and equipment, which lists all products, their trade names, and the manufacturers' addresses.

A special section of each issue has 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, tens 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.

The key man in the 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 the key publication—because it reaches these key men with an information service that has made it the recognized authority on residential air conditioning practice—is AMERICAN ARTISAN. For full information on this field, write for a copy of "The Residential Heating Market."

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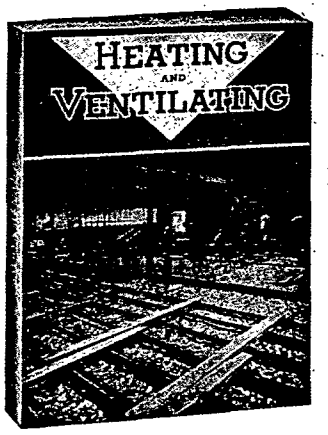

HEATING AND VENTILATING is a monthly publication read by engineers and contractors. These are the men who specify, install and maintain the mechanical equipment used in heating, ventilating, air conditioning and refrigeration systems of industrial, commercial and institutional buildings and residential projects. Its readers include aslo engineers and designers of the firms which manufacture this mechanical equipment.

The editorial content is edited to be of practical use to these engineers, and is prepared under the direction of field-experienced professional engineers.

Generally speaking, the emphasis is on practical rather than on theoretical considerations.

Each month an original Reference Data sheet is included for permanent use in a standard binder (back copies are available).

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Public Utility Group	551
Industrial Firms, their Executives, Engineers and other Employees	2,148
Buildings, Real Estate Management Companies, their Engineers	652
Manufacturers of Air Conditioning, Heating, Piping and Ventilating Equipment, Their Officials and Employees (738) and Designing Engineers (178)	1,057
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Miscellaneous and Unclassified	683
	10,597
Field Staff, Correspondents, Exchanges and Advertising Agencies	392
TOTAL	10,989

Subscriptions to HEATING AND VENTILATING, 148 Lafayette St., New York 13, are $2.00 a year.

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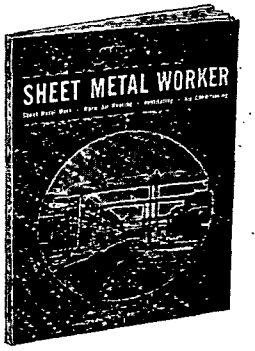

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The market has three main divisions:

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SHEET METAL WORKER also publishes books on heating, ventilating, sheet metal work, air conditioning, etc.

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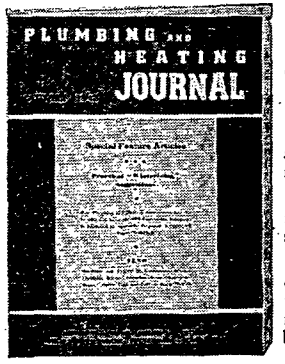

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Central New York
Headquarters, Syracuse, N. Y.
President, C. M. ASHLEY
300 S. Geddes St., Syracuse 1
Secretary, F. E. HOCKENSMITH
400 N. Midler Ave.

Central Ohio
Headquarters, Columbus, Ohio
Meets: Third Monday in Month
President, A. W. WILLIAMS
5 E. Long St., Room 808, Columbus 15
Secretary, H. R. ALLONIER
101 N. High St., Columbus 15

Cincinnati
Headquarters, Cincinnati, Ohio
Meets: Second Tuesday in Month
President, E. J. RICHARD
2137 Reading Rd., Cincinnati 2
Secretary, H. K. JENNINGS
1639 Union Trust Bldg., Cincinnati 2

Officers and List of Chapters, 1945-46—(*Continued*)

Connecticut
Headquarters, New Haven, Conn.
President, STANLEY HART
New Britain
Secretary, WINFIELD ROEDER
405 Temple St.

Delta
Headquarters, New Orleans, La.
Meets: Second Tuesday in Month
President, W. H. GRANT, JR.
209 Vincent Bldg., New Orleans 12
Secretary, J. S. BURKE
317 Baronne St., New Orleans 9

Golden Gate
Headquarters, San Francisco, Calif.
Meets: First Wednesday in Month
President, R. B. HOLLAND
420 Market St., San Francisco 11
Secretary, F. W. KOLB
598 Monadnock Bldg., San Francisco 5

Illinois
Headquarters, Chicago, Ill.
Meets: Second Monday in Month
President, JAMES S. LOCKE
433 E. Erie St., Chicago 11
Secretary, C. M. BURNAM, JR.
Room 1605, 6 N. Michigan Ave., Chicago 2

Indiana
Headquarters, Indianapolis, Ind.
Meets: Fourth Friday in Month
President, G. B. SUPPLE
Architects and Builders Bldg.
Secretary, T. R. DAVIS
1311 N. Drexel Ave., Indianapolis 1

Iowa
Headquarters, Des Moines, Ia.
Meets: Second Tuesday in Month
President, B. E. LANDES
1603 47th St., Des Moines 10
Secretary, C. A. WHEELER
511 Fifth Ave., Des Moines

Kansas City
Headquarters, Kansas City, Mo.
Meets: First Monday in Month
President, R. B. MASON
2014 Wyandotte St., Kansas City
Secretary, P. C. LEFFEL
316 E. 75th St., Kansas City

Manitoba
Headquarters, Winnipeg, Man.
Meets: Third Thursday in Month
President, F. T. BALL
324 Main St.
Secretary, D. S. SWAIN
27 Fawcett Ave.

Massachusetts
Headquarters, Boston, Mass.
Meets: Third Tuesday in Month
President, D. M. ARCHER
143 Federal St., Boston 10
Secretary, A. EHRENZELLER
23 Parklawn Rd., Boston 32

Memphis
Headquarters, Memphis, Tenn.
Meets: First Monday in Month
President, T. J. O'BRIEN
1030 Exchange Bldg.
Secretary, A. T. BEVIL
1521 Waverly Ave., Memphis 6

Michigan
Headquarters, Detroit, Mich.
Meets: First Monday after 10th of Month
President, R. F. CONNELL
1500 United Artists Bldg., Detroit 31
Secretary, A. E. KNIBB
1003 Maryland Ave., Detroit 30

Minnesota
Headquarters, Minneapolis, Minn.
Meets: First Monday in Month
President, R. E. GORGEN
2120 W. Cedar Lake Blvd.
Secretary, O. L. LILJA
5000 16th Ave. S., Minneapolis 7

Montreal
Headquarters, Montreal, Que.
Meets: Third Monday in Month
President, A. B. MADDEN
1170 Beaver Hall Sq.
Secretary, LEO GARNEAU
Room 832, Dominion Sq. Bldg.

Nebraska
Headquarters, Omaha, Nebr.
Meets: Second Tuesday in Month
President, D. E. MCCULLEY
814 S. 14th St., Omaha 2
Secretary, D. D. WILLIAMS
Room 311, Federal Bldg.

New York
Headquarters, New York, N. Y.
Meets: Third Monday in Month
President, H. J. RYAN
101 Park Ave., New York 17
Secretary, CARL H. FLINK
Room 3000, 51 Madison Ave., New York 10

North Carolina
Headquarters, Durham, N. C.
Meets: Quarterly
President, K. W. SELDEN, JR.
701 Builders Bldg., Charlotte 2
Secretary, E. S. DEWITT
1211 Commercial Bank Bldg., Charlotte

Officers and List of Chapters, 1945-46 — (*Concluded*)

North Texas
Headquarters, Dallas, Tex.
Meets: Third Monday in Month
President, E. T. GESSEL
610 Thomas Bldg., Dallas 1
Secretary, C. ROLLINS GARDNER
911 Camp St., Dallas 2

Northern Ohio
Headquarters, Cleveland, Ohio
Meets: Second Monday in Month
President, G. B. PRIESTER
10900 Euclid Ave., Cleveland 6
Secretary, R. L. BYERS
448 Terminal Tower, Cleveland 13

Oklahoma
Headquarters, Oklahoma City, Okla.
Meets: Second Monday in Month
President, EARLE W. GRAY
Third and Harvey Sts.
Secretary, G. T. DONCREL
Oklahoma Natural Gas Co.

Ontario
Headquarters, Toronto, Ont.
Meets: First Monday in Month
President, J. P. FITZSIMONS
4 Mowat Ave.
Secretary, H. R. ROTH
57 Bloor St. W.

Oregon
Headquarters, Portland, Ore.
Meets: Thursday after First Tuesday in Month
President, F. F. URBAN
6726 S. W. Burlingame Ave.
Secretary, E. R. LOKEY
1238 N. W. Glisan St., Portland 9

Pacific Northwest
Headquarters, Seattle, Wash.
Meets: Second Tuesday in Month
President, D. C. GRIFFIN
717 Lloyd Bldg., Seattle 1
Secretary, C. W. FINN
717 Lloyd Bldg., Seattle 1

Philadelphia
Headquarters, Philadelphia, Pa.
Meets: Second Thursday in Month
President, R. D. TOUTON
Ninth and Columbia Ave.
Secretary, F. H. BUZZARD
213 S. Broad St., Philadelphia 7

Pittsburgh
Headquarters, Pittsburgh, Pa.
Meets: Second Monday in Month
President, L. S. MARBLING
778 Country Club Dr., Pittsburgh 16
Secretary, E. H. RIESMEYER, JR.
231-33 Water St., Pittsburgh 22

Rocky Mountain
Headquarters, Denver, Colo.
Meets: First Wednesday in Month
President, G. D. MAVES
1550 Glencoe St., Denver 7
Secretary, FRED JANSSEN
1123 W. Third Ave., Denver 9

St. Louis
Headquarters, St. Louis, Mo.
Meets: First Tuesday in Month
President, W. J. OONK
4548 Red Bud Ave., St. Louis 15
Secretary, W. A. RUSSELL
7918 Kingsbury Blvd., Clayton 5

South Texas
Headquarters, Houston, Tex.
Meets: Third Friday in Month
President, J. A. WALSH
P. O. Box 1773
Secretary, D. M. MILLS
Gray and Crawford, Houston 3

Southern California
Headquarters, Los Angeles, Calif.
Meets: Second Wednesday in Month
President, MARON KENNEDY
5051 Santa Fe Ave., Los Angeles 11
Secretary, R. A. LOWE
3744 Potomac Ave., Los Angeles 16

Utah
Headquarters, Salt Lake City, Utah
Meets: First Wednesday in Month
President, H. G. RICHARDSON
1433 Harvard Ave.
Secretary, E. V. GRITTON
2470 S. 15th East St.

Washington, D. C.
Headquarters, Washington, D. C.
Meets: Second Wednesday in Month
President, W. H. LITTLEFORD
616 New York Ave. N.W.
Secretary, LESTER MAURER
1738-33 Place S.E., Washington 20

Western Michigan
Headquarters, Grand Rapids, Mich.
Meets: Second Monday in Month
President, H. J. METZGER
137 E. Water St., Kalamazoo
Secretary, H. W. WOLTERS
820 Staples Ave. N.W., Kalamazoo 54

Western New York
Headquarters, Buffalo, N. Y.
Meets: Second Monday in Month
President, F. A. MORSEL
31 Main St., Buffalo 3
Secretary, J. S. MEYER
38 Crescent Ave., Buffalo 14

Wisconsin
Headquarters, Milwaukee, Wis.
Meets: Third Monday in Month
President, O. A. TROSTEL
Route No. 2, Thiensville
Secretary, J. R. VERNON
507 E. Michigan St., Milwaukee 2

Roll of Membership

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1946

(Corrected to January 15, 1946)

HONORARY MEMBERS

CARRIER, WILLIS H. (1944) (M 1913), (Presidential Member), Syracuse, N. Y.
FIELDNER, A. C. (1944) Washington, D. C.

BALDWIN, WM. J. (1915) New York, N. Y. (Deceased May 7, 1924).
BILLINGS, DR. J. S. (1896) New York, N. Y. (Deceased March 10, 1913).
BOLTON, REGINALD PELHAM (1897), (Presidential Member), New York, N. Y. (Deceased February 18, 1942).
BRECKENRIDGE, L. P. (1920), North Ferrisburg, Vt. (Deceased August 22, 1940)
GORMLY, JOHN (Charter Member), Norristown, Pa. (Deceased January 31, 1929)
NEWTON, C. W. (Charter Member), Baltimore, Md. (Deceased August 6, 1920).
HOOD, O. P. (1929), Washington, D. C. (Deceased April 22, 1937).
JELLETT, STEWART A. (Charter Member), (Presidential Member), Philadelphia, Pa. (Deceased April 5, 1935).

LIST OF MEMBERS Arranged Alphabetically

(*Asterisk indicates authorship of papers; • indicates address for mail)

(*M* 1923; *A* 1918; *J* 1916) indicates Election as Member 1923; Associate 1918; Junior 1916. (Pres. 1923) indicates Elected President in 1923 and is now a Presidential Member.

A

ABBOTT, Furman S. (*M* 1943) Chief Engr., W. H. Wheeler, Inc., 7 East 47th St., New York, N. Y., and • 80 Afterglow Ave., Montclair, N. J.

ABBOTT, Thomas J. (*M* 1938) Secy.-Treas., • George C. Abbott, Ltd., 52 Shaftesbury Ave. and 294 Oriole Pkwy., Toronto, Ont., Canada.

ABRAMS, Abraham (*M* 1927; *J* 1924) Cons. Engr., 272 North Ave., and • 112 Bon Air Ave., New Rochelle, N. Y.

ABRAMSON, Ralph J. (*A* 1938) Mech. & Elect. Engr., Samuel R. Lewis, 407 S. Dearborn St., Chicago, and • 1834 Pratt Ave., Chicago 26, Ill.

ACHENBACH, Paul R.* (*M* 1942) Asst. Mech. Engr., • National Bureau of Standards, Washington, D. C. and 1912 N. Randolph St., Arlington, Va.

ACHESON, Louis K. (*M* 1944) Laboratory Coordinator, The Hoover Company, and • 816 Portage St., North Canton, Ohio.

ACKERMANN, Reynold H. (*M* 1943) Engr., Charles S. Leopold, Cons. Engr., 213 Broad St., Philadelphia, and • 267 Cooper Ave., Lansdowne, Pa.

ADAIR, James S. (*A* 1941; *J* 1940) • 1420 Valence St., New Orleans 15, La.

ADAM, Ray W. (*A* 1938) Owner, • W. A. Adam Co., 8810 Grinnell Ave., Detroit 13, and 951 E. Grand Blvd., Detroit, Mich.

ADAMS, A. K. (*A* 1944) Mgr., Montana Iron Works, and • 2124 Argyle, Butte, Mont.

ADAMS, Chester Z. (*M* 1939) Owner, • Chet Adams Co., 312 Piedmont Bldg., Box 1356, and 207 E. Avondale Dr., Greensboro, N. C.

ADAMS, Eugene E. (*A* 1938) Sales Engr., • Garden City Fan Co., Room 1508 A, 55 West 42nd St., New York 18, and 35-46 79th St., Jackson Heights, N. Y.

ADAMS, Eugene F. (*M* 1941) Mech. Engr., Peter Kiewit Sons Co., and • 820 Ivinson Ave., Laramie, Wyo.

ADAMS, Frank L. (*M* 1939) Htg. & Air Cond. Engr., • P. O. Box 840, Denver 1, and 1185 Grape St., Denver 7, Colo.

ADAMS, George F. (*M* 1944) Engr., • Midwest Plumbing & Heating Co., 2450 Blake St., Denver, and 3301 Jay St., Wheat Ridge, Colo.

ADAMS, Harold E. (*M* 1930) Chief Engr., • The Nash Engineering Co., South Norwalk, and Merrill Heights, Norwalk, Conn.

ADAMS, N. D. (*M* 1929) (Council, 1938-40) Supt., • Franklin Heating Station, 220 Second Ave., S.W., and R.F.D. 1, Elmi Dell, Rochester, Minn.

ADAMS, Octavius G. (*A* 1944) Sales Engr. & Owner, • Adams Heating Equipment Co., 41 East 42nd St., Room 314, New York 17, and 32-20 93rd St., Jackson Heights, N. Y.

ADAMS, William J. (*A* 1945) Estimator, • 3911 W. North Ave., Chicago 47, and 1718 Nordica Ave., Chicago 35, Ill.

ADAMSON, L. G. (*A* 1945) Owner. • Air Conditioning & Heating Service Co., 3922 Troost, Kansas City 4, and 1920 East 71st Terrace, Kansas City, Mo.

ADDAMS, Homer (*Charter Member; Life Member*), (*Presidential Member*), (Pres., 1924; 1st Vice-Pres., 1923; Treas., 1915-22; Council, 1915-25), Pres., Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York 17, N. Y.

ADDAMS, Paul K. (*M* 1943) Exec. Vice-Pres., • Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York 17, and 465 West 23rd St., New York 11, N. Y.

ADDIE, George R. (*M* 1945) Managing Dir., J. G. Wagstaff, Ltd., Alma Iron Works, Dukinfield, Cheshire, and • "Bryn Hendre," Conway, North Wales, England.

ADDINGTON, Harold M. (*M* 1939) • Aurora Hall, Oak Ridge, Tenn.

ADDINGTON, Herbert B. (*M* 1938) Cons. Engr., • Myers, Fuller and Addington, 21 East 40th St., New York 16, and 699 Ocean Ave., Brooklyn 26, N. Y.

ADEMA, George E. (*M* 1939) Prop. • N. M. Adema & Son, 39 W. Balcom St., Buffalo 9, and 299 Deerhurst Pkwy., Kenmore 17, N. Y.

ADLAM, T. Napier (*M* 1932) Vice-Pres. & Gen. Mgr., Sarco Manufacturing Corp., 475 Fifth Ave., New York, N. Y., and • 124 Forest Hill Rd., West Orange, N. J.

AEBERLY, John J. (*M* 1928), (Council, 1937-39), Chief of Bureau of Htg. Vtg. and Indus. Sanitation, • Dept. of Buildings, Room 702 City Hall, Chicago 2, and 6225 N. Newcastle Ave., Norwood Park P.O., Chicago 31, Ill.

AHEARN, William J. (*M* 1929) • 791 Tremont St., Boston, and 131 Windermere Rd., Auburndale, Mass.

AHLFF, Albert A. (*M* 1923; *A* 1918) Spec. Repr., Hajoca Corp., Box 7319, Philadelphia, and • 1521 Powder Mill Lane, Wynnewood, Pa.

AHRENS, Clarence F. (*A* 1940) Sales Engr., R. A. Dubuque Supply Co., 3960 Duncan Ave., St. Louis 10, and • 4151 Tonenges Ave., St. Louis 16, Mo.

AIKIN, Aziel A. (*M* 1945) Plant Engr., & Design Supvsr., • Oldsmobile, Lansing 21, and Maplewood Manor, Lansing, Mich.

AIKIN, Frank W. (*A* 1945) Pres., • Northern Controlled Heat Co., Inc., 557 Factory Square, Watertown, and 126 Ward St., Watertown, N. Y.

AINSWORTH, Samuel E. (*A* 1939) Partner, • Roche Newton & Co., Box 1049, and 1524-25th St., Lubbock, Texas.

AITKEN, Milton L. (*M* 1945) Gen. Mgr., Propellair, Inc., Springfield, and • 99 S. Broadmoor Blvd., S., Springfield, Ohio.

AKERLOW, R. W. (*A* 1944) Sales Engr., • The Stearns Roger Manufacturing Co., 1720 California St., Denver, and 935 S. Elizabeth St., Denver, Colo.

AKERS, George W. (*M* 1929) Mgr., Engrg. & Sales, • George W. Akers Co., 16525 Woodward Ave., Detroit 3, and 1530 Seward Ave., Detroit 6, Mich.

ALBERT, William J. (*A* 1945) Engr. & Estimator, Buensod-Stacey, Inc., 1001 N. Church St., Charlotte, N. C., and • 665 Randolph St., Camden, N. J.

ALBRIGHT, C. B. (*A* 1945) Director, • Industrial Services Assocs., 120 Liberty St., New York 6, and Mt. Kisco, N. Y.

ALEXANDER, Granville P. (*M* 1944) Secy., • Heating, Piping & Air Conditioning Contractors Philadelphia Assn., 328 Chestnut St., Philadelphia 6, and Montgomery Court, Apt. J-33, Narberth, Pa.

ALEXANDER, Hugh V. (*M* 1943) Branch Mgr., • Johnson Service Co., 922 Architects Bldg., Indianapolis 4, and 5700 Kingsley Dr., Indianapolis 5, Ind.

ALEXANDER, John L. (*A* 1944) Mfrs. Agent, • 2746 Elmwood Dr., S.E., Grand Rapids 6, Mich.

ALEXANDER, John W. (*M* 1945) Design Engr., • Office of the Supervising Architect, Public Buildings Administration, Washington 25, D. C., and 403 Prince George St., Laurel, Md.

ALEXANDER, Keith O. (*M* 1943) Sales Engr., • Neil H. Peterson Co., 1129 Folsom St., San Francisco 3, and 454 Mastick Ave., San Bruno, Calif.

ALEXANDER, Samuel W. (*M* 1935) Pres., S. W. Alexander Co., Ltd., 182 Main St., and • 124 Kingsmount Park Rd., Toronto, Ont., Canada.

ALFERY, Henry F. (*M* 1938) Gen. Mgr., Erie Manufacturing Co., 300 N. Eighth St., Milwaukee 3, and • 1819 W. Center St., Milwaukee 6, Wis.

ALGREN, Axel B. (*M* 1930) Assoc. Prof., Htg., Vent. & Air Cond., Engrg. Experiment Station, University of Minnesota, and • 5109-17th Ave., S., Minneapolis, Minn.

ALLAN, William (*A* 1938) Pres., • Allan Engineering Co., 724 E. Mason St., Milwaukee 2, and 2735 N. Farwell Ave., Milwaukee 11, Wis.

ALLEN, Abraham B. (*A* 1945) Office Mgr., • Oil Burning Engineers, Inc., 3146 W. Chicago Ave., Chicago 22, and 2233 Birchwood Ave., Wilmette, Ill.

ALLEN, A. Walter (*M* 1936) Sales Engr., • Pease Foundry Co., Ltd., 151 Glen Ave., Ottawa, Ont., Canada.

ALLEN, DeWitt M. (*M* 1936) Owner, • D. M. Allen Co., Room 101, B.M.A. Bldg., 215 W. Pershing Rd., Kansas City 8, and 39 West 70th Terrace, Kansas City 5, Mo.

ALLEN, Frank C. (*M* 1944) Mech. Engr., U. S. Bureau of Reclamation, U. S. Customhouse, Denver, and • 4467 King St., Denver 11, Colo.

ALLEN, George W. (*A* 1945) Steamfitter, • 64 Raymond St., Gardner, Mass.

ALLEN, Harold D. (*A* 1944) Mech. Engr., National Advisory Committee for Aeronautics, Cleveland Airport, Cleveland, and • 1584 Larchmont Ave., Lakewood 7, Ohio.

ALLEN, J. Lloyd (*M* 1944) Partner, • Allen & Kelley, 333 N. Penn, Indianapolis, and 5699 N. Delaware, Indianapolis, Ind.

ALLEN, William A. (*A* 1942) Pres., • Sprague & Sprague, Inc., 6230 Penn Ave., and 216 Hilands Ave., Pittsburgh, Pa.

ALLEN, William W. (*A* 1936) Pres., • American Coolair Corp., Box 2300, Jacksonville, Fla.

ALLISON, John H. (*A* 1945) Sales Engr., E. J. Deckman Co., 1502 Oliver Bldg., Pittsburgh 22, and • 601 Sherwood Rd., Pittsburgh 21, Pa.

ALLISON, Robert E. (*A* 1941) Owner, • American Sheet Metal Co., 601 First Ave., Dallas 10, and 7069 Fairdale, Dallas, Texas.

ALLONIER, Howard R. (*A* 1936) Dist. Mgr., • J. J. Nesbitt, Inc., 101 N. High St., Columbus 15, and R. D. No. 1, Powell, Ohio.

ALLSOP, Rowland P. (*A* 1940; *J* 1934) Cons. Engr., • 1221 Bay St., Toronto, and Stop 21 Kingston Rd., Scarboro Bluffs, Ont., Canada.

ALMERT, Harold (*A* 1944) Cons. Engr., • 1605 York Ave., and 519 East 86th St., Apt. 6-B, New York 28, N. Y.

ALT, Harold L.* (*M* 1913) Basic Mech. Design Engr., Voorhees, Walker, Foley & Smith, 101 Park Ave., New York, and • 115-27-225th St., St. Albans 11, N. Y.

ALTEMUELLER, George (*A* 1940) Asst. Utility Officer, Veterans Administration Facility, and • R. F. D. 1, Middleville Rd., Northport, L. I., N. Y.

ALVAREZ, Joaquin (*A* 1946; *J* 1942) Cpl., U. S. A. 34968996, • U. S. Strategic Bombing Survey, Army Hall, City College, and 1560 Amsterdam Ave., New York, N. Y.

ALYEA, Harold W. (*M* 1944) Design and Dvlpt. Engr., • Johnson Service Co., 507 E. Michigan St., Milwaukee, and 919 Arcadian Ave., Waukesha, Wis.

AMBROSE, Alfred H. (*J* 1943; *S* 1941) A. R. T. 3/c, U. S. Naval Reserve, and • 18 Linden St., Woodstock, Vt.

AMBROSE, E. R. (*M* 1940) Air Cond. Engr., • American Gas & Electric Service Corp., 30 Church St., New York, N. Y., and 153 Lincoln Ave., E., Roselle Park, N. J.

AMMERMAN, A. S., Jr. (*M* 1944; *A* 1941; *J* 1937) Dist. Mgr., • Aeronn Corp., 111 W. Washington St., Chicago 2, and 132 N. Wolf Rd., Des Plaines, Ill.

AMMERMAN, Charles R. (*M* 1916) Pres., • Ammerman, Davis & Stout, Inc., 805-9 K. of P. Bldg., and 4916 Rockwood Ave., Indianapolis, Ind.

AMUNDSON, Carl H. (*M* 1944) Product Application Engr., • Herman Nelson Corp., 759 N. Milwaukee, Milwaukee 11, and 2324 E. Park Pl., Milwaukee 2, Wis.

ANDEREGG, R. H. (*M* 1920) Vice-Pres. & Chief Engr., The Trane Co., and • 450 Losey Ct., La Crosse, Wis.

ANDERSON, Carl G. (*M* 1942) Chief Engr., • Chicago Metal Hose Corp., 840 N. State St., and 50 N. Edison St., Elgin, Ill.

ANDERSON, Carl O. (*M* 1944) Htg. & Vent. Engr., Board of Education, Rm. 536, 228 N. LaSalle St., Chicago 1, and • 900 N. Rush St., Chicago 11, Ill.

ANDERSON, Carroll S. (*M* 1920) Dist. Mgr., • American Blower Corp., 1105 Architects Bldg., Los Angeles 13, and 4267 Holly Knoll Dr., Hollywood 27, Calif.

ANDERSON, David B. (*A* 1939; *J* 1936; *S* 1933) Lt., U. S. N. R., Naval Ordnance Laboratory, Navy Yard, Washington, D. C., and • 1999 Pinehurst Ave., St. Paul 5, Minn.

ANDERSON, Edwin C. (*A* 1944) Owner, • Edwin C. Anderson Co., 1863 Wazee St., Denver 2, and 1623 Adams St., Denver 6, Colo.

ANDERSON, Edwin J. (*A* 1939) Mfrs. Agent, • Chicago Pump Co., 14 Smith St., Detroit 2, and 274 Lenox, Detroit 15, Mich.

ANDERSON, Edwin L. (*M* 1941) Cons. Engr., • 63 Oakridge Rd., Verona, N. J.

ANDERSON, Einar (*A* 1940) Sales Engr., Vulcan Iron Works, Ltd., Sutherland and Maple St., and • 152 Bannerman Ave., Winnipeg, Man., Canada.

ANDERSON, George A. M. (*A* 1939; *J* 1936) Pres., • King Ventilating Co., Owatonna, and 717 S. Cedar, Owatonna, Minn.

ANDERSON, Granville W. (*A* 1943) Service Engr., • Koppers Coal Division, 1217 Ford Bldg., Detroit 26, and 5096 Burns Ave., Detroit 13, Mich.

ANDERSON, Harvey E. (*A* 1944) Secy., • Narowetz Heating & Ventilating Co., 1722 Washington Blvd., Chicago 12, and 460 Fair Ave, Elmhurst, Ill.

ANDERSON, John W. (*A* 1944; *J* 1937) Design Engr., Titetlex, Inc., 500 Frelinghuysen Ave., Newark, and • 201 Ayliffe Ave., Westfield, N. J.

ANDERSON, Nathan B. (*A* 1945) Owner, • Temperature Equipment & Engineering Co., 620 Walker, and Ben Milan Hotel, Houston, Texas.

ANDERSON, Robert B. (*A* 1944) Htg. Contr., • 1210 West Byers, and 853 Ogden, Denver, Colo.

ANDERSON, Robert V. (*A* 1945) Div. Engr., • Lennox Furnace Co., Inc., 400 N. Midler Ave., Syracuse, N. Y., and 11 E. Sedgwick St., Sandston, Va.

ANDERSON, Sydney, Jr. (*A* 1943) Dist. Engr., • Chrysler Airtemp Sales Corp., 925 Bowen Bldg., Washington 5, D. C., and 8503 Irvington Ave., Bethesda 14, Md.

ANDERSON, William S., Jr. (*M* 1945) Partner, Axeman-Anderson Assocs., 233 West St., and • 1332 Sheridan St., Williamsport, Pa.

ANDREWS, William G. (*A* 1941) Branch Mgr., • Wholesale Refrigerating Jobbers, 1513 Camden Rd., Charlotte, N. C., and 2232 N.W. Second St., Miami, Fla.

ANDREWS, W. M. (*M* 1941) Partner, • Lockwood & Andrews, 904 Union National Bank Bldg., and 2333 Sunset, Houston, Texas.

ANDREWS, William R. (*M* 1942) Asst. Mgr., • Ross Engineering of Canada, Ltd., Rm. 920, Dominion Square Bldg., and 3770 Cote St. Catherine Rd., Montreal, P. Q., Canada.

ANGELL, Lawrence D. (*M* 1944) Minneapolis-Honeywell Regulator Co., New York, and • 450 East 16th St., Brooklyn, N. Y.

ANGELO, Herbert G. (*A* 1944) Htg. & Vent., 4805 Washington, Denver 16, and • 1225 East 14th Ave., Apt. 28, Denver, Colo.

ANGERMEYER, Albert H. (*A* 1936) Owner, • Plumbing & Heating, 119 N. Commercial St., and 245 Webster St., Neenah, Wis.

ANGUS, Frank M. (*M* 1937) • Texas Refrigeration & Engineering Co., 510 Magnolia Bldg., Dallas 1, and 6033 Victor, Dallas, Texas.

ANGUS, Harry H.* (*M* 1918), (Council, 1927-29) Cons. Engr., 1221 Bay St., and • 34 Farnham Ave., Toronto, Ont., Canada.

ANOFF, Seymour M. (*J* 1940) Cons. Engr., • 5048 Marine Dr., Apt. A-5, Chicago 40, Ill.

ANSPACHER, T. H. (*M* 1939; *J* 1936) Dist. Mgr., • Buffalo Forge Co., Tower Petroleum Bldg., Dallas 1, and 4512 Arcady St., Dallas, Tex.

ANTHES, Lawrence L. (*A* 1935) Pres., • Imperial Iron Corp., Ltd., 30 Jefferson Ave., and 119 Dowling Ave., Toronto, Ont., Canada.

ANWAY, H. Wilbur (*M* 1943) Research Physicist, • Wood Conversion Co., Arch St., and 1012 Prospect Ave., Cloquet, Minn.

APPELT, Joseph E. (*M* 1945) Mgr., The Trane Co., 1200 Godfrey, S.W., Grand Rapids, and • Green Ridge Dr., R. R. 1, Comstock Park, Mich.

APPLEGATE, Francis R. (*A* 1944) Plant Service Engr., Pratt & Whitney Aircraft Corp. of Mo., and • 5129 Wornall Ave., Kansas City, Kans.

APT, Sanford R. (*M* 1935) Mech. Engr., Parsons, Brinckerhoff, Hogan & Macdonald, 142 Maiden Lane, New York 7, and • 36-39-205th St., Bayside, L. I., N. Y.

ARCHAMBAULT, Joseph A. (*A* 1939) Branch Sales Office Mgr., • C. A. Dunham Co., Ltd., 22 Wellington St., N., Rm. 17, and 55A Council St., Sherbrooke, Que., Canada.

ARCHER, David M. (*M* 1934) Sales Repr., • Sarco Co., Inc., 143 Federal St., Boston 10, and 10 Harding Ave., Braintree, Mass.

ARCHER, William J. (*M* 1945) Engr., • Howes & Whitaker, 626 Russ Bldg., and 1443 Funston Ave., San Francisco, Calif.

ARENBERG, Milton K. (*A* 1920) Pres., Robert Barclay, Inc., 122 N. Peoria St., Chicago, and • 1415 Wildwood Lane, Highland Park, Ill.

ARGUE, Edgar J. (*A* 1935) Sales Engr., Anthes Foundry, Ltd., Saskatchewan Ave., and • 773 McMillan Ave., Winnipeg, Man., Canada.

ARMBRUSTER, Frank T. W. (*M* 1936) Service Engr., American Radiator & Standard Sanitary Corp., 73 E. Naghten St., Columbus, and • 105 First Ave., Waverly, Ohio.

ARMISTEAD, William C. (*M* 1937) Sales Engr., • 153 Second Ave., N., Nashville 3, and Granny White Pike, Nashville, Tenn.

ARMSPACH, Otto W. (*M* 1919) Dist. Chief Engr., Carrier Corp., 405 Lexington Ave., New York 17, and • N. 2, Stratford Rd., Scarsdale, N. Y.

ARMSTRONG, Charles E. (*M* 1939) Chief Engr., • Armstrong Heat Control Co., 523 S.E. 39th Ave., and 624 N.E. Hazelfern Pl., Portland 15, Ore.

ARMSTRONG, Walter J. (*M* 1938) Cons. Engr., • 1010 St. Catherine St., W., Montreal, and 15 Willow Ave., Westmount, Que., Canada.

ARNDT, Heinrich W. (*A* 1944) Owner, • The Hobby Shop, Masonic Bldg, Lobby No. 2, and 2007 Central Ave., Augusta, Ga.

ARNOLD, Richard S. (*M* 1944) Asst. Chief Engr., • Carrier Corp., 300 Ivy St., N.E., Atlanta 3, and 2195 E. Lake Rd., Atlanta, Ga.

ARNOLD, Robert S. (*M* 1944; *A* 1926; *J* 1922) Owner, Robert Arnold Sales & Engineering Co., 913 Otis Bldg., Philadelphia 2, and • Sherwood Lane, Wallingford, Pa.

ARNOLD, Thomas L. (*A* 1945) Mgr., • Vulcan Radiator Co., 26 Francis Ave., Hartford 6, and 59 Beverly Rd., West Hartford 7, Conn.

ARONSON, Henry H. (*A* 1939) Engr., Premier Furnace Co., and • Box 221, Dowagiac, Mich.

ARROWSMITH, John O. (*M* 1934) Asst. Supt. of Works, • Canadian Kodak Co., Ltd., 9 Humberview Rd., Toronto 9, Ont., Canada.

ARTHUR, John M., Jr. (*M* 1923) Div. Mgr., • Lighting-Steam & Commercial Service Div., Kansas City Power & Light Co., 1330 Baltimore Ave., Kansas City 10, Mo., and 3311 State Ave., Kansas City, Kans.

ARTRAN, Carlyle (*A* 1945) Engr., General Air Conditioning Corp., 911 East 59th St., Los Angeles 1, and • 2323 St. George St., Los Angeles 26, Calif.

ARVIDSON, Ernest R. (*M* 1943) • 305 W. Sheridan Ave., Shenandoah, Iowa.

ARZET, Fred A. (*A* 1944) Mech. Engr., Argonaut Realty Div., General Motors Corp., Research Bldg., Detroit, and • 25632 Annapolis, Inkster, Mich.

ASH, Norman B. (*A* 1945) Pres., • Braniff Engineering Co., 817 N. Broadway, Oklahoma City, and Britton, Okla.

ASH, Robert S. (*A* 1943; *J* 1940) Lt. S (E) U. S. N. R., Industrial Dept., Navy Yard, Mare Island, Calif., and • 432 Wisconsin Ave., Oak Park, Ill.

ASHCRAFT, J. Phillip (*A* 1943) Branch Mgr., Minneapolis-Honeywell Regulator Co., 1907 Federal St., Dallas 1, and • 2826 Fondren Dr., Dallas 5, Texas.

ASHLEY, Carlyle M.* (*M* 1931), (Council, 1943-45) Dir. of Dvlpt., • Carrier Corp., 300 S. Geddes St., Syracuse 1, and 22 Lynacres Blvd., Fayetteville, N. Y.

ASHLEY, Edward E. (*M* 1912) Cons. Engr., • 10 East 40th St., New York 16, N. Y., and Noroton Heights, Conn.

ASHLEY, Ronald D. (*M* 1944) Partner, • United Equipment Co., 337 S. High St., Columbus 15, and 141 S. Southampton Ave., Columbus 4, Ohio.

ASHTON, Jed L. (*A* 1944) Owner, Ashton Heating & Air Conditioning, 163 Motor Ave., Salt Lake City 1, and • 2686 Alden St., Salt Lake City 5, Utah.

ATHERTON, Alfred E. (*A* 1937) Dir., • A. E. Atherton & Sons Pty., Ltd., 383 Latrobe St., Melbourne, Victoria, and 129 Esplande, Elwood, Vic., Australia.

ATHERTON, Russell C. (*M* 1945) Engr., M. J. Daly & Sons, Inc., 541 Bank St., Waterbury, and • 74 Essex Ave., Waterbury 64, Conn.

ATKINS, George E. (*M* 1941) Cons. Engr., 1308 Hobart Bldg., San Francisco, and • 64 Oak Ridge Rd., Berkeley, Calif.

ATKINS, Thomas J. (*M* 1945) Electrostatic Air Cleaning Engr., Westinghouse Electric Corp., Philadelphia, and • 276 Henley Rd., Philadelphia 31, Pa.

ATWATER, Harry A. (*M* 1945) Chief Engr., Combustion Equipment Co., 1820 Cherry St., and • 641 West 67th St., Kansas City, Mo.

AUSTIN, William H. (*A* 1943; *J* 1940; *S* 1937) Dvlpt. Engr., • United American Soda Fountain Corp., 101 Walnut St., Watertown 72, and 1095 Hyde Park Ave., Hyde Park 36, Mass.

AVERA, Alvan Franklin (*M* 1944) Chief Engr., • Dallas Air Conditioning Co., 2809 Canton St., and 5810 Lewis, Dallas, Texas.

AVERY, Lester T. (*M* 1934) (Council, 1944-45) Pres., • Avery Engineering Co., 1906 Euclid Ave., Cleveland 15, and 21149 Colby Rd., Shaker Heights, Cleveland 22, Ohio.

AXEMAN, James E. (*M* 1932; *A* 1931; *J* 1925) • Axeman-Anderson Assoc., 233 West St., Williamsport 3, and 1328 Woodmont Ave., Williamsport, Pa.

AXTHELM, Fred G. (*A* 1945) Front Rank Furnace Co., and • 648 N. Forest Ave., Webster Groves 19, Mo.

B

BAAK, W. J. (*A* 1944) Moduflow Zone Supvsr., • Minneapolis-Honeywell Regulator Co., 433 E. Erie St., and 7737 N. East Lake Terrace, Chicago, Ill.

BAAN, George B. (*A* 1945) Plant Engr., • Hat Corporation of America, South Norwalk, and Clover Ave., Norwalk, Conn.

BABCOCK, Dan (*M* 1945) Plant Engr., Allison Div., General Motors Corp., Indianapolis, and • 6511 Riverview Dr., Indianapolis 5, Ind.

BABCOCK, Paul R. (*M* 1941) Cons. Engr., G. M. Simonson, 625 Market St., San Francisco, and • 328 24th St., Oakland 12, Calif.

BABER, John E. (*A* 1940) Lt. Comdr., U.S.N.R., and • 1408 Independence Bldg., Charlotte 2, N. C.

BACHMANN, Arthur J. (*J* 1940; *S* 1939) Staff Sgt. 2162 U. S. Army, Administrative Specialist, and • 59-38-69th Ave., Ridgewood 27, N. Y.

BACHOFER, Henry A., Jr. (*A* 1942; *J* 1938) T/Sgt., • Co. "C" 27th Inf. A. P. O. 25, c/o Postmaster, San Francisco, Calif., and 534 S. Eighth St., Salina, Kans.

BACKSTROM, Russell E. (*A* 1931; *J* 1928) Mgr., Indust. Dept., • Wood Conversion Co., W. 1981 First National Bank Bldg., St. Paul 1, and 1655 Hillcrest Ave., St. Paul 5, Minn.

BACON, William H., Jr. (*A* 1942) Automotive Engr., Tide Water Associated Oil Co., Bayonne, and • 2351 Seneca Rd., Westfield, N. J.

BADGETT, W. Howard (*M* 1937; *J* 1932) Lt. Col., Asst. Mgr., • College Construct., Program, A. & M. College of Texas, Box 235 Faculty Exchange, College Station, and 204 Pershing Ave., College Station, Texas.

BAECHLIN, Alfred C., Jr. (*M* 1942) Gen. Sales Mgr., Northern Air Conditioning Corp., 21 Central Ave., Newark, and • 26 Whitney Rd., Short Hills, N. J.

BAGGALEY, Walter (*M* 1938) Advisory Equip. Engr., Research Div., • The Austin Co., 16112 Euclid Ave., Cleveland 12, and 3390 Glencairn Rd., Shaker Heights 22, Ohio.

BAHNSON, Agnew H., Jr. (*A* 1945) Vice Pres., • The Bahnson Co., 1001 S. Marshall St., Winston-Salem, N. C.

BAILEY, Charles F. (*J* 1939) Newport News Shipbuilding & Drydock Co., Newport News, and • Windsor, Va.

BAILEY, Frederick A., Jr. (*A* 1939) Prop., • Bailey's, 130 King St., Charleston 5, and 70 Warren St., Charleston, S. C.

BAILEY, J. L. (*A* 1940) Asst. Chief Engr., • Parks-Cramer Co., Box 946, and 1705 Fountain View Ave., Charlotte, N. C.

BAIN, Lawrence F. (*A* 1944) Constr. Supt., Narowetz Heating & Ventilating Co., 1722 W. Washington Blvd., and • 3324 N. Laramie Ave., Chicago, Ill.

BAIRD, Alexander D. G. (*A* 1944) Pres., • A. G. Baird, Ltd., 286 Lisgar St., Toronto, Ont., Canada.

BAIRD, Floyd E. (*M* 1940) Owner, • Sales & Engrg., 314 Palmer Bldg., Atlanta 3, and 900 Church St., Marietta, Ga.

BAKER, Clyde H. (*M* 1945) Htg., Vent. & Plbg. Engr., Snyder & McLean, 2214 Penobscot Bldg., Detroit, and • 15444 Vaughan St., Detroit 23, Mich.

BAKER, Donald L. (*A* 1940) Engr. Service Dept., Carrier Corp., and • 200 Euclid Ave., Syracuse 10, N. Y.

BAKER, Edward L. (*A* 1945) Mgr., • Burbidge Coal Co., 311 S. Main, Salt Lake City 1, and 1533 Glen Arbor, Salt Lake City 5, Utah.

BAKER, Harland E. (*A* 1946; *J* 1945) Partner, Baker & English, 1360 W. Ninth St., and • 3786 Freemont Rd., South Euclid 21, Ohio.

BAKER, Harold S. (*A* 1937) Sales Engr., Westinghouse Electric Co., San Francisco, and • Rt. 2, Box 341, Gilroy, Calif.

BAKER, Harry L., Jr. (*A* 1943; *J* 1935) Lt. Comdr., U. S. N. R., and • 545 Winton Ter. N.E., Atlanta, Ga.

BAKER, Henry A. (*M* 1945) Vice-Pres., W. B. Connor Engineering Corp., 114 East 32nd St., New York, and 145-53 17th Rd., Whitestone, N. Y.

BAKER, Howard C. (*M* 1945) • Gilbert Associates, Inc., 412 Washington St., Reading, Pa.

BAKER, I. C. (*M* 1921) Vice-Pres., • Chrysler Corp., Airtemp Div., 1119 Leo St., Dayton 1, and 252 E. Monteray Ave., Dayton 9, Ohio.

BAKER, James (*M* 1944) Owner, Ross & Greig Registered, 738 Notre Dame St., W., and • 5715 Darlington Ave., Montreal, Que., Canada.

BAKER, L. P. (*A* 1945) Ass't. Trade & Industrial Commissioner, • Province of Ontario, Ontario House, 13 Charles 11 St., London, S.W. 1, England.

BAKER, Roland H. (*M* 1928; *A* 1924) • Captain, U. S. N. R., Industrial Manager, U. S. N., Navy No. 116, c/o Fleet Post Office, New York, N. Y.

BAKER, Thomas (*M* 1938) Chief Engr., Consolidated Conditioning Corp., 460 S. Tenth Ave., Mt. Vernon, and • 660 East 242nd St., New York, N. Y.

BAKER, T. A. (*M* 1942) Vice-Pres., • Baker Specialty & Supply Co., 701 Erie Ave., and 2205 E. Broadway, Logansport, Ind.

BAKER, William H., Jr. (*A* 1935) Asst. Mgr., • Pittsburgh Sales Office, American Radiator & Standard Sanitary Corp., P. O. Box 1226, Pittsburgh, and 1241 Murdock Rd., Pittsburgh 17, Pa.

BAKKE, George V. (*J* 1944; *S* 1943) • 3649 44th Ave., S., Minneapolis 6, Minn.

BAKKO, Rueben (*A* 1944) • Beverly Dairies, 230 W. Jefferson St., Los Angeles, Calif.

BALCH, Robert M., Jr. (*A* 1944) Sales Mgr., • Apartment Controls Div., Minneapolis-Honeywell Regulator Co., and 402 W. Minnehaha Parkway, Minneapolis, Minn.

BALDWIN, Charles W. (*A* 1945) Vent. Engr., • U. S. Rubber Co., 1230 Sixth Ave., New York, N. Y., and 439 Waltham St., Hammond, Ind.

BALDWIN, Karl F., Jr. (*A* 1941; *J* 1938) Mech. Engr., Clyde E. Bentley, 1441 Franklin St., Oakland, and • 419 Kentucky Ave., Berkeley 7, Calif.

BALL, Fred T. (*A* 1940) Dept. Mgr., • The Canadian Fairbanks-Morse Co., Ltd., 324 Main St., and Ste. 3, 1212 Wellington Crescent, Winnipeg, Man., Canada.

BALL, William (*A* 1936) Pres., • Interstate Heating & Plumbing Co., 521 Southwest Blvd., Kansas City, Mo., and 1026 Shawnee Rd., Kansas City, Kans.

BALLAGH, A. Bruce (*A* 1946; *J* 1943) Sales Engr., B. F. Sturtevant Co., of Canada, Ltd., 137 Wellington St., W., and • 152 Balsam Ave., Toronto, Ont., Canada.

BALLANTYNE, George L. (*A* 1936) Mgr., Htg. Dept., • Crane, Ltd., 1170 Beaver Hall Sq., Montreal, and 140 Ballantyne Ave., S., Montreal West, Que., Canada.

BALLMAN, William H. (*M* 1937) Plant Mgr., Canning Div., • Wm. P. McDonald Corp., Auburndale, and Davenport, Fla.

BALSAM, Charles P. (*M* 1932) Gen. Mgr., • National Home Equipment Co., 50 Church St., New York 7, and 324 Fourth St., Brooklyn, N. Y.

BAMOND, Manuel J. (*M* 1942) Engr., • Barber-Colman Co., 221 N. LaSalle St., Chicago, and 5859 Winthrop Ave., Chicago, Ill.

BANACH, Casimer J. (*A* 1946; *J* 1939) Chief Draftsman, Johnson Fan & Blower Corp., 1319 W. Lake St., Chicago 7, and • 817 W. Agatite Ave., Chicago 40, Ill.

BAND, Jack P. (*A* 1944) R. C. N. V. R., • Naval Service Headquarters, Rm. 4601, and 6 Somerset St., W., Ottawa, Ont., Canada.

BANKS, John B. (*A* 1942) Br. Mgr., • Minneapolis-Honeywell Regulator Co., 122 N. E. Broadway, Portland 12, and 4615 N. E. 40th Ave., Portland 11, Ore.

BANKSTON, Clayton V. (*A* 1945) Dist. Repr., Industries Sales Corp., and • 29 San Jose Ave., Jefferson Parish 20, New Orleans, La.

BANOWSKY, Aubra B. (*M* 1938) Mgr., Retail Div., Payne Furnace Co., P. O. Box 990, Beverly Hills, and • 2652 Glendon Ave., Los Angeles 34, Calif.

BARAGAR, James S. (*A* 1944) Prod. Service Engr., • Asbestos Supply Co., P. O. Box 2234, Spokane 2, and W. 24-25th, Spokane 10, Wash.

BARBER, Richard C. (*J* 1945) Draftsman, J. C. Russell Co., and • 1433 19th St., Columbus, Ga.

BARBIERI, Patrick J. (*A* 1943; *J* 1936; *S* 1933) Field Service Engr., Sperry Gyroscope Co., Inc., Manhattan Bridge Plaza, Brooklyn, and • 2237 Belmont Ave., New York, N. Y.

BARKER, M. Stanley (*A* 1943) Application Engr., Carrier Corp., 122 East 42nd St., New York, and • 111-32 75th Ave., Forest Hills, N. Y.

BARLOW, F. John (*A* 1945; *J* 1943) Asst. Chief Engr., • Western Condensing Co., 935 E. John St., and 1515 N. Durkee St., Appleton, Wis.

BARNARD, M. Everett (*A* 1936) Sales Engr., • Carrier Corp., 12 South 12th St., Philadelphia 7, and 6730 Crittenden St., Philadelphia 19, Pa.

BARNES, Arthur F. (*M* 1938) Owner, • Texas Engineering Co., 602 Kirby Bldg., Houston 2, and 3015 Jarrard St., Houston, Texas.

BARNES, Arthur R. (*M* 1937) Engr., • U. S. Supply Co., 1315 West 12th St., and 326 East 70th Terrace, Kansas City, Mo.

BARNES, Charles A. (*M* 1944) Engr., • E-G Sheet Metal Works, 258 Pryor St. S.W., and 947 Katherwood Dr. S.W., Atlanta, Ga.

BARNES, Hugh S. (*J* 1940) Major, U. S. Army, and • 23 Mastick St., Savannah, Ga.

BARNES, John C. (*A* 1944) Br. Mgr., • National Radiator Co., 4034 Georgia Ave. N.W., Washington 11, D. C., and 4510 Harling Lane, Bethesda 14, Md.

BARNES, Lewis L. (*M* 1943; *A* 1942; *J* 1937) Air Cond. Engr., Carrier Atlanta Corp., 306 Peachtree St., and • 3995 N. Stratford Rd., Atlanta, Ga.

BARNES, Raymond W. (*M* 1939) Cons. Engr., 631 Moore Bldg., San Antonio 5, and • 1208 N. Main Ave., San Antonio 2, Texas.

BARNES, Walter E. (*M* 1933) Pres., Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain 30, Boston, and • 7 Woodlawn Ave., Wellesley Hills 82, Mass.

BARNETT, Harry (*M* 1942) Chief Engr., The Powers Regulator Co., 2720 Greenview Ave., Chicago, and • 923 Vernon Ave., Glencoe, Ill.

BARNETT, Robert E. (*A* 1944) Engr. in charge of Ventilation, L. D. Smith Shipbuilding Co., Sturgeon Bay, and • Hotel Karsten, Kewanee, Wis.

BARNEY, William E. (*M* 1936) Mgr., • Hydraulic Press Brick Co., South Park, and 4929 East 108th St., Cleveland, Ohio.

BARNUM, Willis E., Jr. (*M* 1933; *J* 1930) • 7048 52nd Ave. N.E., Seattle 5, Wash.

BARR, George W. (*Life Member*; *M* 1905) (Board of Governors, 1910) Dist. Mgr., • Acrofin Corp., 2030 Land Title Bldg., Philadelphia 10, and Woods End, Villa Nova, Pa.

BARRETT, Campbell M. (*A* 1941) Capt., Canadian Army, 5 C. I. B., H. S. R. C. O. C., C. A. O., and • Newton, Ont., Canada.

BARRY, Wilfred W. (*A* 1945) Supvsr., • Barry Sheet Metal Co., Ltd., 10171-98th St., and 10132 142 St., Edmonton, Alta., Canada.

BARTELS, Charles J. (*M* 1942) Owner, • Automatic Stoker & Engineering Co., 207-8 Richardson Bldg., and 1416 Washington Ave., Parkersburg, W. Va.

BARTELS, Everett M. (*A* 1941; *J* 1939) Supvsr. of Mech. Equip., Independent School District, 629 Third St., and • 1104 E. Douglas, Des Moines, Iowa.

BARTH, Herbert E. (*M* 1920) Vice-Pres., • American Blower Corp., P. O. Box 58, Roosevelt Park Annex, Detroit 32, and 15 East Kirby Ave., Detroit 2, Mich.

BARTH, John W. (*A* 1943; *J* 1939) • 234 Lime Ave., Long Beach 2, Calif.

BARTHELMESS, Richard (*M* 1944) Mgr., • Richard-Barthelmess Sales Co., 1711 Challen Ave., Jacksonville 5, Fla.

BARTLETT, Amos C. (*M* 1940) Mgr., Htg. & Vent. Dept., • B. F. Sturtevant Co., Hyde Park, Boston 36, and 22 Weston Ave., Braintree 84, Mass.

BARTLETT, C. Edwin (*M* 1922) Pres., • Bartlett & Co., Inc., 201 South 42nd St., Philadelphia 4, and 3111 W. Coulter St., Philadelphia 29, Pa.

BARTON, Fred C. (*M* 1943) Mech. Engr., • Bevington-Taggart & Fowler, 730 K of P Bldg., Indianapolis, Ind.

BARTON, Jay (*M* 1937) Partner, Barton Engineering Co., 416 Stephenson Bldg., Detroit, Mich., and • 808 Holgate Ave., Defiance, Ohio.

BARTON, Kenneth L. (*M* 1945) Indus. Engr., • The Foxboro Co., 420 Lexington Ave., Rm. 1643, New York 17, and 103 Ramsey Ave., Yonkers 2, N. Y.

BARTON, Lawrence R. (*A* 1944) Sales Engr., Hayes Brothers, Inc., 236 W. Vermont St., and • 3055 Ralston Ave., Indianapolis, Ind.

BASHAK, Cihat (*J* 1943) • Istanbul, Turkey.

BASSEVITCH, Abraham (*A* 1945) Design Engr., Meyer, Strong & Jones, 101 Park Ave., New York, and • 3002 West 28th St., Brooklyn 24, N. Y.

BASTEDO, Albert E. (*M* 1919) Vice-Pres., Treas. & Gen. Mgr., • Burnham Boiler Corp., Irvington, and 55 Burnside Dr., Hastings-on-Hudson 6, N. Y.

BASTEDO, George R. (*A* 1942; *J* 1937) Lt. (j.g.), U. S. N. R., and • 102-36 86th Rd., Richmond Hill 18, L. I.

BATEMAN, William Hamilton (*A* 1944) Partner, • Keystone Sales Co., 547 Lincoln Ave., P. O. Box 4201, Bellevue Sta., and 1011 California Ave., Avalon, Pittsburgh 2, Pa.

BATTES, John H. (*J* 1941; *S* 1939) U. S. Army, and • 313 W. Lehigh, Philadelphia, Pa.

BATTAN, S. W. (*M* 1940) Engr., • Battan & 9 Pennsylvania Ave., Avondale, and • Kennett Square, R. D., Pa.

BAUER, Albert E. (*M* 1935) Mgr., Spec. Prod. Div., Stainless & Steel Products Co., 1000 Berry Ave., St. Paul 4, and • 59 S. Victoria St., St. Paul 5, Minn.

BAUER, Ray F. (*M* 1945) Engr., J. M. O'Connor Co., 208 N. Waco, and • 225 N. Rutan, Wichita, Kans.

BAUM, Albert L. (*M* 1916) Partner, • Jaros, Baum & Bolles, 415 Lexington Ave., New York 17, and 600 West 111th St., New York 25, N. Y.

BAUMGARDNER, C. M. (*M* 1928) Vice-Pres., Sales, • U. S. Radiator Corp., 1500 United Artists Bldg., Detroit 31, Mich., and 416 Cumnor Rd., Kenilworth, Ill.

BAUTTS, Ray A. (*M* 1945) Local Mgr., Jonesboro Dist., • Arkansas Power & Light Co., Box 812, and 1005 W. Oak Ave., Jonesboro, Ark.

BAWDEN, Leslie A. (*A* 1945) Midland Area Engr., Spirax Manufacturing Co., Ltd., Beechmount, Cheltenham, Gloucester, and • 38 Windsor Rd., Evesham, Worcestershire, England.

BAXTER, A. G. (*M* 1944) Htg. Engr., • Enterprise Foundry Co., Ltd., Sackville, N. B., and 28 Clarence St., Amherst, Nova Scotia, Canada.

BAXTER, Henry A. (*A* 1944) Vice-Pres. & Sales Mgr., • W. H. Cunningham & Hill, Ltd., 269-271 Richmond St. W., and 93 Castlewood Rd., Toronto, Ont., Canada.

BAXTER, William E. (*A* 1939) Pres., • W. E. Baxter, Ltd., 2200 Hingston Ave., Montreal, and 89-51st Ave., Lachine, Que., Canada.

BAY, Charles H. (*A* 1938) Sales Repr., • The Detroit Edison Co., 2000 Second Ave., Detroit, and Woodward Ave., Bloomfield Hills, Mich.

BAZILLE, Clarence H. (*M* 1944) Mgr., Htg. Dept., Harry Lee Plumbing & Heating, 216 California Dr., Burlingame, and ● 835 Turk St., San Francisco 1, Calif.

BAZZONI, Joseph P. (*A* 1942) Mech. Engr., Goodyear Aircraft Corp., Plant B, Dept. 110, Akron, Ohio, and ● R. 2, Ottawa, Ill.

BEACH, Albert F., Jr. (*M* 1944) Application Engr., Carrier Corp., 405 Lexington Ave., New York, N. Y., and ● 214 Columbia Ave., Cranford, N. J.

BEACH, Ralph L. (*M* 1942) Dist. Engr., York Corp., 412 Houston St. N.E., P. O. Box 2210, Atlanta, and ● 131 Clarion Ave., Decatur, Ga.

BEACH, Walter R. (*A* 1936) Field Engr., Gates Rubber Co., 2240 E. Washington Blvd., Los Angeles 21, and ● 5332 Appian Way, Long Beach 3, Calif.

BEAGHEN, George W. (*A* 1944) Field Engr., ● American Blower Corp., 50 West 40th St., New York 18, and 3044 Park Dr., Pelham Manor, N. Y.

BEAIRD, Benjamin J. (*M* 1945) Chief Engr. and Estimator, Charles G. Heyne & Co., 2002 Rothwell, and ● 1621 Kipling, Houston, Texas.

BEALS, Albert P. (*A* 1945) Gen. Mgr., ● Western Service Co., 910 Front St., and 1220 Broadway, Fargo, N. D.

BEALS, Dowell E. (*M* 1941; *J* 1940) Owner, ● Beals Plumbing & Heating Co., 315 West 13th St., and 4001 Calmont St., Fort Worth, Texas.

BEAN, George S. (*A* 1935) Mgr., Stoker Div., ● North Western-Hanna Fuel Co., 2196 University Ave., St. Paul 4, and 4949–16th Ave., Minneapolis 7, Minn.

BEAN, Thomas H., Jr. (*M* 1944) Cons. Engr., ● 813 Hidalgo St., New Orleans 19, La.

BEARG, Milton J. (*M* 1944) Air Cond. Design Engr., Ansco, Charles St., and ● 122 Matthews St., Binghamton, N. Y.

BEARMAN, Alexander A. (*M* 1937) Engr., ● 20th Century-Fox Film Corp., 444 West 56th St., New York 19, and 3111 Broadway, New York 27, N. Y.

BEARSE, Allen H., Jr. (*A* 1945) Mgr., Solid Fuel Div. and Comb. and Htg. Eng., ● Barrows Coal Co., 35 Main St., and 56 Western Ave., Brattleboro, Vt.

BEATTIE, James (*A* 1940) Htg. Contr., 10001 Broad St., Detroit 4, and ● 17215 Greenlawn Ave., Detroit 21, Mich.

BEATTY, John W. (*J* 1944; *S* 1941) U. S. Navy, ● 612 State St., Beardstown, Ill.

BEAUCHAMP, John H. (*A* 1945) Owner-Mgr., ● Heating & Refrigeration Maintenance Co., 3913 S. Main St., and 1005 Isabella, Apt. 33, Houston 4, Texas.

BEAURRIENNE, Auguste* (*Life Member*; *M* 1912) Cons. Engr., 25 Rue des Marguettes, Paris X11, and ● 18 Rue du petit Val, Sucy en Brie, Seine et Oise, France.

BEAVERS, G. R. (*M* 1929) Chief Engr., Canadian Blower & Forge Co., Ltd., Woodside Ave. and ● 230 Cameron St. N., Kitchener, Ont., Canada.

BECHTOL, J. J. (*A* 1942; *J* 1937) U. S. Navy, Pensacola, Fla., and ● 1282 Quebec Rd., Cincinnati 5, Ohio.

BECKER, C. S. (*M* 1939) Mech. Engr., ● American Blower Corp., 228 N. LaSalle St., Chicago, Ill.

BECKER, George E. (*A* 1943) Secy.-Treas., Sales and Service, ● Frigid Refrigerator Service, 3232 Olive, St. Louis 3, and 5329 Delor St., St. Louis, Mo.

BECKER, Roger K. (*M* 1938) Asst. Gen. Mgr., Ohio Valley Hardware & Roofing Co., Evansville 2, and ● 1215 Akin Dr., Evansville 13, Ind.

BECKER, Roy F. J. (*A* 1943) Pres., ● Oil Burner Service Co., 315 West Lake St., and 3917 Garfield Ave. S., Minneapolis, Minn.

BECKER, Sidney (*J* 1944) Engr., ● J. Becker, Inc., 282 Villeneuve St. W., Montreal 8, Que., Canada.

BECKWITH, F. J. (*M* 1940) Htg. and Vtg. Engr., Henley & Beckwith, 2628 Pearl St., and ● R. F. D. 1, Box 250, Jacksonville, Fla.

BEEBE, Frederick E. W. (*Life Member*; *A* 1915) Retired, Johnson Service Co., 28 East 29th St., New York, N. Y., and ● 20 Denman Pl., Elizabeth 3, N. J.

BEERS, Louis N. (*A* 1942) Sales Engr., Nicolai's Tucson Pipe Works, 416 W. Davis, and ● 2745 N. Martin, Tucson, Ariz.

BEERY, Clinton E.* (*M* 1936) Pres., ● Heat & Fuel Engineering Co., Inc., 1454 Hood Ave., Chicago, Ill.

BEERY, Vernon L. (*A* 1944) Sales Engr., Heating & Ventilating, and ● Lake Delton, Wis.

BEGGS, William E. (*M* 1927) Owner, W. E. Beggs Co., 234 Ninth Ave. N., and ● 2580 W. Viewpoint Way, Seattle, Wash.

BEGOON, George F. (*A* 1945) Mgr., Elec. Air Cleaning Div., Thermix Engineering Co., First National Bank Bldg., Greenwich, Conn.

BEIGHEL, H. A. (*A* 1927) Owner, ● Allegheny Engineering Co., 503 Columbia Bldg., Pittsburgh, and 207 Puritan Rd., Rosslyn Farms, Carnegie, Pa.

BEIRN, John B. (*A* 1945) Sales Engr., American Radiator & Standard Sanitary Corp., Pittsburgh, Pa., and ● 67 Texel Dr., Springfield 8, Mass.

BEISEL, Karl E. (*J* 1944) Asso. Naval Archt., Vent. Sect., Norfolk Navy Yard, Hull and Aero Drafting Office, and ● 813 Western Branch Blvd., Green Acres, Portsmouth, Va.

BEITZELL, Albert E. (*A* 1933; *J* 1930) Vice-Pres., Combustioneer Corp., 409 Tenth St. S.W., Washington, D. C., and ● 6701 N. Central Ave., Chevy Chase, Md.

BEKESHUS, J. Herbert (*M* 1945) Pres., ● Pioneer Oil Heat, Inc., 229 River St., and 1503 Massachusetts Ave., Troy, N. Y.

BELANGER, William (*M* 1944) Owner, ● Southeastern Supply Co., 390 N. Front St., and 4409 Acushnet Ave., New Bedford, Mass.

BELDEN, Albert G., Jr. (*M* 1944) Pres., ● Refrigeration & Appliance Service, Inc., 15 South 13th St., and 2700 Wilson Dr., Terre Haute, Ind.

BELFORD, Louis B. (*A* 1940) Eastern Supvsr., Air Cond. Control Div., ● Minneapolis-Honeywell Regulator Co., 221 Fourth Ave., New York 3, and 295 Murray Ave., Larchmont, N. Y.

BELINE, Martin B. (*M* 1944) Engr., Washington Refrigeration Co., 1733 14th St. N.W., and ● 2440 16th St. N.W., Washington 9, D. C.

BELING, Earl H.* (*M* 1936; *A* 1930; *J* 1925) Owner, Beling Engineering Co., 405 State Trust Bldg., and ● 2428–13th St., Moline, Ill.

BELL, E. Floyd (*M* 1933) Gen. Mgr., ● Bell & Eiss, Inc., 2102 Foshay Tower, Minneapolis 2, and 5224 Oak Lawn Ave., Edina, Minneapolis 10, Minn.

BELL, John A. (*A* 1944) Htg. Foreman, Royal Canadian Air Force, No. 2, Air Command, RCAF Winnipeg, and ● 151 Baltimore Rd., Winnipeg, Man., Canada.

BELL, Sydney R. (*M* 1939) Principal, ● Sydney R. Bell & Associates, 374 Little Collins St., and 12 Queens Rd., Melbourne, Australia.

BELL, Thomas H. (*A* 1944) Engr. and Supt. of Constr., Columbus Heating & Ventilating Co., 182 N. Vale Ave., and ● 296 S. Eureka Ave., Columbus, Ohio.

BELLMAN, John V. (*A* 1946; *J* 1943) Chief Petty Officer, Directorate of Electrical Supply, Naval Service Headquarters, Ottawa, and ● 170 Hope St., Toronto, Ont., Canada.

BELSKY, George A. (*M* 1945) Chief Engr., Majestic Refrigerator Corp., 625 Broadway, New York, and ● 78 Randolph Rd., White Plains, N. Y.

BELYEA, Roy E. (*A* 1944) Mgr., ● Bélyea Bros., Ltd., 1002 Bathurst St., and 207 Glenayr Rd., Toronto, Ont., Canada.

BEMAN, Myron C. (*M* 1926) (Council, 1934–1939) Partner, ● Beman & Candee, 374 Delaware Ave., Buffalo 2, and 262 E. Quaker St., Orchard Park, N. Y.

BEMARKT, George H. (*M* 1945) Htg. Engr., ● N. O. Nelson Co., 4316 Duncan Ave., St. Louis 10, Mo. and 516 North 19th St., East St. Louis, Ill.

BEMIS, Paul D. (*M* 1942) Cons. Engr., ● 36 Pearl St., Hartford 3, and 31 Riggs Ave., West Hartford, Conn.

BEMIS, Wilfred A. (*A* 1942) Engr., American Smelting & Refining Co., 4039 Park Ave., St. Louis, and ● 129 Emerling Dr., Ferguson 21, Mo.

BENHAM, Colin S. K. (*A* 1940, *J* 1937) Dir., ● Benham & Sons, Ltd., 66 Wigmore St., London W. 1, and Hill House, Prestwood, Bucks, England.

BENHAM, F. C., Jr. (*A* 1942; *J* 1938) Engr., ● C. H. Ruebeck Co., 810 Insurance Bldg., San Antonio 5, and 1636 W. Summit St., San Antonio, Texas.

BENNETT, Edwin A. (*M* 1936; *J* 1929) Field Engr., ● American Blower Corp., 50 West 40th St., New York 18, and 128 Randolph Rd., Fulton Park, White Plains, N. Y.

BENNETT, Merle F. (*A* 1942) Chief Engr., ● First National Bank Bldg. Co., 1740 National Bank Bldg., Detroit 26, and 537 Pleasant Ave., Birmingham, Mich.

BENNIS, Raymond R. (*A* 1943) Chief Engr., ● Contractors Refrigeration Corp., 89–35 Queens Blvd., Elmhurst, and 67-47 181st St., Flushing, N.Y.

BENOIST, LeRoy L. (*M* 1934) Partner and Mgr., ● Benoist Bros. Supply Co., 117 S. Tenth St., and 1500 Main St., Mt. Vernon, Ill.

BENOIST, Raymond E. (*A* 1936) Mgr. and Engr., Benoist Bros. Supply Co., 117 S. Tenth St., and ● 811 North 12th St., Mt. Vernon, Ill.

BENSEN, Clarence L. (*M* 1939; *J* 1935) Sales Engr., McQuay, Inc., 1600 Broadway N.E., Minneapolis, and ● 3042 Benjamin St. N.E., Minneapolis 13, Minn.

BENSINGER, Mark (*A* 1946; *J* 1936) Lt. (j.g.) Sub. Repair Unit, ● c/o Postmaster, San Diego, Calif., and 608 Bradley Lane, Fairview Heights, Portsmouth, Va.

BENSON, Foster W. (*M* 1942) Asst. Western Mgr., Niagara Blower Co., 37 W. Van Buren St., Chicago, and ● 418 Emory Ave., Elmhurst, Ill.

BENSON, John C. (*A* 1945) Washington Branch Mgr., ● Carrier Corp., 942 Investment Bldg., Washington 5, D. C., and 103 Southbrook Lane, Bethesda 14, Md.

BENSON, M. A. (*M* 1944) Branch Mgr., ● Johnson Service Co., 2708 Live Oak, Dallas 1, Texas.

BENTLEY, Clyde E. (*M* 1937) Cons. Engr., ● 216 Pine St., San Francisco 4, and 1875 San Antonio Ave., Berkeley, Calif.

BERGAN, John R. (*M* 1945; *A* 1941; *J* 1937) Eastern Sales Mgr., Moduflow Div., ● Minneapolis-Honeywell Regulator Co., 221 Fourth Ave., New York 3, and 293 Murray Ave., Larchmont, N. Y.

BERGER, J. L. (*M* 1939) Pres., ● The W. R. Rhoton Co., 5915 Bonna Ave., Cleveland 3, and 2652 Edgerton Rd., University Heights, Cleveland 18, Ohio.

BERGMAN, A. E. (*A* 1943) Pres., ● A. E. Bergman, Inc., 1913 University Ave., St. Paul, and 2744 W. River Rd., Minneapolis, Minn.

BERKELEY, Clarence D. (*M* 1944) Application Engr., Carrier Corp., 1331 Third Avenue Bldg., Seattle 1, and ● 3312 E. Spring St., Seattle 22, Wash.

BERLET, E. John, Jr. (*M* 1943) Application Engr., ● York Corp., 5051 Santa Fe Ave., Los Angeles 11, Calif.

BERMAN, Louis K. (*M* 1908) Pres., ● Raisler Corp., 129 Amsterdam Ave., New York, and 885 Park Ave., New York 21, N. Y.

BERMEL, Alfred H. (*M* 1943; *A* 1933; *J* 1928) Chief Mech. Estimating Engr., ● August Arace & Sons, Inc., 642 Third Ave., Elizabeth, and 340 Cambridge Dr., Union, N. J.

BERNARD, Edgar L. (*J* 1941; *S* 1940) ● 40 Chester St., Allston 34, Boston, Mass.

BERNHARD, George (*M* 1944) ● 1175 E. Broadway, Hewlett, N. Y.

BERNSTROM, C. Bertil (*M* 1944) Chief Engr., Jaden Manufacturing Co., and ● 223 W. Fifth St., Hastings, Nebr.

BERNZEN, John (*A* 1944) Vice-Pres., ● The City Plumbing & Heating Co., 1123 Walnut, P. O. Box 71, and 833 Spruce, Boulder, Colo.

BERRES, Daniel S. (*A* 1943) Engr., Navy Department, Washington, D. C., and ● 805 N. Wayne St., Arlington, Va.

BERRIDGE, Winston W. (*M* 1938) Htg. Engr., ● McColl-Frontenac Oil Co., Ltd., 2540 Notre Dame, E., Montreal, and 28 Dufferin Rd., Hampstead, Que., Canada.

BERRY, Norton E. (*M*, 1944) Dir. of Research, ● Servel, Inc., Evansville 20, and Newburgh, Ind.

BERRY, Patrick M. (*A* 1944) Chief Engr., ● Standard Asbestos Manufacturing Co., 1844 East 40th St., Cleveland 3, Ohio.

BERRY, Robert U. (*M* 1939) Mgr., Field Engrg. Div., ● Air Cond. Dept., General Electric Co., 5 Lawrence St., Bloomfield, and 548 N. Maple Ave., East Orange, N. J.

BERRYMAN, Richard H. (*A* 1945; *J* 1940) Reg. Mech. Engr.-Plant Engrg. Div., Aircraft Div., Packard Motor Car Co., 1580 E. Grand Blvd., and ● 264 Worcester Pl., Detroit, Mich.

BERTOLETTE, Chester (*M* 1940) Htg. and Vtg. Engr., ● 515 Broadway, Dobbs Ferry, N. Y.

BERTRAND, George F. (*A* 1939) Sales Engr., ● 235 Richfield Rd., Upper Darby, Pa.

BERZELIUS, Carl E. (*M* 1936) Asst. Mdse. Mgr. for Indus. Roofs, ● U. S. Gypsum Co., 300 W. Adams, Chicago, and 1519 Vine, Park Ridge, Ill.

BEST, E. Thompson (*M* 1944) Mgr., Air Cond. and Refrig., Frank A. McBride Co., 160 Ward St., Paterson, and ● 189 Lakewood Ave., Ho-Ho-Kus, N. J.

BETTS, Howard M. (*M* 1927) Sr. Htg. Engr., ● Dept. of Bldgs., City of Minneapolis, 213 City Hall, Minneapolis 15, and 4923 S. Russell Ave., Minneapolis 10, Minn.

BETZ, Harry D. (*M* 1928) Owner, Betz Engineering Co., 1330 Broadway, Kansas City 6, and ● 5414 Aberdeen Rd., Kansas City 3, Kans.

BEVIL, Alexander T. (*A* 1944) Chief Engr., W. F. Slater Engineering Corp., 627 Monroe Ave., Memphis 3, and ● 1521 Waverly Ave., Memphis 6, Tenn.

BEVINGTON, Curtis H. (*M* 1936) Owner, ● C.H. Bevington Co., 330 S. Wells St., Chicago 6, and R. F. D. No. 2, Elmhurst, Ill.

BEVINGTON, Warren C. (*M* 1942) Pres., ● Bevington-Taggart & Fowler, Inc., 730 Indiana Pythian Bldg., Indianapolis 4, and 3101 N. Meridian St., Indianapolis 8, Ind.

BIANCULLI, Vincent A. (*A* 1945) Engr., Syska & Hennessy, 144 East 39th St., New York, and ● 141-35 78th Rd., Flushing, N. Y.

BIBER, Herbert A. (*A* 1937) Engr., Mellon National Bank, 514 Smithfield St., Pittsburgh, and ● 323 Barnes St., Wilkinsburg, Pittsburgh 21, Pa.

BIBLE, Hollis U. (*M* 1940) Lt. (j.g.), U. S. N. R., N. T. Sch., Princeton, Brown Hall 113, Princeton, N. J., and ● c/o Mr. Thos. M. Godfrey, Box 322, Rt. 12, Memorial Dr., Houston, Texas.

BICHOWSKY, F. Russell (*M* 1935) Prof., ● 4200 13th Pl. N.E., Washington, D. C.

BICKELL, George C. (*A* 1945) Sales Engr., ● Tutein Equipment Co., 130 Bradford Ave., Pittsburgh 5, and 1100 Standhope, Pittsburgh 4, Pa.

BIERL, Joseph G. (*M* 1945) Vice-Pres., Schollman Bros. Co., 4114 North 24th St., and ● 4927 N. 34 Ave., Omaha 11, Nebr.

BIGELOW, Folger H. (*A* 1944) Branch Mgr., ● Ilg Electric Ventilating Co., 517 Farnsworth Bldg., Memphis 3, and 353 N. Willett St., Memphis 12, Tenn.

BIGGERS, Richmond H. (*A* 1939) Mfrs. Agent, ● 2229 E. Jefferson Ave., Detroit 7, Mich.

BILDERBACK, George R. (*A* 1945) Comm'l. Supvsr., Advanced Refrigeration, Inc., 350 Peachtree N.E., and ● 1975 Baker Rd. N.W., Atlanta, Ga.

BILLINGSLEY, Oliver F. H., II (*A* 1944; *J* 1937) Pvt., U. S. A., Serial No. 39714096, ● 185th Repl. Co., 43rd Bn., A. P. O. 711, c/o Postmaster, San Francisco, and 2172 West 20th St., Long Beach, Calif.

BINDER, Charles G. (*M* 1920) Mgr., Htg. Dept., Warren Webster & Co., 17th and Federal St., Camden, and ● 115 Oak Terrace, Merchantville, N. J.

BIRD, Charles (*A* 1934) Treas. and Gen. Mgr., ● The Cincinnati Supply Co., 450-456 E. Pearl St., Cincinnati, Ohio.

BIRKETT, Harold S. (*M* 1940) Engr., ● The Brooklyn Union Gas Co., 176 Remsen St., Brooklyn 2, and 87 Deepwood Rd., Roslyn Heights, N.Y.

BIRNER, Ira L. (*M* 1945) Mgr., Tech. Sales & Service Dept., ● The Celotex Corp., 120 S. LaSalle St., Chicago 3, and 6056 Marine Dr., Chicago 40, Ill.

BISBEE, Bertin A. (*A* 1945) Partner, Smith & Bisbee Hardware Co., 412-416 Main St., and ● 212 W. First St. N., Marshall, Minn.

BISHOP, Albert H. (*M* 1944) Elec. Supt., ● Alberta Clay Products, and Suite 34, Granada Apt., Medicine Hat, Alta., Canada.

BISHOP, Charles R. (*Life Member*; *M* 1901) (Council, 1916) ● 22 Sagamore Rd., Bronxville, N. Y.

BISHOP, Frederick R. (*M* 1921) Mgr. of Sales, Blower Div., The Brundage Co., Kalamazoo, and ● 8011 Dexter Blvd., Detroit 6, Mich.

BISHOP, Frederick W. (*A* 1945) Asst. Mgr., Crane, Ltd., Hill O'Chips, St. John's, Newfoundland, and • 89 St. Clare Ave., St. John's, Newfoundland.

BISHOP, J. A. (*M* 1939) Dist. Mgr., • American Blower Corp., 619 Texas Bank Bldg., Dallas 2, and 1115 N. Windomere St., Dallas 8, Texas.

BISHOP, Joseph W. (*M* 1939) Col., • Asst. Dir., Ordnance Services, Headquarters 5th Canadian Armoured Div., c/o Base P. O., Ottawa, and 62 Highland Crescent, R.R. No. 2, York Mille, Ont., Canada.

BISHOP, M. W. (*M* 1942; *A* 1939; *J* 1935) Branch Mgr., • American Blower Corp., 231 W. Wisconsin Ave., Milwaukee 3, and 4958 N. Cumberland Blvd., Whitefish Bay 11, Wis.

BISPALA, John T. (*A* 1940) Partner, • Bispala Bros., 2328 First Ave., Hibbing, Minn.

BJERKEN, Maurice H. (*M* 1938) N. W. Mgr., • Hoffman Specialty Co., 542 Builders Exch., and 4952-17th Ave. S., Minneapolis, Minn.

BLACK, Edgar N., 3rd (*M* 1922) Philadelphia Mgr., • Fitzgibbons Boiler Co., Inc., 1717 Sansom St., Philadelphia, and 111 Woodside Rd., Haverford, Pa.

BLACK, Harry C. (*M* 1917) Owner, • P. Gormly Co., 155 N. Tenth St., and 927 North 65th St., Philadelphia, Pa.

BLACK, James M. (*A* 1945; *J* 1940; *S* 1939) Engr., Avery Engineering Co., 1906 Euclid Ave., Cleveland 15, and • 3140 East 135th St., Cleveland 20, Ohio.

BLACKHALL, Wilmot R. (*M* 1922) Partner, • Blackhall Heating & Plumbing Supplies, 1104 Bay St., and 832 Waverley Rd., Toronto, Ont., Canada.

BLACKMAN, Alfred O. (*Life Member*; *M* 1911) • Greenwich Lodge, 47 Lafayette Pl., Greenwich, Conn.

BLACKMAN, Robert C. (*A* 1945; *J* 1944) Power Engr., Allison Div., General Motors Corp., and • 1267 N. Lynhurst Dr., Indianapolis 8, Ind.

BLACKMORE, F. H. (*M* 1923) Vice Pres., • U. S. Radiator Corp., 1500 United Artists Bldg., Detroit 31, and 515 Tooting Lane, Birmingham, Mich.

BLACKMORE, Joseph J. (*A* 1939; *J* 1937) Mfrs. Agt., • 4030 Chouteau Ave., St. Louis, Mo., and 312 S. Fillmore St., Edwardsville, Ill.

BLACKSHAW, J. L.* (*M* 1937; *J* 1929) Engr., Air & Refrigeration Corp., 475 Fifth Ave., New York, N. Y., and • 247 W. Mercer Ave., College Park, Ga.

BLACKWELL, J. S. (*A* 1944) Prop., • Air Engineering, 870 Massachusetts Ave., Indianapolis 4, and 5034 Broadway, Indianapolis 5, Ind.

BLADEN, Arthur M. (*M* 1945) Sole Owner, • Neiler, Rich & Bladen, 431 S. Dearborn St., Chicago, and 334 N. Kenilworth Ave., Oak Park, Ill.

BLAIR, Donald W. (*A* 1940) 52 Dunster St., Cambridge 38, Mass.

BLAIR, Ernest L. (*M* 1941) Indus. Engr., Stone & Webster Engineering Corp., 49 Federal St., Boston 7, and • 108 Willow Ave., Wollaston 70, Mass.

BLAIR, Howard A. (*A* 1945) Service Mgr., • B. F. Sturtevant Co., Div. of Westinghouse, Hyde Park, Boston 36, and 172 Lindbergh Ave., Needham Heights 94, Mass.

BLAKE, Albert H. (*M* 1943) Gen. Mgr., • B. F. Sturtevant Co. of Canada, Ltd., 137 Wellington St. W., and 5 Glenroy Ave., Toronto, Ont., Canada.

BLAKE, John L. (*A* 1943) Chief Engr., Air Cond. Dept., M. G. M. Studios, 10202 Washington Blvd., Culver City, and • 1700 S. Bedford St., Los Angeles 35, Calif.

BLAKELEY, Hugh J. (*M* 1935) Partner, • Hubbard, Rickerd & Blakeley, 110 Whitney Ave., New Haven 10, and 24 Bradley Ave., East Haven, Conn.

BLAKER, Alfred H. (*A* 1939) Secy.-Treas., • National Korectaire Co., 1619 Cortland St., Chicago 22, Ill.

BLANCHARD, Norris M. (*M* 1942) Western Sales Mgr., • L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., Milwaukee 7, and 2102 N. Hi Mount Blvd., Milwaukee 8, Wis.

BLANDING, Robert L. (*M* 1938) Vice Pres. & Gen. Mgr., • Taco Heaters, Inc., 123 South St., and 1385 Smith St., Providence, R. I.

BLANKIN, Merrill F. (*M* 1927; *A* 1926; *J* 1919) (*Presidential Member*), (Pres., 1943; 1st Vice Pres., 1942; Treas., 1939-41), (Council, 1939-44) (Pres., • Haynes-Blankin Corp., S. E. Cor. Ridge Ave. and Spring Garden St., Philadelphia 23, and 528 E. Gates St., Roxborough, Philadelphia, Pa.

BLAS, Romualdo J. (*M* 1936) Refrig. Engr., • American Embassy, Food Supply Mission, Institute-Inter-American Affairs, and Apartado 1944, Caracas, Venezuela, S. A.

BLAYNEY, W. Ronald (*A* 1939) Owner, • W. B. Graves Heating Co., 1144 W. Chicago Ave., Chicago 22, and 4327 N. Monticello Ave., Chicago 18, Ill.

BLAZER, Benjamin V. (*A* 1940) Owner, • M. Blazer & Son, 173 Market St., Passaic, and 48-13th Ave., Paterson, N. J.

BLEIER, Frank P. (*M* 1945; *A* 1944) Dir. of Research, Ilg Electric Ventilating Co., 2850 N. Crawford, Chicago 41, and • 547 W. Addison St., Chicago 13, Ill.

BLOCH, Leonard L. (*A* 1944) • General Delivery, Los Angeles, Calif.

BLOOM, Louis (*M* 1935) Owner, • Freeport Plumbing & Heating Engrs., 84-A Broadway, Freeport, L. I., N. Y.

BLOOM, Ralph L. (*M* 1945) Mech. Engr., • 623 Sexton Bldg., Minneapolis 15, and 3754 Joppa Ave., Minneapolis 16, Minn.

BLOOMSTER, Edgar L. (*M* 1943) Cons. Engr., • 116 New Montgomery St., San Francisco 5, and 3820 Divisadero St., San Francisco, Calif.

BLUM, Herman, Jr. (*A* 1944; *J* 1936) Cons. Engr., • 1708½ Commerce St., Dallas 1, and 4438 Emerson, Dallas 5, Texas.

BLUM, Richard J., Jr. (*A* 1940) • c/o E. A. Carsey, The Kirk & Blum Mfg. Co., 2850 Spring Grove Ave., Cincinnati 25, Ohio.

BLUMENTHAL, M. I. (*M* 1936) Mech. & Refrig. Engr., Maritime Commission, West Coast Reg. Office, Financial Center Bldg., U. S. M. C., and • 3600 Lakeshore Blvd., Oakland 10, Calif.

BOALES, William G. (*M* 1936; *A* 1923) Owner, • Wm. G. Boales & Assoc., 6429 Hamilton Ave., Detroit 2, and 43 Edgemere Rd., Grosse Pointe Farms 30, Mich.

BOCK, L. I. (*A* 1944) Owner, • Carrier-Bock Co., 708 N. Harwood, Dallas 1, and 4654 N. Versailles, Dallas, Texas.

BODEN, Walter F. (*A* 1937) Branch Mgr., • Modine Manufacturing Co., 424 E. Wells St., Milwaukee 2, and 4472 S. Lake Dr., Cudahy, Wis.

BODINGER, J. H. (*M* 1931) Pres., • J. H. Bodinger Co., Inc., 530 Tenth Ave., New York 18, N. Y.

BODMER, Emmanuel (*M* 1937) Tech. Dir., Ste. Indle de Chauffage Rationnel et d'Applications Mecaniques, 180 Rue de Vaugirard, and • 1 Rue Leon Seche, Paris 15, France.

BODWELL, George B. (*M* 1944) Asst. Dist. Mgr., • The Philip Carey Manufacturing Co., 5906 Euclid Ave., Cleveland, and 3851 Parkdale Rd., Cleveland Heights, Ohio.

BOEDDENER, George (*A* 1946) Managing Dir., • National Warm Air Heating & Air Conditioning Assn., Society for Savings Bldg., Cleveland 14, and 19832 Roslyn Dr., Rocky River, Ohio.

BOEHLER, Carl L. (*A* 1944) Sales Mgr., Dealer Sales, • J. A. Walsh & Co., Inc., 3215 McKinney, P. O. Box 1773, Houston 1, and Rt. 12, Box 700, Houston, Texas.

BOEHM, John C., Jr. (*A* 1945) Engr., • The John C. Boehm Co., 9002 Madison Ave., Cleveland 2, and 10909 Clifton Blvd., Cleveland, Ohio.

BOELTER, L. M. K. (*M* 1944) Dean, • College of Engineering, University of California, and 254 Bronwood Ave., Los Angeles 24, Calif.

BOESTER, Carl F.* (*M* 1939; *A* 1936) Housing Research Exec., • Purdue Research Foundation, Purdue University, Lafayette, and Union Club, West Lafayette, Ind.

BOGATY, Hermann S. (*M* 1921) • 735 E. Phil-Ellena St., Philadelphia, Pa.

BOGEN, Arthur D. (*M* 1945) Mgr., • The Home Furnace Co., 330-34 E. Livingston Ave., Columbus 15, and 2207 Yorkshire Rd., Columbus 8, Ohio.

BOGGS, Dennis (*M* 1943) Asst. Chief Engr., Avery Engineering Co., 1906 Euclid Ave., Cleveland 15, and • 5315 Thomas St., Maple Heights, O.

BOLAND, L. C., Jr. (*M* 1941) Cons. Engr., 774 Spring St. N.W., and • 1140 Rosedale Dr. N.E., Atlanta, Ga.

BOLAND, Roy O. (*A* 1938) • Industrial Steel & Fibre Products, Ltd., 507 Place D'Armes, Rm. 1405, Montreal, and 348 Kensington Ave., Westmount, Que., Canada.

BOLES, Harry S. (*A* 1944) Plant Engr., Mid-State Frozen Egg Corp., 333 West 18th St., Indianapolis, and • 903 E. Maple Rd., Indianapolis 5, Ind.

BOND, Harry H. (*M* 1938) Partner, • Edward E. Ashley, 10 East 40th St., New York 16, and 137-81 Belknap St., Springfield Gardens 13, L. I., N. Y.

BOND, Horace A. (*M* 1930) Prof. Engr., 152 Washington Ave., and • 12 Ramsey Pl., Albany, N. Y.

BONDE, Ira E. (*J* 1945) Vent. Engr., Ind. Design Sect., Navy Yard, Navy No. 128, F. P. O., San Francisco, and • 1057 Fulton St., Palo Alto, Calif.

BONNER, John (*M* 1943) Htg. Engr., • The John Bonner Co., 121 Hanover St., Boston 13, and 27 Park Circle, Arlington 74, Mass.

BOOKER, Jack W. (*M* 1943) Engr., • Westinghouse Electric International Co., 40 Wall St., New York, N. Y., and 407 Elm St., Cranford, N. J.

BOONE, Fred S. (*A* 1943) Secy.-Treas., • Hall-Neal Furnace Co., 1324 N. Capitol Ave., Indianapolis 7, and 17 West 35th St., Indianapolis 8, Ind.

BOOT, Arthur (*M* 1943) Owner, • Arthur Boot Co., 1444 Lake Dr. S.E., and 928 Orchard Ave. S.E., Grand Rapids 6, Mich.

BOOTH, Charles A. (*Life Member*; *M* 1917) Vice Pres., • Buffalo Forge Co., 490 Broadway, and 142 Summit Ave., Buffalo, N. Y.

BOOTH, Clifford A. (*A* 1942) Sales Engr., • Fiberglas Canada, Ltd., 1025 Confederation Bldg., and 4523 Kensington Ave., Montreal, Que., Canada.

BOOTH, Roger W. (*A* 1945) Chief Engr., The Lennox Furnace Co., and • 1516 Kalsem Blvd., Marshalltown, Iowa.

BORAK, Eugene (*M* 1937) Engr., • The Johnson Service Co., 1905 Dunlap St., Cincinnati 14, and 6135 Hammel Ave., Golf Manor, Cincinnati 12, Ohio.

BORG, Elmer H. (*M* 1938) Partner, • Brooks & Borg, 820 Hubbell Bldg., Des Moines 9, and 3101 Easton Blvd., Des Moines, Iowa.

BORKAT, Philip (*A* 1943; *J* 1936) Chief Engr., Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, and • 869 East 128th St., Cleveland 8, Ohio.

BORNEMANN, Walter A. (*M* 1924; *J* 1923) • Carrier Corp., 12 South 12th St., Philadelphia 7, and 123 W. Wharton Ave., Glenside, Pa.

BORNQUIST, George W. (*A* 1943) Pres., • Bornquist, Inc., 629 W. Washington Blvd., Chicago 6, and 517 N. Euclid Ave., Oak Park, Ill.

BORNSTEIN, Alfred B. (*A* 1942) Partner, Wm. Bornstein & Son, 2209 Channing St. N.E., and • 5206-14th St. N.W., Washington, D. C.

BORNSTEIN, William (*M* 1943; *A* 1937) Partner, • William Bornstein & Son, 2209 Channing St. N.E., and 5206-14th Sts. N.W., Washington, D. C.

BORTON, A. Robert (*A* 1943; *J* 1939) Engrg. Dept., • John J. Nesbitt, Inc., State Road and Rhawn St., Philadelphia, and 3332 Decatur St., Holmesburg, Philadelphia 36, Pa.

BOSTROM, H. R. (*A* 1945) Owner, • Bostrom Sheetmetal Works, 1509 Marshall Ave., St. Paul, and 3529-46th Ave. S., Minneapolis, Minn.

BOTELHO, Nanto J. (*A* 1937) Dir. & Chief Engr., • Geibrasil Cia. de. Engenharia, Av. Pres. Wilson 198-12 Andar, P. O. Box 1269, and Av. Copacabana 209, Apt. 803, Rio de Janeiro, Brazil, S. A.

BOTTUM, Edward W. (*A* 1942; *J* 1938) Chief Engr. Skuttle Manufacturing Co., 517 E. Larned, Detroit, and • 19303 Rutherford, Detroit 19, Mich.

BOUCHER, Kent D. (*M* 1944) Mgr., • Carrier Div., United Clay Products Co., 931 Investment Bldg., Washington 5, D. C., and 1002 S. Mansion Dr., Silver Spring, Md.

BOUEY, Angus J. (*A* 1937; *J* 1930) Sales Engr., • The B. F. Sturtevant Co., 681 Market St., San Francisco 5, and 4810 Fulton St., San Francisco, Calif.

BOUILLON, Lincoln (*M* 1933) Prof. Engr., • 426-1411 Fourth Ave., Bldg., and 3220 Sierra Dr., Seattle, Wash.

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BOWEN, Judson P. (*A* 1945) Branch Mgr., • Penn Electric Switch Co., 36 Fifth St. N.W., and 725 Woodward Way, N.W., Atlanta, Ga.

BOWEN, Leroy E. (*A* 1942) Mech. Engr., Federal Public Housing Authority, 760 Market St., San Francisco, and • 1124 (F) Eighth St., Berkeley, Calif.

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BOWERS, A. F. (*A* 1919) Pres., • Industrial Heating & Engineering Co., 828 N. Broadway, Milwaukee 2, and 2853 N. Hackett Ave., Milwaukee, Wis.

BOWERS, Charles L. (*A* 1944) Mgr., E-G Sheet Metal Works, 258 Pryor St. S.W., Atlanta 3, and • 3050 Sylvan Rd., P. O. Box 290, Hapeville, Ga.

BOWIE, Robert (*M* 1945) Mgr., Engrg. Dept., • H. G. Skelton & Co., Box 1140, and 5 Ave. "LeSueur, Sea Point, Capetown, South Africa.

BOWLER, R. W. (*S* 1943) Ensign, U. S. Navy, at Sea, • 2733 France Ave. S., Minneapolis, Minn.

BOWLES, Potter (*A* 1928) Pres., • Hoffman Specialty Co., 1001 York St., and Lawrence Dr., Brendonwood, Indianapolis, Ind.

BOXALL, Frederick (*M* 1937) Air-Cond. Refrig. Engr., John Inglis Co., Ltd., 14 Strachan Ave., and • Windsor Arms Hotel, Toronto, Ont., Canada.

BOYAR, Sidney L. (*A* 1944; *J* 1938) Buyer, Sears, Roebuck & Co., 925 S. Homan Ave., Chicago, and • 9514 S. Hamilton Ave., Chicago 43, Ill.

BOYD, Lyle E. (*M* 1944; *A* 1941) Engr., Natkin & Co., 1800 Baltimore, Kansas City 8, and • 430 West 59th Terrace, Kansas City 2, Mo.

BOYD, Robert L., Jr. (*A* 1946; *J* 1941) 1st Lt., 0-1116048, • Hq. 4th A. F., B. & G., 180 New Montgomery St., San Francisco 6, Calif., and 4017 Coleridge Ave., Houston 5, Texas.

BOYD, Spencer Wallace (*M* 1937) Cons. Engr., • Newcomb & Boyd, 615 Trust Co. of Ga. Bldg., Atlanta 3, and 1505 Fairview Rd., Atlanta, Ga.

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CARPENTER, R. H. (*M.* 1921), (Council, 1930-35) Mgr., New York Office, • Nash Engineering Co., Graybar Bldg., 420 Lexington Ave., New York 17, and 20 Jefferson Ave., White Plains, N. Y.

CARRIER, Earl G. (*M.* 1936; *J* 1929) Branch Mgr., Carrier Corp., 419 Boylston St., Rm. 624, Boston 16, and • 326 Highland Ave., Winchester, Mass.

CARRIER, Willis H. (*Honorary Member; Life Member; M* 1913), (*Presidential Member*) (Pres., 1931; 1st Vice-Pres., 1930; 2nd Vice-Pres., 1929), (Council, 1923-32) • Chairman of the Board, Carrier Corp., 300 S. Geddes St., Syracuse 1, and 2570 Valley Dr., Nedrow, N. Y.

CARROLL, Daniel E. (*A* 1941) Pres., Carroll Sheet Metal Works, Inc., 46-10 70th St., Woodside, and • 216-09 40th Ave., Bayside, L. I., N. Y.

CARROLL, Edgar E. (*A* 1939) Owner, • Kleenair Furnace Co., 5329 N. E. Sandy Blvd., and 2434 Northeast 43rd Ave., Portland 13, Ore.

CARROLL, William M. (*A* 1943; *J* 1938) Gen. Mgr., • Macklanburg Supply Co., 129-133 Northwest 23rd, Oklahoma City, Okla.

CARSEY, E. A. (*M* 1944) Engr., The Kirk & Blum Manufacturing Co., 2850 Spring Grove Ave., Cincinnati 25, and • 359 Howell Ave., Cincinnati 20, Ohio.

CARSON, C. C. (*M.* 1943) Equip. Dvlpt. and Design Sec. 638, U. S. Navy Dept., Washington, and • 6753 Potomac Dr., N. W., Washington 16, D. C.

CARTER, A. Walter (*M* 1940; *J* 1936) Htg. Engr., Therm-O-Rite Products, Ltd., 67 Yonge St., Toronto, Ont., Canada.

CARTER, Chester A. (*M.* 1944) Mech. Engr., Hq. Seventh Service Command, R. & U. Div., and • 3071 South 33rd St., Omaha, Nebr.

CARTER, Doctor (*M* 1934) Cons. Engr., • Hongkong, China.

CARTER, Henry G. (*A* 1944) Mfrs. Agent, • 5505 Branch Ave., Tampa 4, Fla.

CARTER, John H.* (*M* 1936) Partner, Bodine & Carter, 122-A W. Lockwood Ave., and • 710 Key West Ave., Webster Groves 19, Mo.

CARTIER, Marcel E. (*M* 1944) Research Engr., • American Radiator & Standard Sanitary Corp., 675 Bronx River Rd., Yonkers 4, and 1428 Midland Ave., Bronxville, N. Y.

CARY, Edward B. (*M* 1935) Capt., U. S. N., and • 448 Terminal Tower, Cleveland 13, Ohio.

CASE, Delbert V. (*M* 1937) Engrg. Consultant, War Food Administration, 2200 Fidelity Bldg., Kansas City, and • R. R. 1, Hickman Mills, Mo.

CASE, Donald P. (*A* 1944) Engr. & Estimator, • W. C. Wiedenmann & Son, 1820 Harrison St., Kansas City 8, and 8631 Hiawatha Rd., Kansas City 5, Mo.

CASE, Walter G. (*A* 1930) Mgr., Ideal Boilers & Radiators, Ltd., Ideal House, Great Marlborough St., London, W. 1, and • 66 The Ridgeway, Kenton, Harrow, Middlesex, England.

CASEY, Byron L. (*M* 1921) Mgr., • Northern District, Ilg Electric Ventilating Co., 222 N. LaSalle St., Chicago 1, and 404 Vine Ave., Park Ridge, Ill.

CASKEY, Luther H., Jr. (*J* 1941; *S* 1938) Capt., C. E., Asst. Wing Engr., Hq., West Coast Wing, 1502d A. A. F., B. U., P. O. A. T. C., 49 Fourth St., San Francisco 3, Calif., and • 520 W. Stephen St., Martinsburg, W. Va.

CASKEY, Thomas C. (*M* 1943) Owner, Caskey Engineering Co., 1736 First Ave. S., Seattle 4, and • 11642-26th Ave. S., Seattle 88, Wash.

CASSELL, William L. (*M.* 1936) Principal, • William L. Cassell, 912 Baltimore Ave., Kansas City, and R. F. D. 6, Independence, Mo.

CATLETT, W. A. (*A* 1943) Owner, • Catlett Engineers, 1350 Buckner Blvd., and 9602 El-Patio Dr., Dallas 18, Texas.

CATON, Ross R., Jr. (*J* 1945) Pres., Caton Supply Co., 828 W. Vernor Highway, Detroit 1, and • 15395 Linwood, Detroit 21, Mich.

CAUTHORN, A. V. (*A* 1943) Owner, • A. V. Cauthorn Co., 10001 Broadstreet, and 12660 Montevista, Detroit 4, Mich.

CAVANAGH, P. A. (*M.* 1944) Mgr., • The Herman Nelson Corp., 1015 Chestnut St., Philadelphia 7, and 15 Runnemede Ave., Lansdowne, Pa.

CAZIN, Gerald R. (*A* 1945) Co-Owner, • Cazin & Houtz, Box 162, and 1117 Center St., Evanston, Wyo.

CHADWICK, George R. (*M.* 1945) Chief, Htg., Vent. & Air Cond. Br., Air Installations Div., Air Technical Service Command, Wright Field, Area "A", Dayton, and • 355 Morris St., Osborn, Ohio.

CHADWICK, John B. (*M* 1944) Engrg. Repr., Textile Air Systems, Ltd., Bury Road Works, Radcliffe, Nr. Manchester, and • 11 Orville Dr., Levenshulme, Manchester 19, England.

CHALMERS, C. H. (*Life Member; M* 1938) Gen. Mgr., • Chalmers Oil Burner Co., 318 First Ave. N., Minneapolis 1, and 523 Seventh St. S.E., Minneapolis 14, Minn.

CHAMBERS, Fred W. (*M* 1936) Pres., • F. W. Chambers & Co., Ltd., 96 Bloor St. W., Toronto 5, and 55 Glengowan Rd., Toronto 12, Ont., Canada.

CHAMBERS, S. A. (*M* 1944) Vice-Pres. & Secy., • Almirall & Co., Inc., 53 Park Pl., New York 7, N. Y., and 81 Ross Ave., Hackensack, N. J.

CHAMPLIN, Robert C. (*M.* 1945; *A* 1938) Air Cond. Engr., Timken Silent Automatic, 100-400 Clark Ave., Detroit 32, and • 13640 Mendota Ave., Detroit 4, Mich.

CHANDLER, Roy (*A* 1944) Export Mgr., W. H. Wheeler, Inc., 7 East 47th St., New York 17, and • P. O. Box 591, Grand Central Annex, New York, N. Y.

CHANG, John H. (*J* 1945) Jr. Engr., • Carrier Corp., International Div., Syracuse, N. Y., and 1317 Emma St., Honolulu, T. H.

CHANG, Ting Chu (*J* 1945) Student Engr., Carrier Corp., 300 S. Geddes St., and • 1011 Walnut Ave., Syracuse, N. Y.

CHAPIN, C. Graham (*M* 1933) Treas. • Hopson & Chapin Manufacturing Co., 231 State St., and 66 Faire Harbour Pl., New London, Conn.

CHAPIN, Harvey G. (*M* 1935) Capt., U. S. A. A. F., • 883rd Bomb Sq., 500th Bomb. Group, A. P. O. 237, c/o P. M., San Francisco, Calif., and 8352 Maryland Ave., Chicago, Ill.

CHAPMAN, D. Bascom (*M* 1941) Dist. Office, • Clarage Fan Co., 323 Curtis Bldg., 2842 W. Grand Blvd., Detroit 2, and 16713 Blackstone, Detroit 19, Mich.

CHAPMAN, William A., Jr. (*M* 1936) Lt. Comdr., U. S. N. R., Officer in Charge, • Navy Recruiting & Induction, c/o Navy Recruiting Station, P. O. Bldg., and 31 Northview Dr., Indianapolis, Ind.

CHAPPELL, Henry D. (*M* 1942) Mech. & Elec. Engr., • Burroughs Adding Machine Co., 6071 Second Ave., Detroit 32, and 15493 Whitecomb Ave., Detroit 27, Mich.

CHAR, Raghav D. (*M* 1945) Owner, • Industrial & Agricultural Engineering Co., 43 Forbes St., Bombay 1, and "Villa Vyla" 17th Rd., Khar, Bombay 21, India.

CHARLES, Paul L. (*M.* 1938) Mgr. & Owner, • Walsh & Charles, Ltd., 206 Tribune Bldg., and 145 Ash St., Winnipeg, Canada.

CHASE, Arthur M., Jr. (*M.* 1938) Owner, • Production Equipment Co., P. O. Box 895, 316 Keller Bldg., Houston 1, and 3333 Ozark St., Houston 4, Texas.

CHASE, Chauncey L. (*M.* 1931) Cons. Engr., • 15 William St., New York 5, and 222 Chapel Rd., Manhasset, N. Y.

CHASE, Jim C. (*J* 1945) • 1729 Fleming Rd., Louisville 5, Ky.

CHASE, Peter S. (*A* 1940) Owner, • 936 Oak St., and 1167 Ferry St., Eugene, Ore.

CHASE, R. E., Jr. (*A* 1946; *J* 1941) Capt., • 1389th A. A. F., B. U., N. A. W. A. T. C., A. P. O. 856, c/o P. M., New York, N. Y., and 117 N. Tacoma Ave., Tacoma, Wash.

CHASE, Roger E. (*A* 1939) Pres., • R. E. Chase & Co., Inc., Tacoma Bldg., Tacoma 2, and 117 N. Tacoma Ave., Tacoma 3, Wash.

CHATFIELD, Arthur J. (*M* 1943) Htg. & Vent. Engr., Hope's Heating & Lighting, Ltd., 17 Berners St., London, W. 1, and • 30 Woodside Rd., New Malden, Surrey, England.

CHEASLEY, Thomas C. (*M* 1943) Fuel Engr., • Sinclair Coal Co. & Affiliates, 114 West 11th St., Kansas City 6, and 437 East 72nd St., Kansas City 5, Mo.

CHEESEMAN, Evans W. (*A* 1946; *J* 1937; *S* 1934) Lt. Col., U. S. Army, and • 115 Catskill Ave., Pittsburgh 10, Pa.

CHENEVERT, J. G. (*A* 1938) Cons. Engr., • Arthur Surveyer & Co., 1203 Dominion Square Bldg., Montreal 2, and 536 Outremont Ave., Montreal, Que., Canada.

CHERNE, Realto E. (*M* 1938; *J* 1929) Cons. Engr., • 311 Alexander St., Rochester 7, N. Y.

CHERRY, Lester A.* (*M* 1921) Cons. Engr., • Cherry, Cushing & Preble, 361 Delaware Ave., Buffalo 2, and 151 Euclid Ave., Kenmore 17, N. Y.

CHESAREK, Joseph H. (*M* 1944), Chief, Refrig. & Vent. Sec., • Office of Service Command Engineer, Hq. 9th Service Command, Ft. Douglas, and 964 Atkin Ave., Salt Lake City 5, Utah.

CHESTER, F. L.* (*A* 1940) Mgr., • W. G. Chester & Son, 222 Bannatyne Ave. and 26 C Panama Court, Winnipeg, Man., Canada.

CHESTER, Thomas* (*M.* 1917) Cons. Engr., • 700 Seward Ave., Detroit 2, Mich.

CHEWNING, Ray C. (*A* 1945) Asst. J. Donald Kroeker, 618 Park Bldg., and • 6736 N. Wall, Portland, Ore.

CHEYNEY, Charles C. (*A* 1913) Sales Mgr., • Buffalo Forge Co., 490 Broadway, Buffalo 5, and 255 Lincoln Parkway, Buffalo 16, N. Y.

CHILDS, Lewis A. (*M* 1938) Lt. U. S. N. R., and • 330 Harrison Ave., Glenside, Pa.

CHIPMAN, Edward E. (*S* 1943) Lt. (j.g.) U. S. N. R., U. S. S. Litchfield, c/o Fleet Post Office, New York, N. Y., and • 281 Locust Ave., Washington, Pa.

CHOATE, James E. (*M* 1945) Branch Mgr., Mundet Cork Corp., 339 Elizabeth St. N.E., and • 193 Princeton Way, N.E., Atlanta, Ga.

CHRISTENSON, Harry (*A* 1931) Co-Partner, ● Hunter-Prell Co., 15-19 E. Jackson, and 121 Sunset Blvd., Battle Creek, Mich.

CHRISTESEN, Martin C. (*A* 1944) Field Engr., Sarco Co., Inc., 475 Fifth Ave., New York, N. Y., and ● 196 Everett Pl., Englewood, N. J.

CHRISTIE, J. A. (*A* 1943) Mfrs. Agent, ● 60 Front St. W., Toronto 1, and 46 Spencer Ave., Toronto 3, Ont., Canada.

CHRISTIE, Robert W. (*A* 1945) Research Engr., ● United States Testing Co., Inc., 1415 Park Ave., Hoboken, and 722 Hollywood Ave., Ho-Ho-Kus, N. J.

CHRISTIERSON, C. A. (*M* 1945; *A* 1939; *J* 1937) Mgr., ● Air Conditioning & Engineering Co. (Natal), P. O. Box 2421, 91 Smith St., and 316 Cato Rd., Durban, South Africa.

CHRISTMANN, William F. (*A* 1931) Engr., ● Kroeschell Engineering Co., 215 W. Ontario St., and 6551 N. Maplewood Ave., Chicago, Ill.

CHRISTOPHERSEN, Andrew E. (*M* 1935) Board of Education, ● Amundsen High School, 5110 N. Damen Ave., and 2923 N. Kilpatrick Ave., Chicago, Ill.

CHU, Yao-Han (*J* 1945) Student Engr., ● Carrier Corp., 300 S. Geddes St., Syracuse 1, and 1205 Harrison St., Syracuse, N. Y.

CHUKWUEMEKA, Nwankwo (*J* 1945) Student, Carrier Corp., 300 S. Geddes St., Syracuse 1, N. Y.

CHUR, Charles H., Jr. (*A* 1945) Mgr., W. A. Case & Son Manufacturing Co., 115 N. Geddes St., Syracuse 1, and ● 119 Plymouth Dr., Syracuse 6, N. Y.

CHURCH, H. J. (*M* 1922) Mgr., ● Darling Brothers, Ltd., 137 Wellington St. W., Toronto, and 358 Main St. N., Weston, Ont., Canada.

CHURCH, Lloyd M. (*M* 1943) Dist. Mgr., ● Carrier Corp., 12 South 12th St., Philadelphia, and 21 E. Levering Mill Rd., Bala-Cynwyd, Pa.

CHURCH, Richard A. (*J* 1945) Engr., Carrier Corp., S. Geddes St., Syracuse, and ● 134 Oakley Dr., Syracuse 5, N. Y.

CHYTRAUS, Oscar E. (*A* 1945) Owner, Oscar E. Chytraus Co., 328 W. Second S., and ● 1173 South 11th E., Salt Lake City, Utah.

CLAPPERTON, Robert (*A* 1946; *J* 1944) Engr., ● Dominion Rubber Co., Ltd., Engrg. Dept., 550 Papineau St., Montreal, Que., and 25 Chester Ave., Town of Mount Royal, Que., Canada.

CLAPPERTON, T. Clyde (*M* 1944) Vice-Pres., ● Michael Stuart Co., Ltd., 116 Blood St. W., and 68 The Kingsway, Toronto, Ont., Canada.

CLARE, Fulton W. (*M* 1927) Mech. Engr., U. S. Division Engineer, 50 Whitehall St., and ● 935 Plymouth Rd. N.E., Atlanta, Ga.

CLARK, Albert C. (*A* 1939) Capt., U. S. A., 0-272843, and ● 8705 N. Foss Ave., Portland 3, Ore.

CLARK, Allan M. (*M* 1945; *J* 1942) Sales Engr., ● Canadian Blower & Forge Co., Ltd., 1221 Bay St., Rm. 301, Toronto 5, and 11 Langton Ave., Toronto 12, Ont., Canada.

CLARK, Arthur E. (*A* 1945) Engr., ● Grinnell Company of Canada, Ltd., 2440 Dundas St. W., and 49 Mountview Ave., Toronto 9, Ont., Canada.

CLARK, Dwight L. (*A* 1945) Estimator, Puget Sound Naval Shipyard, and ● 10 Schley Blvd., Bremerton, Wash.

CLARK, E. Harold (*M* 1936) Mfrs. Agent, ● 600 Michigan Theatre Bldg., Detroit 26, and 2539 Lakewood, Detroit 15, Mich.

CLARK, Harry E. (*A* 1944) Mfrs. Repr., ● P. O. Box 370, Houston 1, Texas.

CLARK, James R. (*J* 1942) Sgt., A. S. N., 14053050, ● 24th Depot Repair Sq., 24th Air Depot Group, A. P. O. 264, c/o Postmaster, San Francisco, Calif., and 1501 Pecan Ave., Charlotte 4, N. C.

CLARK, L. O. Ray (*M* 1944) Dist. Mgr., ● B. F. Sturtevant Co., 36 Pearl St., and 230 Farmington Ave., Apt. G5, Hartford, Conn.

CLARK, Lynn W. (*M* 1944; *A* 1938) Engr. & Salesman, ● Hall-Neal Furnace Co., 1324 N. Capitol Ave., and 737 West 32nd St., Indianapolis, Ind.

CLARK, Robert J. (*A* 1944) Factory Repr., ● 3032 Seventh Ave. N., St. Petersburg, Fla.

CLARK, Robert L. (*A* 1939) Pres., The Clark Asbestos Co., 1893 East 55th St., Cleveland 3, and ● 927 Caledonia Ave., East Cleveland 12, Ohio.

CLARKE, John H. (*M* 1942; *A* 1941) Asst. Marine Engr., Marinship Corp., Sausalito, and ● R. D. 2, Box 583, Mill Valley, Calif.

CLARKSON, Robert C., Jr. (*M* 1943) Cons. Engr., ● 1006 Edmonds Ave., Drexel Hill, Pa.

CLAY, Wharton (*M* 1939; *A* 1938) Secy., ● National Mineral Wool Assn., 1270 Sixth Ave., Rm. 2906, New York 20, and 127 S. Broadway, Nyack, N. Y.

CLEGG, Carl (*M* 1922) Dist. Mgr., ● American Blower Corp., 550 Dwight Bldg., and 3513 Gillham, Kansas City 6, Mo.

CLEMENS, Joseph D. (*J* 1942; *S* 1940) Major, A. C., ● Box 5, Base Hq., Kelly Field, Texas.

CLEMENS, J. Edward (*M* 1944) Engr. in charge, Plant Engrg. Dept., Allison Div. of General Motors Corp., Plant No. 5, P. O. Box 894, Indianapolis, and ● 2625 N. Meridian St., Indianapolis 5, Ind.

CLEMENT, Eugene R., Sr. (*A* 1942) Pres., ● E. R. Clement, Inc., 297 Washington Ave., and 205 Washington Ave., Bridgeport 4, Conn.

CLEMINS, Richard J. (*J* 1945) Jr. Engr., Carrier Corp., 300 S. Geddes St., Syracuse, N. Y., and ● 1614 W. Scott St., Milwaukee 4, Wis.

CLIFTON, John A. (*A* 1938) Mgr., ● Renown Plumbing Supplies, Ltd., 236 Parliament St., and 369 Belsize Dr., Toronto, Ont., Canada.

CLO, Harry E. (*A* 1943; *J* 1939) Engr., ● American Air Filter Co., 228 N. LaSalle St., Chicago 1, and 10565 S. Hale Ave., Chicago 43, Ill.

CLOSE, James W. (*A* 1944) Pres., Sheet Metal Workers Union, 620 S. Ashland Blvd., Chicago, and 6114 N. Washtenaw Ave., Chicago 45, Ill.

CLOSE, Paul D.* (*M* 1928) Tech. Secy., ● Insulation Board Institute, 111 W. Washington St., Chicago, Ill.

CLOSE, Robert (*M* 1938) Chief Air Cond. Engr., National Broadcasting Co., 30 Rockefeller Plaza, New York, N. Y., and ● 185 Glenwood Ave., Leonia, N. J.

CLOW, Sherwood A. (*J* 1942) 1st. Lt., Signal Corps, ● Signal Office, Camp Howze, Texas, and 109 N. Chatsworth Ave., Larchmont, N. Y.

CLUCAS, Edward T.* (*M* 1944) Application Engr., ● Minneapolis-Honeywell Regulator Co., 4030 Chouteau Ave., St. Louis 10, and 6707 Arthur Ave., St. Louis 9, Mo.

COAD, J. Dennis (*A* 1943) Service Engr., ● D. C. Air Conditioning Service Co., 1125 Locust St., St. Louis 1, and 6439 Lloyd Ave., St. Louis 10, Mo.

COBURN, Joseph B. V. (*M* 1944) Combus. & Service Engr., International Chimney Corp., Curtis Bldg., Buffalo, and ● 189 Mill St., Williamsville, N. Y.

COCHRAN, L. H. (*M* 1934) Dist. Mgr., ● American Blower Corp., 625 Market St., San Francisco 5, and 130 Camino Del Mar, San Francisco, Calif.

COCHRAN, Lucius D. (*A* 1945) ● 3447 Logan Rd., Youngstown, Ohio.

COCHRANE, William (*A* 1944) Air Cond. Engr., Federal Shipbuilding & Drydock Co., Engrg. Div., Hull Dept., 744 Broad St., Newark, and ● 91 Beech St., Arlington, N. J.

COCKINS, William W. (*A* 1941; *J* 1937) Scott Co., 243 Minna St., San Francisco 3, and ● 555 Cragmont Ave., Berkeley 8, Calif.

COCKLEY, Jonathan E. (*M* 1943) Htg. & Air Cond. Engr., ● Indiana Gas & Water Co., Inc., Traction Terminal Bldg., Indianapolis 9, and 3463 N. Sherman Dr., Indianapolis 1, Ind.

COCKMAN, Thomas E. (*A* 1945) Salesman, ● B. F. Sturtevant Company of Canada, 620 Cathcart St., and 4823 Patricia Ave., Montreal, Que., Canada.

CODERE, Jean-Moise (*A* 1944) Dir., ● Codere, Ltd., 18 Wellington N., and 71 Brooks, Sherbrooke, Que., Canada.

CODY, Henry C. (*M* 1936) Sales Engr., Herman Goldner Co., Inc., 425 W. Lehigh Ave., Philadelphia 33, and ● 7336 North 21st St., Philadelphia 38, Pa.

COE, Seymour A. (*J* 1944; *S* 1942) ● 54 Waverly St., New Haven, Conn.

COFFEY, George B. (*A* 1945) Field Service Engr., ● A. M. Byers Co., 111 W. Washington St., Rm. 1217, Chicago 2, and 2311 West 111th St., Chicago 43, Ill.

COGHLAN, Sherman F. (*A* 1937) Pres., J. M. Montgomery & Co., Inc., 306 W. Third St., Los Angeles, and ● 414 Ninth St., Santa Monica, Calif.

COHAGEN, Chandler C. (*M* 1919) Archt., ● 212 Hedden Bldg., and 235 Avenue G, Billings, Mont.

COHEN, Milton L. (*J* 1945) Asst. Engr., Boston Filter Co., 43 Harvard Sq., Charlestown, and ● 111 Atlantic Ave., Revere 51, Mass.

COHEN, Philip (*M* 1932) Dist. Mgr., ● B. F. Sturtevant Co., 933 Leader Bldg., Cleveland 14, and 12805 Shaker Blvd., Apt. 511, Cleveland 20, Ohio.

COHN, Henry (*J* 1942) Owner, Republic Heating & Cooling Co., Dallas, Texas.

COLBY, John H. (*A* 1944; *J* 1939) ● 20 Winchester St., Boston, and 25 Jefferson Rd., Wellesley Hills, Mass.

COLBY, John R. (*A* 1944) Owner, ● Colby Equipment Co., 243 E. Ohio St., Indianapolis 4, and 5514 Rosslyn Ave., Indianapolis, Ind.

COLCLOUGH, Otho T. (*A* 1933) ● American Embassy, and 399 Hamilton Ave., Ottawa, Ont., Canada.

COLE, C. Boynton (*M* 1940; *J* 1937) Owner, Boynton Cole, 1873 Piedmont Rd. N.E., and ● 1843 Flagler Ave. N.E., Atlanta, Ga.

COLE, Grant E. (*A* 1925) Vice-Pres. & Mgr., ● Trane Company of Canada, Ltd., 4 Mowat Ave., and 112 Tyndall Ave., Toronto, Ont., Canada.

COLE, Harold S. (*M* 1944) Engr., Stone & Webster Engineering Corp., 49 Federal St., Boston 7, and ● 1960 Commonwealth Ave., Boston 35, Mass.

COLE, Victor (*J* 1945) Engr., Carrier Corp., 405 Lexington Ave., New York 17, and ● 1555 Grand Concourse, New York 52, N. Y.

COLEMAN, John B. (*M* 1920) Chief Engr., ● Grinnell Corp., P. O. Box 1435, and 237 Cole Ave., Providence, R. I.

COLFORD, John (*A* 1937) Pres., ● John Colford, Ltd., 2007 Guy St., Montreal 25, and 51 Upper Bellevue Ave., Westmount, Que., Canada.

COLLE, S. S. (*A* 1938) Engr., & Owner, ● Air Conditioning Engineering Co., 79 Vitre St. W., Montreal 1, and 4968 Fulton Ave., Montreal, Que., Canada.

COLLIER, J. Robert (*A* 1945) Dist. Repr., Taco Heaters, Inc., P. O. Box 402, Minneapolis 1, and ● 4808 Pleasant Ave., Minneapolis 9, Minn.

COLLIER, William I. (*M* 1921) Mech. Engr., ● W. I. Collier & Co., 3414 Duvall Ave., Baltimore 16, and Ellicott St., Ellicott City, Md.

COLLINS, George J. (*A* 1945) Supt., ● Barber Plumbing Co., Box 84, and 529 Fraser, Houston, Texas.

COLLINS, John F. S., Jr. (*M* 1933), (Council, 1940-45) Secy.-Treas., ● National District Heating Assn., 827 N. Euclid Ave., Pittsburgh 6, Pa.

COLLINS, Joseph A. (*M* 1943) Mgr., ● Frontier Oil Refining Corp., Oil Burner Div., 367 Northampton St., and 289 Commonwealth Ave., Buffalo, N. Y.

COLLINS, Leo F.* (*M* 1941) Cons. Engr., ● 14615 Prevost Ave., Detroit 27, Mich.

COLMAN, Robert C. (*A* 1940) Vice-Pres., McQuay, Inc., 1600 Broadway N.E., Minneapolis 13, and ● 102 Exeter Pl., St. Paul 4, Minn.

COLMENARES, Gaspar Vizoso (*A* 1938) Vice-Pres. & Dir., ● Refrigeracion y Aire Acondicionado, S. A., Lamparilla No. 9, P. O. Box 210, and Calle 10—No. 34, Miramar, Habana, Cuba.

COLTON, Frederick R. (*M* 1945) Branch Mgr., ● Norris Warming Co., 94 Exchange Bldgs., New St., Birmingham, and 36 Palace Rd., Flat 12, London S.W. 2, England.

COLVIN, Oliver Dyer (*M* 1943) Pres., ● Cargocaire Engineering Corp., 15 Park Row, New York, N. Y.

COMO, Jack A. (*M* 1939) Mech. Engr., ● Independent Plumbing Co., 171 Luckie St. N.W., and 2865 Elliott Circle, Atlanta, Ga.

COMSTOCK, Glen M. (*A* 1926) Engr., Repr., ● L. J. Wing Manufacturing Co., 1319 Murdoch Rd., Pittsburgh 17, Pa.

CONATY, Bernard M. (*M* 1935) Mfrs. Repr., ● 505 Crosby Bldg., Buffalo 2, and P. O. Box 342, Jennings Rd., Eden, N. Y.

CONGER, Henry L. (*M* 1943) Assoc. Engr., U. S. Engineering Dept., Punahou, Honolulu, and ● 1550 K Young St., Honolulu 19, T. H.

CONNELL, E. C. (*M* 1944) Supt. Construction, ● Sullivan Valve & Engineering Co., Box 688, and 158 Lewis St., Klamath Falls, Ore.

CONNELL, Maurice H. (*M* 1945) Cons. Engr., ● Maurice H. Connell & Assoc., 816 Langford Bldg., Miami 32, and 1101 N. Greenway Dr., Coral Gables, Fla.

CONNELL, Richard F. (*M* 1916) Mgr., Sales Engrg. Div., ● U. S. Radiator Corp., 1500 United Artists Bldg., Detroit 31, and 2970 Burlingame, Detroit, Mich.

CONNER, R. M. (*M* 1931) Dir. Lab., ● American Gas Association Laboratories, 1032 East 62nd St., Cleveland 14, and 2459 Dysart Rd., Cleveland, Ohio.

CONNORS, Edward C. (*A* 1940) Engr. Custodian, Chicago Board of Education, 5500 Madison St., and ● 6556 Ponchartrain Blvd., Chicago 30, Ill.

CONOVER, Donald Davis (*A* 1944) Gas Repr., ● Philadelphia Electric Co., 5 E. Lancaster Ave., Ardmore, and 430 Manoa Rd., Brookline, Del. Co., Pa.

CONOVER, E. W. (*M* 1944) Research Engr., Detroit Steel Products Co., 2250 E. Grand Ave., Detroit 11, and ● 9165 Stoepel Ave., Detroit 4, Mich.

CONRAD, Roy (*M* 1935) Sales Engr., Carrier Corp., 610-1331 Third Ave. Bldg., Seattle 1, and ● 3421 Bella Vista Ave., Seattle 44, Wash.

CONROY, W. T. (*A* 1944) Htg. & Plbg. Contr., ● 1525 Locust St., and 1810 East 78th St., Kansas City, Mo.

CONSTANT, Earl S. (*A* 1942; *J* 1935) Sales Engr., ● Arthur Forsyth Co., 500 First Ave. S., Seattle 4, and 3015-30th Ave. W., Seattle 99, Wash.

CONVERSE, Thornton J. (*M* 1941) Engr., ● Douglas Orr, 96 Grove St., New Haven, and Stoney Creek, Conn.

COOK, Benjamin F. (*M* 1945) ● 1720 Overton Ave., Independence, Mo.

COOK, Clifford B. (*A* 1944) Partner, ● F. G. Cook & Sons, 102 Deerhurst Park Blvd., and 337 McKinley Ave., Kenmore, N. Y.

COOK, H. Dale (*A* 1938) Sales Engr., Perfex Corp., 500 W. Oklahoma, Milwaukee, and ● 5061 N. Idlewild, Milwaukee 11, Wis.

COOK, Ralph P. (*M* 1930) Supt., Engrg. & Maintenance Dept., in charge of Engrg. Div., ● Eastman Kodak Co., Kodak Park Works, Rochester 4, and 663 Seneca Pkwy., Rochester, N. Y.

COOK, Russell M. (*A* 1945) Vice-Pres., ● Thatcher Furnace Co., Center St., Garwood, and 15 Afterglow Ave., Montclair, N. J.

COOK, Vernon D. (*A* 1944) Partner ● F. G. Cook & Sons, 102 Deerhurst Park Blvd., Kenmore, N. Y.

COOK, W. L. (*A* 1945) Owner & Supt., ● P. S. Cook, 1722 Thomes St., and 609 East 21st St., Cheyenne, Wyo.

COOKE, William L. (*A* 1944) Managing Dir., ● W. L. Cooke, Ltd., Cor. Kent and York St., Newmarket, Auckland S.E. 1, and 747 Remuera Rd., Auckland S.E. 2, New Zealand.

COOLEY, Edgerton C. (*M* 1938) Owner, ● E. C. Cooley Co., 625 Market St., San Francisco 5, and P. O. Box 789B, Route 1, Los Altos, Calif.

COOMBE, James (*A* 1932) Pres., ● William Powell Co., 2525 Spring Grove Ave., and 2363 Grandin Rd., Cincinnati, Ohio.

COON, Thurlow E. (*Life Member; M* 1916) Pres., ● The Coon-DeVisser Co., Inc., 2051 W. Lafayette, Detroit 16, and 826 Edison Ave., Detroit 2, Mich.

COOPER, Albert W. (*M* 1944) Branch Mgr., ● Johnson Service Co., 1230 California St., Denver 4, and 1213 Olive St., Denver 7, Colo.

COOPER, C. H. (*A* 1944) Htg. Dept., Chase Co., 936 Oak, and ● 1345 East 22nd, Eugene, Ore.

COOPER, Dale S. (*M* 1938; *A* 1937) Cons. Engr., 206 West Bldg., Houston 2, and ● 216 E. Cowan Dr., Houston, Texas.

COOPER, Donald E. (*J* 1939) Partner, ● D. E. Cooper & Son, 540 Hood St., and 740 Lomond Dr., Salem, Ore.

COOPER, George P. (*A* 1944) Combus. Engr., ● Empire-Hanna Coal Co., Ltd., 805 C. P. R. Bldg., Toronto, and 27 Dartmouth Cres., Mimico, Ont., Canada.

COOPER, John W. (*M* 1932; *A* 1925; *J* 1921) Repr., ● Buffalo Forge Co., 2726 Locust St., St. Louis, and 612 Hawbrook Dr., Kirkwood, Mo.

COOPER, W. B. (*A* 1946) ● 163 Barlett Ave., N. Wilbraham, Mass.

COOPERMAN, Edward (*J* 1943, *S* 1940) Ensign, B. O. Q. Bks. E-3, N. C. T. C., Davisville, R. I., and • 3444 Ward St., Pittsburgh 13, Pa.

CORBIT, Charles A. (*A* 1943) Pres., • Corbit's Inc., 225 S. Front St., and 901 Centre Ave., Reading, Pa.

COREY, George R., Sr. (*M* 1945) • Superior Coal Co., 78 Narragansett Ave., Providence 7, R. I.

CORLEW, James B. (*A* 1945) Application Engr., Worthington Pump & Machinery Corp., Air Cond. & Refrig. Div., and • 72 Martin St., Holyoke, Mass.

CORNELIUS, George E. (*A* 1943) Elec. Engr., Philadelphia Suburban Transportation Co., 69th St. Terminal Bldg., Upper Darby, and • 316 Burmont Rd., Drexel Hill, Pa.

CORNWALL, George L. (*Life Member, M* 1919) Sales Engr., • Burnham Boiler Corp., 701 Spring St., Elizabeth, N. J.

CORRIGAN, James A. (*M* 1944, *A* 1940, *J* 1935, *S* 1930) Treas., • Corrigan Co., 2501 St. Louis Ave., St. Louis, and 7128 Washington Ave., University City, Mo.

CORRIGAN, Thomas J. (*A* 1945) Vice Pres., • Corrigan Co., 2501 St. Louis Ave., St. Louis, and 7554 Parkdale, Clayton, Mo.

CORYELL, Glynn L. (*M* 1943) Chief Field Engr., U. S. Machine Corp., and • Country Club Park, Lebanon, Ind.

COSGROVE, James J. (*A* 1944) Wholesale Mgr., LaSalle Coke Co., 660 St. Catherine St. W., Montreal 2, and • 1610 University Tower, Montreal, Que., Canada.

COST, George W. (*A* 1946, *J* 1939, *S* 1938) 1st Lt., and • 404 Pennsylvania Ave., Irwin, Pa.

COSTA, Ciro A. (*M* 1945) Merchandise Engr., • Westinghouse Electric International Co., Paseo Colon 221, Buenos Aires, and Santa Paula, Marcos Paz, F. C. S., Argentina, S. A.

COTTON, Irwin W. (*M* 1943) Owner, • The I. W. Cotton Co., 1035 N. Pennsylvania St., Indianapolis 4, Ind.

COUCH, George R. (*A* 1944) Owner, George R. Couch Co., 157 Charter Oak Ave., Hartford 5, and • 15 Bushnell St., Hartford 6, Conn.

COULTER, Thomas H. (*A* 1943) Vice Pres., • Univereal Zonolite Insulation Co., 135 S. LaSalle St., Chicago 3, and 201 Kedzie St., Evanston, Ill.

COVER, E. B. (*M* 1937) Dist. Repr., • General Electric Co., 2206 K. C. Power & Light Bldg., Hotel President, Kansas City, Mo.

COVER, Richard R. (*A* 1936) Lt., U. S. N. R., and • 1914 N. Upton St., Arlington, Va.

COWAN, Armand. (*A* 1945) Pres., • Stuart Cooling Corp., 153 East 26th St., New York 10, and 1401 Elm Ave., Brooklyn, N. Y.

COWARD, Charles W. (*M* 1935) • Coward-Eastman Co., 2206 Chestnut St., Philadelphia 3, Pa., and Cherry Lane, Riverton, N. J.

COX, Edward H. (*M* 1943) Owner, • Heating & Ventilating Equipment, 303 Crosby Bldg., Buffalo 2, and 106 S. Lake St., Hamburg, N. Y.

COX, LeRoy E. (*A* 1944) Htg., Vent. & Power Engr., Almirall & Co., Inc., 53 Park Pl., New York, N. Y., and • 20 Laurel Pl., Glen Ridge, N. J.

COX, Samuel F. (*M* 1939) Research Dir., • Pittsburgh Plate Glass Co., Creighton, and 1722 Orchard Ave., Arnold, Pa.

COX, Vernon G. (*A* 1939) Dist. Mgr., • Century Electric Co., 514 Texas Bank Bldg., Dallas 2, and 207 Yarmouth St., Dallas 8, Texas.

COX, William W. (*Life Member, M* 1923) Mgr., • Heating Service Co., 326 Columbia St., Seattle 4, and 7857–56th Pl. N. E., Seattle, Wash.

COYNE, John H. (*A* 1943) Foreman Htg. Vent., Wright Aeronautical Corp., and • 6303 Hammel Ave., Cincinnati 12, Ohio.

CRABBIN, William W. (*A* 1944) Master Sheet Metal Worker, U. S. Coast Guard, Navy Yard, Curtis Bay, and • 312 Washburn Ave., Baltimore 25, Md.

CRAIG, Joseph A. (*J* 1940) Partner, • Roy E. Gorgen Co., 507 Wesley Temple Bldg., and 2634 W. River Dr., Minneapolis, Minn.

CRAIG, William C. (*M* 1945) • Young Radiator Co., Racine, Wis.

CRANAGE, Thomas (*M* 1943) Dist. Sales Mgr., • Clarage Fan Co., 333 N. Michigan Ave., Chicago, and 399 Sunset Lane, Glencoe, Ill.

CRARY, James O. (*M* 1944) Mgr., Appliance Div., • Higgins Industries, Inc., 521 City Park Ave., New Orleans 19, and 410 Glendale Blvd., New Orleans, La.

CRAWFORD, Arthur C. (*A* 1938) Air Cond. Engr., Potomac Electric Power Co., Washington, and • 429 Butternut St. N.W., Washington 12, D. C.

CRAWFORD, Edward F. (*A* 1945) Owner, • M. B. Crawford & Son, 46 W. Bridge St., and 43 W. Fifth St., Oswego, N. Y.

CRAWFORD, Fred A. (*A* 1945) Owner, • Home Comfort Heating Co., 5131 Rohms Ave., Detroit 13, Mich.

CRAWFORD, John H., Jr. (*A* 1936, *J* 1930) Engr., Westinghouse Electric Corp., Air Cond. Div., 9 Rockefeller Plaza, New York, N. Y., and • 289 Reynolds Terrace, Orange, N. J.

CREEARS, Tom H. (*M* 1945) Owner, • 1824 S. Hope St., Los Angeles 15, and 3458 Knollcrest Ave., Los Angeles 43, Calif.

CRESSY, L. Rene (*A* 1943) Partner, • Cressy & Leggett, 208 Balter Bldg., and 1431 Marais St., New Orleans, La.

CRESSY, L. Villere (*M* 1940) • 309 Vincent Bldg., New Orleans 12, La.

CREW, F. D. (*A* 1944, *J* 1941) Pres., • The F. D. Crew Co., 1539 Race St., Philadelphia 2, and Paoli, Pa.

CREW, Morris W. (*M* 1944) Mgr., Indus. Control Div., Perfex Corp., and • Shorecrest Hotel, Milwaukee, Wis.

CRIQUI, Albert A. (*M* 1919) Chief Engr., Buffalo Forge Co., 490 Broadway, Buffalo, and • 39 St. Johns Ave., Kenmore, N. Y.

CRIQUI, Albert E. (*M* 1944) Mech. Engr., • Buffalo Forge Co., 490 Broadway, Buffalo, and 89 Dorset Dr., Kenmore 17, N. Y.

CRISMAN, Leon J. (*M* 1945) Owner, Eng., • Crisman Heating & Air Conditioning Co., 502 N. Larch St., Lansing, and Apt. 252-C, Hillcrest Village, East Lansing, Mich.

CROFT, Huber O. (*M* 1941) Dept. Head Mech. Engrg., • University of Iowa, 122 Engineering Bldg., and 250 North St., Iowa City, Iowa.

CROLEY, Jack G. (*J* 1940) • 406 E. Temple Ave., College Park, Ga.

CRONE, Charles E. (*M* 1922) Pres., • Charles E. Crone Co., 1656 N. Ogden Ave., Chicago 14, Ill.

CRONE, Thomas E. (*Life Member, M* 1920) Retired, and • 164th and Chapin Parkway, Jamaica 3, L. I., N. Y.

CRONEY, P. Alfred (*M* 1938) Sr. Mech. Engr., Federal Public Housing Authority, 24 School St., Boston, and • 72 Arlington St., Newton, Mass.

CRONSTROM, Kenneth A. (*A* 1944) Prop., • Cronstroms Heating & Sheet Metal, 4225 Hiawatha Ave., Minneapolis 6, and 4210 Standish Ave., Minneapolis, Minn.

CROPPER, Robert O. (*M* 1938) Major, Q. M. C., • McGuire General Hospital, Richmond, and 300 River Rd., Matoaca, Va.

CROSBY, Edward L. (*M* 1936) Pres., • Henry Adams, Inc., 403-407 Calvert Bldg., Baltimore 2, and 700 Brookwood Rd., Baltimore 29, Md.

CROSS, Freeman G. (*M* 1936) Vice Pres. & Gen. Sales Mgr., • Fulton Sylphon Co., and 31 Nokomis Circle, Knoxville, Tenn.

CROSS, Lloyd E. (*A* 1945) Gen. Supt., • American Foundry & Furnace Co., 709 North 11th St., Milwaukee 3, and 2407 North 44th St., Milwaukee 10, Wis.

CROSS, Robert C. (*M* 1937) Head of Mech. Div., • Sears, Roebuck & Co., Dept. 817, 925 S. Homan Ave., Chicago, and 4106 Linden Ave., Western Springs, Ill.

CROSS, Robert E. (*M* 1938, *A* 1931) Dist. Mgr., Minneapolis-Honeywell Regulator Co., P. O. Box 559, Springfield, and • Granville Center, Mass.

CROSS, William D. (*A* 1943) Pres., • Cross Engineering Co., 11 E. Broad St., and Old Raritan Rd., Westfield, N. J.

CROUT, M. M. (*M* 1939, *A* 1938) Mgr., Southern Dist., • York Corp., 412 Houston St. N. E., Atlanta 1, and 537 E. Wesley Rd. N. E., Atlanta, Ga.

CRUMLEY, Mearl T. (*M* 1941) Mgr., Mech. Engrg. Section, U. S. Naval Air Station, Jacksonville, and • 1132 Ingleside Ave., Jacksonville 5, Fla.

CRUMP, Alvin L. (*M* 1937) Mgr., Chicago Contract Sales Dept., • Powers Regulator Co., 2720 N. Greenview Ave., Chicago 14, and 2701 Payne St., Evanston, Ill.

CRUMP, Samuel (*A* 1943) Pres., • Samuel Crump Co., Ltd., 317 Eglinton Ave. W., Toronto 12, and 34 Braemar Ave., Toronto, Ont., Canada.

CUCCI, Victor J. (*M* 1930) Cons. & Marine Engr., • 50 Church St., Rm. 282, New York, and 451–55th St., Brooklyn, N. Y.

CULBERT, William P. (*M* 1943, *A* 1929) Partner, • Culbert-Whitby Co., 2019 Rittenhouse Sq., Philadelphia 3, and 929 Alexander Ave., Drexel Hill, Pa.

CULVER, Frank C. (*A* 1945) Mfrs. Repr., and • 4710 University Way, Seattle 5, Wash.

CUMMING, Robert W. (*M* 1928) Engr. & Sales Exec., Sarco Co., Inc., 475 Fifth Ave., New York, and • 81 Altamont Ave., Scarsdale, N. Y.

CUMMINGS, Carl H. (*A* 1927, *J* 1926) • The Bryant Heater Co., 1020 London Rd., Cleveland, Ohio.

CUMMINGS, George C. (*M* 1944) N. Y. Repr., Dravo Corp., 512 Fifth Ave., and • 4 West 43rd St., New York 18, N. Y.

CUMMINGS, G. J. (*M* 1923) Business & Economic Reports, and • 851 Trestle Glen Rd., Oakland, Calif.

CUMMINGS, Ira R. (*M* 1943) Precipitron Application Engr., Precipitron Div., Westinghouse Electric Elevator Co., 150 Pacific Ave., Jersey City, and • 717 Willow St., Cranford, N. J.

CUMMINGS, Robert J. (*A* 1944, *J* 1940) Engr., • Franck & Fric Co., 7109 Kinsman Rd., Cleveland 4, and 18110 Maple Heights Blvd., Bedford, Ohio.

CUMMINGS, Thomas P. (*A* 1942) • Hayes Furnace & Manufacturing Co., Inc., 2831 Exposition Pl., Los Angeles 16, and • 5145 Seventh Ave., Los Angeles 43, Calif.

CUMMINS, George H. (*M* 1919) Dist. Mgr., • Aerofin Corp., 1200 United Artists Bldg., Detroit 26, Mich.

CUMMISKEY, Jerome F. (*A* 1940) Minneapolis-Honeywell Regulator Co., 433 E. Erie, Chicago 11, and • 7706 Eastlake Terrace, Chicago 26, Ill.

CUMNOCK, H. (*A* 1938) Pres., • Little Rock Refrigeration Co., Inc., 417 W. Capitol Ave., Little Rock, Ark.

CUNNINGHAM, John S. (*A* 1941, *J* 1937, *S* 1935) Factory Mgr., Meyer Furnace Co., 1300 S. Washington St., Peoria 2, and • 2200 Peoria Ave., Peoria 4, Ill.

CUNNINGHAM, Thomas M. (*M* 1931, *A* 1930) Dir. of Indus. Relations, • Carrier Corp., 300 S. Geddes St., Syracuse 1, N. Y.

CUPITT, Robert (*A* 1943) Gen. Mgr., Metaducta Ltd., Catherine Wheel Rd., Brentford, Middx., and • 60 Gunnersbury Ave., Ealing, London W. 5, England.

CURL, Robert S. (*A* 1941) Mech. Engr. in charge of High Altitude Chamber, U. S. Naval Ordnance Plant, 6000 East 21st St., Indianapolis 6, and • 3445 Winthrop, Indianapolis 5, Ind.

CURLEY, Ellis I. (*A* 1941) Lt., U. S. N. R., • 7028 Greene St., Philadelphia 19, Pa.

CURRY, Roger F. (*J* 1940, *S* 1938) S 2/C, A. & R. Bldg. 631, N. A. T. B., and • 227 Gregory Ave., Pensacola, Fla.

CURTICE, Jean M. (*A* 1936) Colo. Dist. Mgr., • Citizens Utilities Co., 15 W. Fourth St., and 906 Lincoln Ave., La Junta, Colo.

CURTIS, A. Ray (*A* 1945) Owner & Mgr., • A. Ray Curtis Co., 2205 Highland Dr., and 2642 Beverly St., Salt Lake City, Utah.

CURTIS, Herbert F. (*A* 1934) Sales Mgr., • Auer Register Co., 3608 Payne Ave., Cleveland, and 59 Fourth Ave., Berea, Ohio.

CUSHING, C. F. (*M* 1938) Vice Pres., • Payne Furnace Co., 336 N. Foothill Rd., Beverly Hills, Calif., and 13415 S. Woodland Ave., Cleveland, Ohio.

CUSHING, R. C. (*A* 1940) Control Engr., • Minneapolis-Honeywell Regulator Co., 1136 Howard St., San Francisco 3, and 1330 Oxford St., Berkeley 7, Calif.

CUTLER, Joseph A. (*M* 1916) (Council 1920–26) Pres. & Gen. Mgr., • Johnson Service Co., 507 E. Michigan St., Milwaukee 2, and 4811 N. Lake Dr., Milwaukee 11, Wis.

CUTTING, Richard Hawley (*M* 1942) Archts., Richard Hawley Cutting, Anthony S. Ciresi, and Associated Architects, 4900 Euclid Ave., Cleveland 3, and • 3795 Glenwood Rd., Cleveland Heights, Ohio.

D

DABBS, John T. (*A* 1940) Capt., C. A. A. P. O. 937, c/o Postmaster, Seattle, Wash., and • 330 N. Green St., Tupelo, Miss.

DADDARIO, F. T. (*A* 1946, *J* 1939) Lt. (j.g.) U. S. N. R., • U. S. S. Yorktown, c/o F. P. O., San Francisco, Calif., and 720 Strong St., Schenectady, N. Y.

DAFTER, Edwin H. (*M* 1938) Sales Engr., • Carrier Corp., 12 South 12th St., Philadelphia 7, and 117 Crosshill Rd., Overbrook Hills, Philadelphia 31, Pa.

DAHL, Lawrence J. (*S* 1943) A. R. T. 3/C, • Aviation Radio Technician School, Nattc, Bris. No. 1, Ward Island, Corpus Christi, Texas, and Route No. 7, Minneapolis 11, Minn.

DAHLGREN, Gustave E. (*A* 1940) Mgr., Insulation Dept., • Thorkelsson, Ltd., 1331 Spruce St., and 533 Basswood Pl., Winnipeg, Man., Canada.

DAHLSTROM, Godfrey A. (*M* 1943, *A* 1927) Mech. Engr., Minnesota Mining & Manufacturing Co., St. Paul, and • 3721–47th Ave. S., Minneapolis, Minn.

DAHLSTROM, John A. (*M* 1944) Hd. of Furnace Development Div., • Perfection Stove Co., 7609 Platt Ave., Cleveland, and 420 Canterbury Rd., Bay Village, Ohio.

DAHMS, Alfred A. (*A* 1944) Dist. Mgr., • Allis-Chalmers Manufacturing Co., 1410 Waldheim Bldg., Kansas City 6, and 7431 Mercier, Kansas City 5, Mo.

DAILEY, J. F. (*M* 1944) • Typhoon Air Conditioning Co., Inc., 252 West 26th St., New York, N. Y.

DAITSH, Abe. (*A* 1944, *J* 1938) Air Cond. & Refrig. Engr., • General Assurance Bldg., 86 St. George's St., Cape Town, South Africa.

DALTON, Robert T. (*A* 1943) Owner, • Dalton Sales & Service, 3602 East 138th St., Cleveland 20, Ohio.

DALY, Robert E. (*M* 1931) • American Radiator & Standard Sanitary Corp., P. O. Box 1226, Pittsburgh 30, and 271 Lenforest Dr., Mt. Lebanon 16, Pa.

D'AMBLY, A. Ernest (*M* 1924, *J* 1921) Owner, • A. Ernest D'Ambly, 117 South 17th St., Rm. 900, Philadelphia 3, and 1835 DeLancey St., Philadelphia, Pa.

DANIEL, William E. (*A* 1941, *J* 1939) Junior Partner, • E. Ashby & Co., Jamaica Wharf, 20 Upper Ground, Black Friars, and 91 Dudley House, Westmoreland St., London W. 1, England.

DANIELSON, Wilmot A.* (*M* 1935, Council 1944-45) Brig. Gen., U. S. Army, • Memphis Army Service Forces Depot, Memphis 2, and Route 3, Box 416, Memphis, Tenn.

DANOWITZ, Chester J. (*J* 1942) Ensign, U. S. N. R., and • 75 Poplar Ave., Red Bank, N. J.

DAPPER, M. J. (*A* 1944) Owner, • 609 South 11th, Tacoma 2, and 3819 North 38th, Tacoma, Wash.

DARBY, F. Norman (*M* 1944) Engr. in charge Htg. & Vent. Sect., Sun Shipbuilding & Drydock Co., Chester, and • 5236 Cedar Ave., Philadelphia 43, Pa.

DARBY, Thomas E. (*M* 1944) Utilization Engr., • The Ohio Fuel Gas Co., 99 N. Front St., Columbus 15, and 29 Tibet Rd., Columbus 2, Ohio.

DARLING, A. B. (*A* 1929) Comptroller, • Darling Brothers, Ltd., P. O. Box No. 187, and 4009 Grey Ave., Montreal, Que., Canada.

DARLING, Robert B. (*A* 1944) • Robert B. Darling Co., 280 Madison Ave., New York 16, and 65 Highland Ave., Port Washington, N. Y.

DARRAH, Richard D. (*M* 1944) Sales Engr., Dravo Corp., 300 Penn Ave., and • 515 Roslyn Pl., Pittsburgh 6, Pa.

DARTS, John A. (*M* 1919) Vice Pres., • Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York 17, and 610 West 110th St., New York 25, N. Y.

DASING, Emil (*M* 1937) Design Engr., Sears, Roebuck & Co., 925 S. Homan Ave., Chicago, and • 2618 W. Eastwood Ave., Chicago 25, Ill.

DASTUR, Maneck N. (*J* 1945) Jr. Engr., • Carrier Corp., 300 S. Geddes St., and 1011 Walnut Ave., Syracuse, N. Y.

DAUBER, Oscar W. (*M* 1937) Cons. Engr., • 224 S. Michigan Ave., Chicago, and 532 Greenwood Ave., Kenilworth, Ill.

DAUCH, Emil O. (*M* 1921) Pres., • McCormick Plumbing Supply Co., 1675 Bagley Ave., and 729 Bedford Rd., Grosse Point Park, Detroit, Mich.

DAUSSAT, Waldo J. (*A* 1945) Chief Engrg. Draftsman (Mech.), U. S. Navy, Public Works Dept., 1015 Federal Bldg., and • 4436 Elba St., New Orleans, La.

DAVENPORT, Lind B. (*M* 1944) Chief Engr., • Air Conditioning Co. of Southern California, 1003 Santa Fe Ave., Los Angeles 21, and 139 Melrose Ave., Monrovia, Calif.

DAVEY, Geoffrey I. (*M* 1937) • Gutteridge, Haskins & Davey, City Mutual Bldg., 60-66 Hunter St., Sydney, and Netherby, Bangalla St., Warrawee, New South Wales, Australia.

DAVIDSON, James W. (*A* 1944) Pres. & Owner, • Davidson Heating Regd., Val Morin, Co. Terrebonne, Quebec, Canada.

DAVIDSON, John C. (*M* 1940; *J* 1936) • 4708 Isabel Ave., Minneapolis, Minn.

DAVIDSON, L. Clifford (*M* 1927) Partner, • Davidson & Hunger, 220 South 16th St., Philadelphia, and 322 Winding Way, Merion, Pa.

DAVIDSON, Philip L. (*M* 1924; *J* 1921) Cons. Engr., • New Hope, Pa., and Greenville, S. C.

DAVIDSON, William J. (*M* 1945) Application Engr., • N. O. Nelson Co., 118 E. River St., and 602 Gaylord, Pueblo, Colo.

DAVIES, Charles (*M* 1945) Pres., • Davies Air Filter Co., 250 East 43rd St., New York, and 265 Colgni Ave., New Rochelle, N. Y.

DAVIES, Edwin A. (*M* 1944) Dist. Repr. Engr., Hoffman Specialty Co., 1001 York St., Indianapolis, Ind., and • 3022 Kingman Blvd., Des Moines, Iowa.

DAVIES, George W. (*M* 1918) Mgr., • F. W. Davies & Co., 19 Maclaggan St., Dunedin, C. 1., and P. O. Box 390, Dunedin, N. 2, Colinswood, Macandrew Bay, New Zealand.

DAVIES, Richard H. (*A* 1946; *J* 1943) Htg. & Mech. Engr., County Architect's Dept., County Hall, Council, Glamorgan County, Cardiff, and • c/o 29 Grove Rd., Bridgend, Glamorgan, South Wales, England.

DAVIS, Bert C. (*Life Member*; *M* 1904), (Council, 1917) • Big Flats, N. Y.

DAVIS, Charles (*M* 1938) U. S. Navy (Sea Bees), Chief Petty Officer, U. S. N. R., and • 1066 Walton Ave., New York, N. Y.

DAVIS, C. R. (*M* 1927) Branch Mgr., • Johnson Service Co., 2328 Locust St., St. Louis 3, and 7527 Forsythe Blvd., Clayton 5, Mo.

DAVIS, Clemant A. L. (*A* 1942) Mgr., Ref. Dept., John H. Kitchen & Co., 1016 Baltimore, Kansas City 6, and • 3200 East 32nd St., Kansas City, 3, Mo.

DAVIS, Donald W., Jr. (*A* 1946; *J* 1939) Dist. Mgr., • B. F. Sturtevant Co., 854 Empire Bldg., Milwaukee 3, and 3748 North 56th St., Milwaukee 10, Wis.

DAVIS, Edward James (*A* 1943; *J* 1938) Sales Engr., Gurney Foundry Co., Ltd., 4 Junction Rd., and • 224 St. Clements Ave., Toronto, Ont., Canada.

DAVIS, George C. (*M* 1939; *J* 1936) Vice-Pres., • Northern Public Service Corp., Ltd., 307 Power Bldg., and 366 Ash St., Winnipeg, Man., Canada.

DAVIS, George L., Jr. (*A* 1938) Estimator, R. L. Spitzley Heating Co., 1200 W. Fort St., Detroit 26, and • 3636 Bedford St., Detroit 24, Mich.

DAVIS, John C. (*M* 1945) Pres. & Mgr., • J. C. Davis, Ltd., 408 New Hargrave Bldg., Winnipeg, and 152 La Verandrye St., St. Boniface, Man., Canada.

DAVIS, John T. (*M* 1945) Sales Engr., The Trane Co., and • 1836 Sul Ross, Houston 6, Texas.

DAVIS, Joseph (*M* 1927; *A* 1926) Htg. Engr. & Contr., • 120 W. Tupper St., Buffalo 1, and 166 Huntington Ave., Buffalo, N. Y.

DAVIS, Keith T. (*M* 1937) • Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, and 2984 Claremont Rd., Shaker Heights 22, Ohio.

DAVIS, Leonard V. (*A* 1945) Research & Design Engr., Russell Stovers Candies of Denver, 2015 Arapahoe St., and • 17 E. Maple St., Denver, Colo.

DAVIS, Robert H. (*A* 1945) Sales Engr., • Johnson Service Co., 507 E. Michigan St., and 633 N. Fourth St., Milwaukee, Wis.

DAVIS, Telford R. (*M* 1942) Cons. Engr., • Ammerman, Davis & Stout, Inc., 805 K. P. Bldg., Indianapolis 4, Ind.

DAVIS, Wayne M. (*A* 1944) Field Engr., Field Control Div., • Conco Engineering Works, Mendota, Ill., and 1401 Manor Ave., Box 990, Munice, Ind.

DAWSON, Eugene F. (*M* 1934) Dir., School of Mech. Engrg., • University of Oklahoma, and 719 Chautauqua St., Norman, Okla.

DAWSON, Fred C. (*M* 1944) Engr., B. F. Sturtevant Co., 89 Broad St., Boston, and • 7 Sutcliffe Ave., Canton, Mass.

DAY, Harold C. (*A* 1934) Mgr., • American Radiator & Standard Sanitary Corp., 1807 Elmwood Ave., Buffalo 7, and 223 Woodcrest Blvd., Kenmore, N. Y.

DAY, Irving M. (*A* 1936) Sales Engr., • Binks Manufacturing Co., 718 Mills Bldg., Washington 6, D. C., and 405 Cumberland Ave., Chevy Chase 15, Md.

DAY, Sidney S. (*A* 1945) Owner, • Sidney S. Day, 3333 N. Vancouver Ave., Portland 12, and 447 N. E. Hazelfern Pl., Portland 15, Ore.

DAY, V. S.* (*M* 1924) Prod. & Price Controller, • Carrier Corp., S. Geddes St., and 316 Highland Ave., Syracuse, N. Y.

DEAN, Carl H. (*M* 1936) Htg. Engr., • Oklahoma Natural Gas Co., Box 871, and 1532 East 35th St., Tulsa, Okla.

DEAN, Charles L. (*M* 1932) Assoc. Prof. of Mech. Engrg., University of Wisconsin, 305 University Extension Bldg., Madison 5, and • 102 Grand Ave., Madison, Wis.

DEAN, David (*M* 1943) W. O. (j.g.) A. A. F., W-2139554, Adjutant, 12th Army Air Force, Emergency Rescue Boat Squadron, A. P. O. 825, c/o Postmaster, New Orleans, La., and • 171 Radford St., Yonkers 5, N. Y.

DEAN, Edwin C. (*M* 1945) Owner, Russellville Plumbing & Heating Co., 113 E. Main St., and • 901 N. Denver, Russellville, Ark.

DEAN, Frank J., Jr. (*A* 1942; *J* 1935; *S* 1934) Lt. Commander, U. S. N. R., 1210 Navy Dept., Washington, D. C., and • 2822 S. Abingdon St., Arlington, Va.

DEAN, Marshall H. (*A* 1946; *J* 1938; *S* 1936) Secy.-Treas., • Temperature Engineering Corp., 1338 McGee St., Kansas City 6, and 6709 Cherry, Kansas City 5, Mo.

DEAN, Vernon C. (*M* 1945) Cons. Engr., • 603 Murray Bldg., Grand Rapids 2, and 1812 Martin S.E., Grand Rapids, Mich.

DeBRA, Frederick B. (*A* 1945) Owner, • Fred. B. DeBra Co., 223 E. Eighth St., and 6217 Cambridge Ave., Cincinnati, Ohio.

DEES, Leonard L. (*M* 1945) Owner • Kansas Sheet Metal Co., 206 W. Sixth, and 2529 Maryland, Topeka, Kans.

DEEVES, Edward W. (*A* 1944; *J* 1940) Partner, • Fred Deeves & Sons, 1422A 17th Ave. W., and 2403-33rd St. W., Calgary, Alta., Canada.

DeFLON, James G. (*A* 1946; *J* 1942) Cooling Tower Engr., The Fluor Corp., 2500 S. Atlantic Blvd., and • 2563 Fidelia St., Los Angeles 22, Calif.

DEGILIO, Louis (*J* 1943) Owner, • Artic Refrigeration Service, 35 Kimberley Ave., 202 Cedar St. S., Timmins, Ont., Canada.

DEHLER, Frank C. (*M* 1944) Chemical Engr., • The Davison Chemical Corp., 20 Hopkins Pl., Baltimore 3, and 3439 Guilford Terrace, Baltimore 18, Md.

DeJARLAIS, Gordon M. (*A* 1946; *J* 1945) Test Engr., Seeger-Sunbeam Corp., Evansville, and • 735 E. Powell, Evansville 13, Ind.

DELALANDE, Andre (*M* 1945) Cons. Engr. • 51 rue la Fontaine, Paris (XVIe), France.

DeLAND, Charles W. (*M* 1924; *J* 1923) Vice-Pres., • C. W. Johnson, Inc., 211 N. Desplaines St., and 2021 Estes Ave., Chicago, Ill.

DeLAUREAL, W. David (*A* 1946; *J* 1940) Chief Engr., Sidney N. Pratt Sheet Metal Works, 2606 Toulouse St., and • 2036 Broadway, New Orleans, La.

DeLUCIA, F. Arthur (*A* 1945) Engr.-Estimator, • A. J. Mongillo Co., 928 Grand Ave., and 325 Willow St., New Haven 11, Conn.

DEMAREST, Richard T. (*J* 1938) Personnel Mgr., • Fitzgibbons Boiler Co., Inc., 23 Mercer St., and Hotel Pontiac, Oswego, N. Y.

DEMETER, Julius (*A* 1939) Cons. Engr., Huerfanos 979—Oficina 605, and • Casilla 9209, Santiago, Chile, S. A.

DEMING, Roy E. (*M* 1941; *A* 1939) Designing Engr., Furnaces, • Kalamazoo Stove & Furnace Co., Rochester Ave., Kalamazoo, Mich.

DEMPSEY, Edward P. (*M* 1945) Htg. & Vent. Engr., The Glenn Martin-Nebraska Co., Box 1269, Omaha 1, and • 2431 Camden Ave., Omaha 11, Nebr.

DEMPSEY, Stephen J. (*A* 1938) Owner, • Stephen J. Dempsey Co., 79 Harvard St., P. O. Box 714, Battle Creek, Mich.

DeNEILLE, J. Lawrence (*M* 1943) Pres., • Eichler Co., 2125 Locust St., St. Louis, and 7227 Maryland Ave., University City, Mo.

DENHAM, Howard S. (*M* 1939) Mech. Checker, Stone & Webster, 149 Federal St., Boston, and • 80 Dexter St., Malden, Mass.

DENNY, Harold R. (*A* 1934) Eastern Merchandise Mgr., • American Blower Corp., 50 West 40th St., New York, N. Y., and 429 Edgewood Ave., Westfield, N. J.

DENSON, Walter, Jr. (*A* 1944) Secy.-Treas., • Walter Denson & Son, 902 N. Myrtle Ave., and 2730 Cherokee Ave., Jacksonville, Fla.

DEPPMANN, Ray L. (*A* 1937) Owner, • R. L. Deppmann Co., 5853 Hamilton Ave., and 13201 Cloverlawn Ave., Detroit, Mich.

DeROO, William C. (*A* 1945) Engr., • Hart & Cooley Manufacturing Co., East Eighth St., and 567 Central Ave., Holland, Mich.

DeSALES, Monteiro, Jr. (*M* 1939) Engr., Isnard & Cia, Rue Lavradio, 67, and • P. O. Box 3963, Rio de Janeiro, Brazil, S. A.

DeSOMMA, A. Edward (*A* 1943; *J* 1937) Sr. Engr., Naval Archt., Bureau of Ships, U. S. N., Washington, D. C., and • 48-29 (a) South 28th St., Arlington, Va.

DETERLING, W. C. (*A* 1937) • General Electric Co., 570 Lexington Ave., New York, and 32 W. Milton St., Freeport, L. I., N. Y.

DETWEILER, John R. (*J* 1945) Dist. Repr., • Minneapolis-Honeywell Regulator Co., 633 S. Fifth St., and R-7 Greentree Manor, Louisville, Ky.

DEUBEL, Justin A. (*M* 1945) Dvlp. Engr., Perfex Corporation, 500 W. Oklahoma Ave., Milwaukee 7, and • 3710 South 46th Pl., Milwaukee 14, Wis.

DEUTH, Gerald Q. (*A* 1944) Pvt., U. S. Army, • Co. C, 150th Bn., 91st Regt., I. R. T. C., Camp Hood, Texas, and 39 West 11th Ave., Columbus, Ohio.

DEVER, Henry F. (*M* 1936; *A* 1935) Pres., • Brown Instrument Co., Wayne & Roberts Aves., Philadelphia 44, and 544 Howe Rd., Merion, Pa.

DEVERALL, Charles R. (*A* 1945; *J* 1944) Engr., Philip Carey Manufacturing Co., Lockland, Cincinnati 15, Ohio.

DEVLIN, John (*M* 1940) Partner, • Devlin Brothers, 1003 Maritime Bldg., New Orleans 12, and 5100 Pitt St., New Orleans, La.

DEVORE, Angus B. (*A* 1937) Staff Engr., • James A. Messer Co., 1206 K St. N.W., Washington 5, and 4817-36th St. N.W., Washington, D. C.

DeVRIES, Don E. (*A* 1944) Supvsr. of Process Engrg., • Nash-Kelvinator Corp., 1545 Clyde Park Ave. S.W., and 1149 Cadillac Dr. S.E., Grand Rapids, Mich.

DeWITT, Earl S. (*A* 1936) Branch Mgr., • American Blower Corp., 1211 Commercial Bank Bldg., and 2329 Westfield Rd., Charlotte, N. C.

DIAMOND, David D. (*A* 1942; *J* 1937) Staff Sgt., • Co. A, S. T. C., Fort Monmouth, N. J., and 1516 Sargent St., St. Paul, Minn.

DIBBLE, S. E.* (*M* 1917), (*Presidential Member*), (Pres., 1925; 1st Vice-Pres., 1924; 2nd Vice-Pres., 1923; Council, 1921-1926) Prof., Supt., • Patton Masonic School for Boys, Patton School, Elizabethtown, Pa.

DICK, Harold S. (*J* 1942; *S* 1940) Engr., • Keystone Sheet Metal Works, 319 Academy St., Newark 3, and 1235 Park Ave., Hoboken, N. J.

DICKASON, Gray D. (*M* 1938) Pres.-Treas., • Genesee Heating Service, Inc., 950 Mercantile Bldg., Rochester 4, and 140 Windemere Rd., Rochester, N. Y.

DICKENS, Lester A. (*A* 1941) Owner, • Dickens & Co., 1794 S. Taylor Rd., and 3710 Grosvenor Rd., Cleveland Heights 18, Ohio.

DICKENSON, Malcolm E. (*M* 1936) Pres., • Livingston Stoker Co., Ltd., 33 Sanford Ave. S., and 964 Cumberland Ave., Hamilton, Ont., Canada.

DICKEY, Arthur J. (*M* 1921) Vice-Pres. & Gen. Mgr., C. A. Dunham Co., Ltd., 1523 Davenport Rd., Toronto 4, and • 9 Mossom Pl., Toronto 3, Ont., Canada.

DICKINSON, Neville S. (*A* 1943) Pres., • Motor Sales & Engineering Co., Inc., 1060 Broad St., Newark 2, and 507 Richmond Ave., Maplewood, N. J.

DICKINSON, Robert P., Jr. (*A* 1946; *J* 1938) Sales Engr., • Minneapolis-Honeywell Regulator Co., 405 Penn Ave., Pittsburgh, and 521 S. Lang Ave., Pittsburgh 8, Pa.

DICKSON, Donald R. (*S* 1941) Capt., A. C., U. S. Army Air Forces, 340th N. T. S., Ellington Field, and • Box 234 B. S., A. A. F., Big Spring, Texas.

DICKSON, George P. (*M* 1936) Mgr., B. F. Sturtevant Co., Haddon and Crestmont Ave., Camden, and • 430 Cooper St., Woodbury, N. J.

DICKSON, Robert B. (*M* 1919) Pres., • Kewanee Boiler Corp., and 145 E. Division St., Kewanee, Ill.

DICKSON Robert W., Jr. (*A* 1943; *J* 1938) American Blower Corp., 1302 Swetland Bldg., Cleveland 15, and • 880 Elbon Rd., Cleveland Heights 21, Ohio.

DIETER, George H. (*M* 1941) Dist. Engr., • The Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, and 618 N. Kenwood St., Glendale 6, Calif.

DIETZ, C. Fred (*M* 1938) • Haynes-Blankin Corp., 1124 Spring Garden St., Philadelphia 23, and 1215 Allengrove St., Philadelphia 24, Pa.

DILL, Richard S.* (*M* 1939) Chief, Heat Transfer Sect., • National Bureau of Standards, Washington, D. C., and 1603 S. Springwood Dr., Silver Spring, Md.

DILLENDER, Eugene A. (*M* 1939) • 1502 McCullough, San Antonio 2, Texas.

DINHAM, Robert E. (*S* 1943) Lt. (j.g.) U.S.N.R., and • 4412 W. Lake Harriet Blvd., Apt. 102, Minneapolis, Minn.

DION, A. M. (*M* 1937) Sales Engr., Trane Co. of Canada, King and Mowat Sts., and • 342 St. Clair Ave., Toronto, Ont., Canada.

DIRECTOR, Isadore (*A* 1944) Application Engr., and • 6738 Lynford, St., Philadelphia 24, Pa.

DISNEY, Melvin A. (*M* 1942; *A* 1934) Mgr., Air Cond. & New Constr. Sales Div., The Laclede Gas Light Co., 1017 Olive St., St. Louis 1; and • R. R. 1, Box 556, Glencoe, Mo.

DISTEL, Robert E. (*M* 1941; *J* 1938) Partner & Gen. Mgr., • Distel Heating Co., 720 E. Shiawassee St., P. O. Box 133, Lansing 1, and 547 Bailey St., East Lansing, Mich.

DIVER, M. L. (*M* 1925) Cons. Engr., • P. O. Box 1016 San Antonio 6, Texas.

DIXON, Arthur G. (*M* 1928) Secy., • Modine Manufacturing Co., and 442 Wolff St., Racine, Wis.

DOCKERAY, F. A. (*A* 1945) Sales Repr., • Fitzgibbons Boiler Co., 1717 Sansom St., Philadelphia 3, and 225 North 63rd St., Philadelphia 39, Pa.

DODD, John A. (*M* 1944) Owner, • John A. Dodd Co., 299 Techwood Dr. N.W., and 3370 Nancy's Creek Rd., Atlanta, Ga.

DODDS, Forrest F. (*M* 1920) Mgr., Kansas City Sales Office, • American Radiator & Standard Sanitary Corp., 503-6 National Fidelity Life Bldg., 1002 Walnut St., Kansas City 6, and 4600 Mill Creek Pkway., Kansas City 2, Mo.

DODGE, Harry A. (*M* 1936) Elec. Engr., S. H. Kress & Co., 114 Fifth Ave., New York 11, and • 514 West End Ave., New York 24, N. Y.

DOERING, Frank L. (*M* 1919) • American Radiator & Standard Sanitary Corp., 238 Boston Ave., Lynchburg, Va.

DOERRFUSS, Harry W. (*M* 1944) Sales Engr., • Warren Webster Co., 26 South 20th St., Philadelphia 3, Pa., and 35 Harvard Rd., Audubon, N. J.

DOHERTY, John J. (*A* 1942) • Fells Plumbing, Heating & Builders' Supplies, Inc., 654 Main St., and 10 Maple Rd., Winchester, Mass.

DOHERTY, Roger J. (*A* 1945) Vice-Pres., Shaw-Perkins Manufacturing Co., 1645 Oliver Bldg., Pittsburgh, and • 363 Lehigh Ave., EE. Pittsburgh 6, Pa.

DOING, L. J. (*M.* 1944) Mech. Engr., 8th Service Command, Santa Fe Bldg., Dallas, and ● 1104 Elsbeth, Dallas 8, Texas.

DOLAN, H. P. (*M.*, 1944) Director of Engrg. & Operation of Bldgs., Board of Education, 1354 Broadway, Detroit 26, and ● 14419 Strathmoor Ave., Detroit 27, Mich.

DOLAN, Raymond G. (*M.* 1926, *J.* 1922) Secy.-Treas. ● Tom Dolan Heating Co., Inc., 614-16 W. Grand, Oklahoma City 2, and 708 Northwest 40th Oklahoma City, Okla.

DOLAN, William H. (*A.* 1941) Pres., ● The Jennison Co., 17 Putnam St., and 65 Highland Ave., Fitchburg, Mass.

DOLGE, Emil A. (*A.* 1945) Mgr., Htg. Dept., A. Y. McDonald Manufacturing Co., and ● P. O. Box 976, Des Moines 4, Iowa.

DOME, Alan G. (*M.* 1945; *A.* 1938; *J.* 1936) Disposal Agent, United States Maritime Commission, Philadelphia, Pa., and ● 314 E. Allens Lane, Philadelphia 19, Pa.

DOMINY, Charles B. (*J.* 1942) U. S. N. R. ● U. S. S. Hammondsport A. K. V.-2, c/o Fleet Post Office, San Francisco, Calif.

DONCEEL, Guillaume T. (*M.* 1944) Indus. Engr., ● Oklahoma Natural Gas Co., P. O. Box 1620, Oklahoma City 1, and 1727 Northwest 13th St., Oklahoma City 6, Okla.

DONELSON, William N. (*M.* 1943; *J.* 1937) Engr., ● *Modine Manufacturing Co., Great Lakes Life* Bldg., and 4461 Glenview, Warrensville Heights, Cleveland, Ohio.

DONNELLY, James A.* (*Life Member; M.* 1904) (Treas., 1912-14), (Board of Governors, 1913), (Council, 1914) ● Largent, West Va.

DONNELLY, Joseph F., Sr. (*M.* 1943) Combus. Engr., Dist. Sales Mgr., Cleveland Steel Products Corp., 7304 Madison Ave., Cleveland, Ohio, and ● 409 Highland Ave., Upper Montclair, N. J.

DONNELLY, Russell (*M.* 1923) Sales Engr., ● Nash Engineering Co., 420 Lexington Ave., New York 17, N. Y.

DONOHOE, Charles F. (*M.* 1941) Tech. Engr., Central Heating Dept., ● The Detroit Edison Co., 2000 Second Ave., Detroit 26, and 10065 Lincoln Dr., Huntington Woods, Mich.

DONOHOE, John B. (*A.* 1937, *J.* 1935) Engr. & Estimator, ● B. F. Donohoe & Co., 51 Albany St., Boston 11, and 23 Primrose St., Roselindale 31, Mass.

DONOVAN, William J. (*A.* 1930) ● 2239 North 67th St., Philadelphia, Pa.

DORFAN, Morton I. (*M.* 1929) Cons. Engr., 336 Fourth Ave., Pittsburgh, and ● 1217 Malvern Ave., Pittsburgh 17, Pa.

DORNHEIM, O. A. (*M.* 1912; *J.* 1906) Vice-Pres., Buensod-Stacey, Inc., 60 East 42nd St., New York 17, and ● 15 Hamilton Ave., Bronxville 8, N. Y.

DORSEY, Francis C. (*M.* 1920) Pres., ● Francis C. Dorsey, Inc., 4520 Schenley Rd., and 4505 Roland Ave., Baltimore, Md.

DOSTER, Alexis (*A.* 1934) Vice-Pres. & Secy. ● The Torrington Manufacturing Co., 70 Franklin St., Torrington, and Litchfield, Conn.

DOTSON, Russell F. (*A.* 1945) Dist. Mgr., Bastian-Morley Co., Inc., La Porte, Ind., and ● 1596 Barclay Pl. N.E., Atlanta, Ga.

DOUGHTY, Charles J. (*M.* 1925) Mech. Supvsr. of Constr. & Design, ● The Austin Co., 19 Rector St., and Murray Hill Hotel, Park Ave. and 40th St., New York, N. Y.

DOUGLASS, Thomas C. (*M.* 1943) Secy., ● *Pacific Electrical & Mechanical Co., Inc., 16th* at Vermont, San Francisco 3, and 1400 Alvarado Ave., Burlingame, Calif.

DOVENER, Robert F. (*A.* 1941) Engr., ● Atchison & Keller, 1246 Taylor St. N.W., Washington 11, D. C., and 4365 N. Pershing Dr., Arlington, Va.

DOVER, Donald E. (*J.* 1945) Refrig. Engr., ● Carrier Corp., 405 Lexington Ave., New York, N. Y., and Middlebury, Ind.

DOWDELL, J. R. (*A.* 1941) Owner, ● J. R. Dowdell & Co., 1003 Southwestern Life Bldg., Dallas 1, and 4400 Windsor Pkway., Dallas 5, Texas.

DOWDY, Rufus B. (*M.* 1939) Sales Engr., ● Haydn Myer Co., Inc., P. O. Box 746, and 203 Capitol Pkway., Montgomery, Ala.

DOWLER, Edward A. (*M.* 1937) Major, Canadian Army, ● 9 Prince Arthur Ave., Toronto, Ont., Canada.

DOWNE, Edward R. (*M.* 1927) Engr., C. A. Olsen Manufacturing Co., and ● 261 E. Broad St., Elyria, Ohio.

DOWNES, Alfred H. (*A.* 1937) Draftsman, ● Gay Engineering Co., 2730 East 11th St., Los Angeles 23, and 1342½ Bond St., Los Angeles 15, Calif.

DOWNES, H. H. (*M.* 1923) S. E. Div. Mgr., ● American Blower Corp., 438 Woodward Bldg., Washington 5, D. C.

DOWNES, Nate W. (*M.* 1917), (Council, 1928-30) Asst. Supt. in charge of Bldgs. & Grounds, ● School Dist. of Kansas City, Mo., 1840 E. Eighth St., Kansas City 1, and 2119 East 68th St., Kansas City 5, Mo.

DOWNS, Charles R., PhD. (*M.* 1936) Cons. Chem. Engr., ● Chemists' Bldg., 50 East 41st St., New York 17, N. Y., and 5 Sylvan Lane, Old Greenwich, Conn.

DOWNS, Sewell H. (*M.* 1931), (*Presidential Member*), (Pres., 1944; 1st Vice-Pres., 1943; 2nd Vice-Pres., 1942; Council, 1936-45) Vice-Pres., Clarage Fan Co., Kalamazoo, and ● 1562 Spruce Dr., Kalamazoo 40, Mich.

DRAKE, George M. (*A.* 1940; *J.* 1936) Cons. Engr. ● Poirier & McLane Corp., 33 West 42nd St., New York 18, N. Y., and Bayberry Lane, Westport, Conn.

DRAVAGE, Thom D. (*A.* 1945) Htg. & Air Cond. Engr., McGeorge & Hargett, 9400 Quincy Ave., Cleveland, and ● *Chillicothe Rd., R. 2, Willoughby*, Ohio.

DRESSELL, Russell E. (*M.* 1942; *A.* 1938) Mech. Engr., Riggs, Distler Co., Inc., 216 N. Calvert St., and ● 920 E. Preston St., Baltimore, Md.

DRIEMEYER, R. C. (*A.* 1942; *J.* 1937) Lt., U. S. N. R., and ● 5410 Vernon Ave., St. Louis 12, Mo.

DRING, Robert W. (*A.* 1945) Partner, ● R. W. Dring & Son, 614 Monroe St., and 101 South Ave., La Porte, Ind.

DRINKER, Philip* (*M.* 1922) Prof. of Indus. Hygiene, ● Harvard University, School of Public Health, 55 Shattuck St., Boston, and 12 Hubbard Park, Cambridge, Mass.

DRISCOLL, William H.* (*Life Member; M.* 1904), (*Presidential Member*), (Pres., 1926; 1st Vice-Pres., 1925; 2nd Vice-Pres., 1924; Treas., 1923; Council, 1918-27) Vice-Pres., ● Carrier Corp., 300 S. Geddes St., Syracuse, N. Y., and 50 Glenwood Ave., Jersey City, N. J.

DROBA, Charles B. (*M.* 1944) Sales Engr. & Mgr., ● Ehret Magnesia Manufacturing Co., 312 Seventh St., S.W., and 3412 Oliver St., N.W., Washington, D. C.

DROPKIN, David* (*M.* 1942) Asst. Prof. of Mech. Engrg., ● College of Engineering, Cornell University, and Slaterville Rd., Ithaca, N. Y.

DRUM, Leo J., Jr. (*A.* 1946; *J.* 1939) ● 7 Gilmer Ave., Montgomery, Ala.

DRUM, Robert Erin (*A.* 1944) General Foreman, Air Cond., Allison Div., General Motors Corp., Speedway City, and ● 5740 Oak Ave., Indianapolis 1, Ind.

DUBE, John E. (*M.* 1944) Vice-Pres. & Gen. Mgr., ● Alco Valve Co., 865 Kingsland Ave., St. Louis 5, and R. 2, Box 142, Chesterfield, Mo.

DuBOIS, Louis J. (*M.* 1931) York Corp. and ● 331 Pine Lane, Hillcroft, York, Pa.

DUBRY, Ernest E. (*M.* 1924) Supt., Central Htg., Detroit Edison Co., 2000 Second Ave., Detroit 26, and ● 9116 Dexter Blvd., Detroit 6, Mich.

DuCHATEAU, Manuel F. (*A.* 1942; *J.* 1938) Mgr., Greensboro Branch, ● Crane Co., 205 S. Lyndon St., Greensboro, N. C.

DUDLEY, William H., Jr. (*M.* 1945; *A.* 1940) Dist. Mgr., The Trane Co., New Orleans, and ● 340 Audubon Blvd., New Orleans 15, La.

DUFAULT, Felix H. (*A.* 1936) Mgr., Furnace Div., ● General Steel Wares, Ltd., 2355 Delisle St., Montreal 2, and 5115 Bordeaux St., Apt. 6, Montreal 34, Que., Canada.

DUGAN, Thomas M. (*M.* 1920) Sanitary Htg. Engr., and ● 1308 Freemont St., McKeesport, Pa.

DUGGER, Earl R. (*A.* 1945) Chief Engr., ● Oklahoma Refrigerating Co., 1908 N. Broadway, Oklahoma City 3, and 2559 Northwest 18th, Oklahoma City 7, Okla.

DUKEHART, Morton McI. (*M.* 1942) Owner, ● Morton McI. Dukehart & Co., 801 Maryland Trust Bldg., Baltimore 2, and 419 Woodlawn Rd., Baltimore 10, Md.

DULLE, Wilfred L. (*A.* 1942; *J.* 1936) Vice-Pres., E. E. Souther Iron Co., 1952 Kienlen Ave., St. Louis 20, and ● 7230 Normandy Pl., Normandy, 21, Mo.

DUNBAR, Leon W. (*M.* 1944) Dist. Mgr., ● Clarage Fan Co., 314 Bulkley Bldg., Cleveland, and 1795 Cadwell Ave., Cleveland Heights, Ohio.

DUNHAM, Clayton A.* (*Life Member; M.* 1911) Pres., ● C. A. Dunham Co., 450 E. Ohio St., Chicago, and 150 Maple Hill Rd., Glencoe, Ill.

DUNLAP, R. L. (*M.* 1944) John H. Kitchen & Co., 1016 Baltimore Ave., and ● 5533 Holmes, Kansas City, Mo.

DUNN, Robert (*A.* 1944) Chief Engr., ● The Canadian Bank of Commerce, 25 King St. W., Toronto 1, and 43 Pacific Ave., Toronto 9, Ont., Canada.

DUNNE, Russell V. D. (*M.* 1937) ● Carrier Corp., Chrysler Bldg., New York, and 210 Robineau Rd., Syracuse, N. Y.

DUPLANT, Jean L. (*A.* 1940) ● Maskati Court, Queens Rd., Bombay, India, and 137-43 219th St., Springfield Gardens, L. I., N. Y.

DUPUIS, Joseph E. R. (*M.* 1942) Branch Mgr., ● Trane Co. of Canada, Ltd., 660 St. Catherine St. W., Montreal, and 5642 Queen Mary Rd., Hampstead, Que., Canada.

DUSOSSOIT, Edmond A. (*M.* 1944) Treas., ● Lynch & Woodward, Inc., 28 Oak St., Boston, and 173 Warren St., Newton Centre, Mass.

DUTTON, Gayle R. (*M.* 1945) Partner, ● Western Fiberglas Supply, Ltd., 739 Bryant St., San Francisco 7, and 215 Coleridge Ave., Palo Alto, Calif.

DWYER, Patrick (*A.* 1945) Draftsman, Richard Crittall & Co., Ltd., Aldwych House, London, and ● Blacklake Cottage, Tilford Rd., Farnham, Surrey, England.

DWYER, Thomas F. (*M.* 1923) Chief, Htg. & Vtg. Div., ● Board of Education, 49 Flatbush Ave., Brooklyn, and 82 Iris Ave., Floral Park, L. I., N. Y.

DWYER, William H. (*A.* 1944) Sales Mgr., ● Central Supply Co., 210-230 S. Capitol Ave., and 4455 Marcy Lane, Apt. 164, Indianapolis, Ind.

DYER, Alvin R. (*A.* 1944) Mgr., Htg. & Air Cond. Dept., Utah Builders Supply Co., 503 W. Fourth South St., Salt Lake City 4, and ● 2005 Herbert Ave., Salt Lake City 5, Utah.

DYER, Wilfrid S. (*A.* 1939) Partner, ● H. W. Dyer & Son, 92 Byron St., Battle Creek, Mich.

DYKES, James B. (*A.* 1939; *J.* 1936) Vice-Pres., ● T. A. Morrison & Co., Ltd., 1070 Bleury St., Montreal, I., and 42 Dobie Ave., Mt. Royal, Montreal 16, Que., Canada.

E

EADIE, John G. (*Life Member; M.* 1909) Cons. Engr., ● Eadie, Freund and Campbell, 500 Fifth Ave., Rm. 1702, New York 18, and 11 Blackburn Rd., Summit, N. J.

EAGLETON, Sterling P. (*M.* 1936) Chief Engr. & Bldg. Supt., National Gallery of Art, Fourth and Constitution Ave., Washington, D. C., and ● 615 St. Andrews Lane, Silver Spring, Md.

EARHART, Joe S. (*M.* 1943) Vice-Pres. & Treas., Thermxchanger, Inc., 4232 E. Whiteside St., Los Angeles 33, and ● 845 S. Lorraine Blvd., Los Angeles 5, Calif.

EAST, Roy H. (*A.* 1944) Chief, Fuel & Htg. Section, Office, Service Command Engineer, Headquarters Ninth Service Command, Fort Douglas, and ● 326 E. First South, Salt Lake City, Utah.

EASTMAN, C. B. (*M.* 1932, *J.* 1929) Partner, Sales Engr., ● Coward-Eastman Co., 2206 Chestnut St., Philadelphia 3, and 530 Brookview Lane, Brookline, Delaware, Pa.

EASTWOOD, E. O. (*Life Member; M.* 1921), (*Presidential Member*), (Pres., 1942; 1st Vice-Pres., 1941; 2nd Vice-Pres., 1940; Council 1931-33; 1937-43) Prof. Mech. Engrg., ● University of Washington, Seattle 5, and 4702-12th Ave. N.E., Seattle, Wash.

EASTWOOD, Harry P. (*M.* 1938) Fuel Engr., Philadelphia & Reading Coal & Iron Co., 140 Cedar St., New York, and ● 157 Frankel Blvd., Merrick, N. Y.

EATON, Byron K. (*M.* 1920) Delco Heat Div., General Motors Corp., 157 W. Wacker Dr., Chicago, and ● 311 N. Spring Ave., La Grange, Ill.

EATON, William G. M. (*M.* 1942; *A.* 1934) Sales Engr., Pease Foundry Co., Ltd., 227 Victoria St., Toronto, and ● 300 Wellesley St., Toronto 5, Ont., Canada.

EBERT, William A. (*M.* 1920) Owner, ● Ebert Air Conditioning, 1026 W. Ashby Pl., and 2151 W. Kings Highway, San Antonio 1, Texas.

ECKHARDT, Elroy, Jr. (*J.* 1943) Partner, ● James F. O'Neil Plumbing & Heating Co., 2800 Howard Ave., New Orleans, and 34 Beverly Gardens, Metairie, La.

EDGE, Alfred J. (*M.* 1938) Cons. Engr., Reynolds Smith & Hills, 11-S. Laura St., Jacksonville, and ● 2864 Olga Pl., Jacksonville 5, Fla.

EDMONDSON, John V. (*A.* 1944) Asst. to Mgr., Crane Co., 814 Young St., Dallas 2, and ● 3465 Amherst, Dallas 5, Texas.

EDWARDS, Arthur F. (*A.* 1945) Mech. Designer, Dept. of Pub. Wks., Ont., Provincial Govt., Parliament Bldgs., Toronto, and ● 118 Airdrie Rd., Leaside, Tor. 12, Ont., Canada.

EDWARDS, Arthur W. (*M.* 1936) Dist. Mgr., ● The Trane Co., 626 Broadway, Cincinnati 2, and 3423 Paxton Ave., Cincinnati 8, Ohio.

EDWARDS, C. Eugene (*M.* 1942) Piping & Air Cond. Designer, ● General Delivery, Lake Jackson, and 111 Samuels Ave., Ft. Worth 3, Texas.

EDWARDS, Don J. (*A.* 1933) Sales Repr., ● Lateseel, Inc., 84 State St., Boston, Mass.

EDWARDS, George B. (*A.* 1945) Owner, ● Plbg. & Htg. Contractor, 1140 Ontario Ave., Niagara Falls, N. Y.

EDWARDS, Junius David* (*M.* 1936) Asst. Dir. of Research, ● Aluminum Company of America, Aluminum Research Labs., P. O. Box 772, New Kensington, and 536 Sixth St., Oakmont, Pa.

EDWARDS, Paul A. (*M.* 1919) Pres., ● The G. F. Higgins Co., 608 Wabash Bldg., Pittsburgh, 22, and 3074 Pinehurst Ave., Pittsburgh 16, Pa.

EELLS, H. B. (*M.* 1944) ● Barnes & Jones, 101 Park Ave., New York 17, and 1049 East 27th St., Brooklyn, N. Y.

EGGERS, William K (*S.* 1941) Lt. (j.g.) U.S.N.R., ● U. S. Sub Base, Box 20, New London, Conn.

EHLERS, Jacob (*A.* 1939; *J.* 1937) Chief Engr., ● c/o Jamesons Premier Engineers, Ltd., 33 Salt River Rd., Salt River, Cape Town, South Africa.

EHRENZELLER, Adolph (*M.* 1941) Owner, ● A. Ehrenzeller, 329 Washington St., Dorchester 21, and 23 Parklawn Rd., West Roxbury 32, Mass.

EHRLICH, M. William* (*M.* 1916) ● 56 Ridge Rd., Lyndhurst, N. J.

EICHER, Hubert C. (*M.* 1922) Chief, School Plant Div., Dept. of Public Instruction, and ● 207 North 30th St., Harrisburg, Pa.

EICHOLTZ, Meryl V. (*A.* 1942) Dist. Mgr., Ilg Electric Ventilating Co., 1012 N. Third St., Milwaukee 3, and ● 2045 Pleasant St., Wauwatosa 13, Wis.

EISS, Robert M. (*M.* 1933; *J.* 1930) Process Engr., Kimberly-Clark Corp., P. O. Box 31, and ● Rt. 1, Adella Beach, Neenah, Wis.

EKLUND, Karl G. (*M.* 1938) Cons. Engr., ● Karl G. Eklunds Ingeniorsbyra A. B., Brunkebergstorg 15, Stockholm, and Storangen, Parkgaven 19, Sweden.

ELDRED, Harold (*M.* 1944) Htg. & Vtg. Contractor, ● 15 The Fairway, North Wembley, England.

ELIZARDI, Ralph (*A.* 1943; *J.* 1940) Engr., Air Cond., Cary B. Gamble Canal Bldg., New Orleans 12, and ● 2222 Dublin St., New Orleans 18, La.

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ENGLISH, Harrold (*M* 1935) Pres., ● English & Lauer, Inc., 1978 S. Los Angeles St., Los Angeles 11, and 515 S. Norton Ave., Los Angeles, Calif.

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ERICKSON, Harry H. (*M* 1944; *A* 1929) Sales Engr., ● Haynes-Blankin Corp., 1124 Spring Garden St., Philadelphia 23, and 25 Old Eagle School Rd., Strafford, Pa.

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EZZ-EL-DIN, Kamal (*M* 1944; *A* 1941; *J* 1938) Service Mgr., Carrier-Egypt, S. A. E., 37 Sharia Kasr El Nil, and ● 78 Sharia Helwan, Munira, Cairo, Egypt.

F.

FABLING, Walter D. (*A* 1937) Owner, W. D. Fabling Co., 117 W. Ninth St., Los Angeles, and ● 1424 Van Dyke Rd., San Marino 9, Calif.

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FAHNESTOCK, M. K.* (*M* 1927) Research Prof. & Asst. Dir., ● Engineering Experiment Sta., University of Illinois, 214 M. E. Laboratory, and 702 W. Vermont St., Urbana, Ill.

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FANNING, Erroll C. (*M* 1941) Engr., ● Atlas Heating & Ventilating Co., Ltd., 557 Fourth St., San Francisco 7, and 24 Oval Rd., Oakland 11, Calif.

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FARLEY, Willoughby S. (*A* 1941) Capt., M.A.C., Box 109, Oliver General Hospital, Augusta, Ga., and ● W. S. Farley, Contr. & Engr., 865 Paxton Ave., Danville, Va.

FARNBACHER, John M. (*J* 1945) Student Engr., Chrysler Corp., Airtemp Div., 1119 Leo St., Dayton, and ● 924 N. Euclid Ave., Dayton 7, Ohio.

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FARNUM, Warren S. (*A* 1945) Mgr., Franchise Sales, ● Westinghouse Electric Corp., 89 Broad St., Boston, and 255 Mystic St., Arlington 74, Mass.

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FARRAR, Cecil W. (*M* 1920; *A* 1918), (Treas., 1930; Council, 1930) Sales Mgr., Plbg. Div., Richmond Radiator Co., 19 East 47th St., and ● Murray Hill Hotel, 40th St. and Park Ave., New York 17, N. Y.

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FARROW, E. E. (*A* 1938) E. E. Farrow Co., 6109 Forest Park Rd., Dallas, and ● 1518 Kings Highway, Dallas 11, Texas.

FARROW, Hollis L. (*A* 1942; *J* 1937) Service & Installation Mgr., Sprague Breed Stevens & Newhall, Inc., 153 Broad St., and ● 73 Victory Rd., Lynn, Mass.

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FAULKNER, John H. (*M* 1941) Owner & Mgr., ● John H. Faulkner Co., 805 Skinner Bldg., Seattle 1, and Mercer Island, Wash.

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FAVA, Albert A. (*M* 1945) Plant Engr., ● Atlantic Gelatin Co., Inc., Hill St., Woburn, and 6 Pine St., Stoneham 80, Mass.

FAVRET, Louis E. (*M* 1945) Pres., ● Williamson-Favret Furnace Co., 55 E. Goodale St., Columbus 8, and 314 Oakland Park Ave., Columbus 2, Ohio.

FAXON, Harold C. (*M* 1937) In Service, and ● 1621–19th St. N.W., Washington, D. C.

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FEHLIG, John B., Jr. (*A* 1941) Asst. Mgr., Excelsior Heating Supply Div., 528 Delaware St., Kansas City 6, and ● 6412 Paseo, Kansas City 5, Mo.

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FIDELIUS, Walter R. (*M* 1936) Supt., Combat Tank Div., Fitzgibbons Boiler Co., Inc., and • 69 W. Front St., Red Bank, N. J.

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FISCHER, Frank P. (*A* 1940) Owner, • Frank P. Fischer Engineering Co., 412 Dryades St., New Orleans 13, and 1820 Audubon St., New Orleans 18, La.

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FORBES, H. Burke (*J* 1941; *S* 1938) Lt., U.S.N.R., and • 3218 Balmoral Ave., Chicago 25, Ill.

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FORFAR, D. M. (*M* 1917) Mech. Engr., • Grinnell Co., Inc., 240 Seventh Ave. S., and 4817 Emerson Ave. S., Minneapolis, Minn.

FORRESTER, Charles O. (*A* 1945) Mech. Engr., Federal Public Housing Authority, and • 278-12th St. N.E., Atlanta, Ga.

FORRESTER, Norman J. (*A* 1936) Mgr., Contract Div., The Garth Co., 750 Belair Ave., and • 23 Nelson Ave., Montreal West, Que., Canada.

FORSBERG, William (*M* 1919) Secy., The Hopson & Chapin Manufacturing Co., 231 State St., New London, and • P. O. Box 42, Quaker Hill, Conn.

FORSLUND, Oliver A. (*M* 1936) Owner & Mgr., Forslund Pump & Machinery Co., 1717-19 Main St., Kansas City 8, and • 108th St. and State Line, Kansas City 5, Mo.

FORSYTH, Stuart L. (*M* 1944) Advisory Engr., • Westinghouse Electric Corp., 5915 Green St., Emeryville 8, and 1654 Capistrand Ave., Berkeley 6, Calif.

FOSS, Benjamin S., Jr. (*A* 1944) Field Engr., • B. F. Sturtevant Co., 401 Magnolia Bldg., Dallas 1, and 4400 Winston Court, Dallas 9, Texas.

FOSS, Edwin R. (*A* 1936) Dist. Mgr., • The Powers Regulator Co., 135 Luckie St. N.W., Atlanta 3, and 257 Bolling Rd. N.E., Atlanta, Ga.

FOSTER, Charles (*Life Member*; *M* 1923) Cons. Engr., • 316 Medical Arts Bldg., Duluth 2, and 2831 E. First St., Duluth, Minn.

FOSTER, John G. (*J* 1938) Lt. Col., 39th Photo Recon. Sq., Santa Maria A. A. F., Santa Maria, Calif., and • 2635 Sedgwick Ave., New York, N. Y.

FOULDS, P. A. L. (*Life Member*; *M* 1916) Partner, • Hubbard, Rickerd & Blakeley, 84 State St., Boston, and 72 Whitin Ave., Revere, Mass.

FOWLES, Harry H. (*A* 1940; *J* 1934) Engr. & Estimator, V. J. Kenneally Co., 104 Hanover St., Boston, Mass., and • 4 Haskell St., Auburn, Maine.

FOX, Ernest (*M* 1935) Asst. Engr., • C. A. Dunham Co., Ltd., 1523 Davenport Rd., Toronto 4, and 531 Rushton Rd., Toronto, Ont., Canada.

FOX, Ernest C. (*M* 1944) Mgr., Htg. & Cooling Dept., • Hager & Cove Lumber Co., 1125 S. Pennsylvania Ave., and 928 W. Michigan Ave., Lansing, Mich.

FOX, Francis J. (*A* 1945) Owner, • Gas Heat Engineering Co., 3020 E. Franklin Ave., Minneapolis, and 3501-11th Ave. S., Minneapolis 7, Minn.

FOX, John H. (*M* 1935) Sales Engr., Minneapolis-Honeywell Regulator Co., Ltd., 117 Peter St., and • 37 MacDonell Ave., Toronto, Ont., Canada.

FOX, Winford L. (*A* 1944) Engr., U. S. Machine Corp., Lebanon, and • 2035 N. Delaware St., Indianapolis, Ind.

FRANCK, Peter (*J* 1938) Rogers Diesel & Aircraft Corp., 1120 Leggett Ave., Bronx, and • 17 Tecumseh Ave., Mt. Vernon, N. Y.

FRANK, George D. (*M* 1945) Mgr., Industrial Div., York-Shipley, Inc., and • 1303 W. Philadelphia St., York, Pa.

FRANK, John M. (*M* 1918; *A* 1912) Pres., • Ilg Electric Ventilating Co., 2850 N. Crawford Ave., Chicago, and 1152 Chatfield Rd., Hubbard Woods, Ill.

FRANKEL, Gilbert S. (*M* 1926) Mgr., Federal & Marine Dept., • Buffalo Forge Co., 512 Woodward Bldg., Washington 5, D. C., and 5800 Kirkside Dr., Chevy Chase, Md.

FRANKFURT, W. W. (*M* 1943) Asst. Plant Engr., • Douglas Aircraft Co., Inc., and 1524 Northwest 38th, Oklahoma City, Okla.

FRANKLE, Harry R. (*M* 1941) Partner, • Midwest Air Control, 707 Locust St., Des Moines 9, and 547-44th St., Des Moines, Iowa.

FRANKLIN, Ralph S. (*M* 1919) • 320 Grove St., Melrose, Mass.

FRANKLIN, Sam H., Jr. (*A* 1938) Major, Overseas, and • 204 Colonial Court, Lynchburg, Va.

FRANTZ, Adolph R. (*M* 1944) Service Mgr., Duo-Therm Div., Motor Wheel Corp., Lansing, and • 1206 W. Michigan Ave., Lansing 15, Mich.

FRASER, James J. (*A* 1936) Mgng. Dir., • Honeywell-Brown, Ltd., Wadsworth Rd., Perivale, Greenford, Middlesex, England.

FRAZIER, J. Earl (*A* 1936) Pres., • Frazier-Simplex, Inc., 436 E. Beau St., and 7 Wilmont Ave., Washington, Pa.

FREDERICK, Holmes W. (*M* 1937) Asst. Dir. Dept. of Buildings & Grounds, • 301 University Hall, Columbia University, New York 27, N. Y., and R. D. 1, Box 3, Riverside Ave., Riverside, Conn.

FREEMAN, Alfred W. (*J* 1940; *S* 1939) Capt., Air Corps, • 999th Base Unit (Hq. A. F. T. A. C.), Orlando, Fla., and 3105-88th St., Jackson Heights, L. I., N. Y.

FREEMAN, Casstus W. (*M* 1945) Chief Designer & Assoc., • Paul D. Bemis, 36 Pearl St., Hartford 3, and 290 Forest St., East Hartford, Conn.

FREEMAN, Edwin M. (*A* 1937) Vice-Pres., Canadian Asbestos Co., 322 Youville Sq., and • 685 Rockland Ave., Montreal, Que., Canada.

FREEMAN, J. Albert (*A* 1940; *J* 1933) Partner Engr., • Western Engineering Co., 2105 S.E. Ninth Ave., Portland 14, and 4456 S.W. Dosch Rd., Portland 11, Ore.

FREIJE, W. F. (*M* 1943) Engr., Mgr., Refrig. & Air Cond. Div., • Hayes-Brothers, Inc., 236 W. Vermont St., Indianapolis 4, and 5411 W. Morris St., Indianapolis 8, Ind.

FREITAG, Frederic G. (*M* 1932) • 64 Elm Ave., Mt. Vernon, N. Y.

FREITES, Andres A. (*J* 1945) Jr. Engr., • Carrier Corp., 300 S. Geddes St. and 1011 Walnut Ave., Syracuse, N. Y.

FRENCH, Donald (*M* 1926) Vice-Pres., • Carrier Corp., Syracuse 1, and Cazenovia, N. Y.

FRENTZEL, Herman C. (*M* 1936) Cons. Engr., Frentzel Engineering Co., 724 E. Mason St., Milwaukee 2, and • 4365 N. Wildwood Ave., Milwaukee 11, Wis.

FREYDER, G. Gill (*A* 1945) Air Cond. Engr., • Commonwealth Edison Co., 72 W. Adams St., Chicago 90, and 131 Clinton St., Park Ridge, Ill.

FRIEDLER, Joseph J., Jr. (*M* 1940) Dist. Mgr., • Ilg Electric Ventilating Co., 304 Natchez Bldg., New Orleans 12, and 500 Audubon Blvd., New Orleans 15, La.

FRIEDLIEB, Morton J. (*J* 1942) Plant Engr., Super Electric Products Corp., 1057 Summit Ave., Jersey City, N. J., and • 65-60 Booth St., Forest Hills, N. Y.

FRIEDLINE, James M. (*A* 1942; *J* 1937) Capt., 0317228, Hdq. 79th Engr., Const. Bn., A. P. O. 74, San Francisco, Calif., and • Plymouth, Iowa.

FRIEDMAN, Arthur (*A* 1936) Pres., • Air Controls, Inc., 2310 Superior Ave., Cleveland 14, and 15700 S. Morland Blvd., Shaker Heights, Ohio.

FRIEDMAN, David H., Jr. (*M* 1936) Major, and • 531 Whitney Ave., New Haven, Conn.

FRIEDMAN, F. J.* (*M* 1921) • Mech. Engr., • McDougall & Friedman, 1235 McGill College Ave., Montreal 2, and 1260 Mackay St., Montreal, Que., Canada.

FRIEDMAN, Milton (*A* 1939; *J* 1935; *S* 1933) H. F. Klawuhn, 34-24-82nd St., Jackson Heights, and • 333 West End Ave., Apt. 14C, New York, N. Y.

FRIEND, Walter F. (*M* 1941) Mech. Engr., • Ebasco Services, Inc., 2 Rector St., New York 6, and 131 Riverside Dr., New York 24, N. Y.

FRISSE, John L. (*A* 1945) Owner, The Thermal Products Co., 4500 Euclid Ave., Cleveland 3, and • 17412 Shelburne Rd., Cleveland Heights 18, Ohio.

FRITSCHE, Carl (*M* 1943) • The Carl Fritsche Co., 424 Park Ave., East Orange, N. J.

FRITZBERG, Lawrence H. (*M* 1941) Engr., Minneapolis-Honeywell Regulator Co., 2753 Fourth Ave. S., Minneapolis, and • 4628 Columbus Ave. S., Minneapolis 7, Minn.

FROELICH, H. Allen (*A* 1939) Owner & Mgr., • Air Conditioning & Appliances, 110 W. Seventh St. and 114½ W. Eighth St., Junction City, Kans.

FROESE, Erhard A. (*M* 1944) Branch Mgr., York Corp., 804 Security Bldg., 650-17th St., and • 2581 Elm St., Denver, Colo.

FUITE, Richard J. (*A* 1945) Dist. Repr., • Hoffman Specialty Co., 116 New Montgomery St., San Francisco 5, and 2300 N. Point St., San Francisco 23, Calif.

FULLER, Charles A. (*M* 1940) • Myers, Fuller and Addington, 21 East 40th St., New York 16, and 501 E. Fifth St., Mt. Vernon, N. Y.

FULLER, Clayton W. (*A* 1945) Engr., Dixie Manufacturing Co., Inc., 1310 Russell St., Baltimore 30, and • 5900 Highgate Dr., Baltimore 15, Md.

FULLER, Elbridge W. (*M* 1938) Engr., and • 1524 Garcia Ave., Coral Gables, Fla.

FULLMAN, James B. (*A* 1946; *J* 1941) Lt., U. S. Naval Reserve, Bldg. 109, Brooklyn Navy Yard Annex, Bayonne, N. J., and • 25 East 10th St., New York 3, N. Y.

FULMOR, Ira P. (*M* 1942) Owner, • Air Conditioning Co. of S. Calif., 1003 Santa Fe Ave., Los Angeles 21, and 1230 Hillside Rd., Pasadena 2, Calif.

FURBER, Stanley L. (*A* 1941) Modutrol Div., Minneapolis-Honeywell Regulator Co., 3101 Gillham Plaza, and • 7429 Forest Ave., Kansas City 5, Mo.

FYFE, John H. (*A* 1944) Owner, Suburban Engineering Co., 211 Center Bldg., Lansdowne, and ● 810 Eaton Rd., Drexel Hill, Pa.

G

GABBERT, William L. (*M* 1945) Dist. Sales Mgr., American Air Filter Co., Inc., 215 Central Ave., Louisville 8, and ● 1138 Alfresco Pl., Louisville 5, Ky.

GABLE, H. R. (*A* 1943) Design Engr., ● Union Steel Products Co., 500 N. Berrien, and 917 N. Clinton, Albion, Mich.

GAGE, Burford B. (*M* 1944) Owner, ● Gage Plumbing & Heating Co., P. O. Box 1035, and 139 Overhill Rd., Salina, Kans.

GALABA, Alexander (*M* 1944) Chief Engr., Morrison Products, Inc., 16816 Waterloo Rd., Cleveland 10, and ● 1861 Beverly Hills Dr., Euclid, Ohio.

GALLMEIER, Sigmund H. (*M* 1943) Mech. Engr., Argonaut Realty Div., General Motors Corp., 307 G. M. Research Bldg., Second and Milwaukee, and ● 18495 Ilene, Detroit 21, Mich.

GALLOWAY, David (*M* 1941) Vice-Pres., ● Bowen Refrigeration Supplies, Inc., 323 Spring St. N.W., Atlanta, and 1018 E. Lake Dr., Decatur, Ga.

GALLOWAY, Max W. (*A* 1946; *J* 1944) Power House Foreman, Allison Div., General Motors Corp., and ● 1428 N. Pennsylvania St., Indianapolis 2, Ind.

GAMBLE, Burton L. (*A* 1946; *J* 1944) Design Engr., Ferro Enamel Corp., 4150 East 56th St., Cleveland 5, and ● 1857 East 85th St., Cleveland 6, Ohio.

GAMBLE, Cary B. (*M* 1940; *A* 1935) Cons. Engr., ● Canal Bldg., New Orleans 12, and 3432 Upperline St., New Orleans 15, La.

GAMMILL, O. E., Jr. (*M* 1940; *A* 1937; *J* 1930) Director of Sales, Carrier Corp., 300 S. Geddes St., Syracuse 1, and ● 403 Oak St., Syracuse, N. Y.

GANNON, Russell R. (*M* 1939) Owner, ● Russell R. Gannon Co., Gwynne Bldg., Cincinnati, and 6442 Grand Vista Ave., Pleasant Ridge, Cincinnati, Ohio.

GANT, H. P.* (*M* 1915), (*Presidential Member*), (Pres., 1923; 1st Vice-Pres., 1922; 2nd Vice-Pres., 1921; Council, 1918; 1921-24) ● R. D. 1, Glenmore, Pa.

GARBER, William E., Jr. (*A* 1944; *J* 1938) Sales Mgr., Farquar Heating Service Co., 3406 East 10th St., Indianapolis, and ● R. R. 1, Fairland, Ind.

GARDNER, C. Rollins (*A* 1937) Vice-Pres., ● Martyn Brothers, Inc., 1000 St. Louis St., Dallas 2, Texas.

GARDNER, Francis B. (*M* 1944) Htg., Vent. & Air Cond., ● 3534 Whittier Blvd., Los Angeles 23, and 1654 Riverside Dr., Glendale 1, Calif.

GARDNER, Marvin K. (*A* 1945; *J* 1944) ● Ilg Electric Ventilating Co., 1215 Texas Bank Bldg., Dallas 1, and 2725 W. Avon, Dallas 11, Texas.

GARNEAU, Leo (*M* 1938; *J* 1930) Sales Engr., ● C. A. Dunham Co., Ltd., Dominion Square Bldg., 1010 St. Catherine St. W., Montreal 2, and 34 Coolbreeze Ave., Lakeside, Que., Canada.

GARNETT, William (*M* 1945) Mech. Engr., War Ordnance, ● Frankford Arsenal, Philadelphia, and R. F. D. 3, Bethlehem, Pa.

GARRISON, C. Hill (*M* 1943) Owner, ● C. H. Garrison Co., 406 West 34th St., Kansas City 2, and R. F. D. 3, Lee's Summit, Mo.

GARVIN, Joshua L. (*A* 1945) Engr., Army Air Field, and ● 122 Woodland Shores Rd., Charleston, S. C.

GATES, A. S., Jr. (*M* 1943; *A* 1941) Engr. (Naval Archt.), Bureau of Ships, Code 638, Navy Department, Washington, D. C., and ● 111 County Rd., Kensington, Md.

GATES, James N. (*M* 1944) Engr., Air Conditioning Engineering Co., 90 Memorial Dr., Cambridge, and ● 179 Washington St., Reading, Mass.

GAULIN, Richard P. (*A* 1945) Mech. Engr., Federal Public Housing Authority, Longfellow Bldg., and ● 5414 Eighth St. N.W., Washington, D. C.

GAULT, George W. (*A* 1946; *J* 1937; *S* 1934) Norair Engineering Corp., 1114-22nd St. N.W., Washington, D. C., and ● 403 Luray Ave., Alexandria, Va.

GAUSE, H. Chester (*M* 1937) Indus. Power Engr., Alabama Power Co., 600 North 18th St., P. O. Box 2641, Birmingham 2, and ● 3916 Montevallo Rd., Birmingham 9, Ala.

GAUSEWITZ, William H. (*A* 1937) Pres., ● Vale Engineers, Inc., 2919 Fremont S., Minneapolis 8, and 1321 W. Minnehaha Pkway., Minneapolis 9, Minn.

GAUSMAN, Carl E. (*M* 1923) Partner, Gausman & Moore, 1026 First National Bank Bldg., and ● 2360 Chilcombe Ave., St. Paul, Minn.

GAY, Clarence E., Sr. (*M* 1944) Mgr., ● Heaven Engineering Co., 1330 Broadway, Kansas City 6, and 410 West 17th St., Kansas City 8, Mo.

GAY, Clarence E., Jr. (*J* 1944) Asst. Engr., E. K. Campbell Heating Co., 2445 Charlotte, and ● 311 East 48th St., Kansas City 2, Mo.

GAYLORD, F. H. (*M* 1921) Vice-Pres. & Gen. Sales Mgr., Hoffman Specialty Co., Indianapolis, Ind., and ● 362 N. York St., Elmhurst, Ill.

GAYMAN, Paul D. (*M* 1938) Branch Mgr., ● Johnson Service Co., 2142 East 19th St., Cleveland, and 20875 Endsley Ave., Rocky River, Ohio.

GAYNER, James (*M* 1937) Cons. Engr., ● 624 Hearst Bldg., San Francisco, and 327 Magnolia Ave., Piedmont, Calif.

GEBEL, Kurt M. (*J* 1944; *S* 1940) ● U. S. Army Air Forces, T/3, 31242168, Hq. Mis (Main) U. S. F. E. T., A. P. O. 757 c/o P. M., New York, and 229 Beach 120 St., Rockaway Park, N. Y.

GEHRS, William (*A* 1939) Branch Mgr., ● Johnson Service Co., 316 American Bank Bldg., Portland 5, and 3801 S.E. Woodward St., Portland 2, Ore.

GEIGER, Irvin H. (*M* 1919) Reg. Prof. Engr., ● 410 Telegraph Bldg., and 240 Maclay St., Harrisburg, Pa.

GEIGER, Raymond L. (*M* 1939) Mech. Engr., Austin Co., Noble Rd. and Euclid Ave., and ● 1536 Dille Rd., Euclid, Ohio.

GEIRINGER, Paul L. (*M* 1943) Licensed Prof. Engr., ● 11 Blake St., Newtonville, Mass.

GEISSENHAINER, Louis R. (*A* 1945) Owner, ● L. R. Geissenhainer, 89 Broad St., Boston, and 64 Walton Park, Melrose, Mass.

GELERT, Carl R. (*A* 1945) Partner, ● The C. R. Gelert Co., 35 N. Raymond St., Pasadena, and 490 Maylin St., Pasadena 2, Calif.

GELTZ, R. W. (*A* 1943; *J* 1936) Dist. Engr., York Corp., 2700 Washington Ave., Cleveland, and ● 14213 Glenside Rd., Cleveland 10, Ohio.

GENONE, Henry W. (*M* 1941) Vice-Pres. in charge of Engrg., ● Dinkler Hotels Co., Inc., Ansley Hotel, and 83-26th St. N.E., Atlanta, Ga.

GERDSEN, Arno H. (*A* 1945) Pres., ● The A. H. Gerdsen Co., 626 Broadway, Cincinnati 2, and Box 532, R. R. 14, Beechgrove Dr., Cincinnati 11, Ohio.

GERMAIN, Oscar (*M* 1935) Vice-Pres., Germain & Frere, Ltd., 237 St. Antoine St., and ● 1343 Boulevard St. Louis, Trois-Rivieres, Que., Canada.

GERRISH, Grenville B. (*M* 1944; *A* 1936; *J* 1930) Pres., Wolverine Equipment Co., 31 Maine St., Cambridge 42, and ● 26 Standish Rd., Melrose 76, Mass.

GERRISH, Harry E. (*M* 1910), (Council, 1919) Pres., ● Morgan-Gerrish Co., 307 Essex Bldg., Minneapolis 2, and 4534 S. Fremont Ave., Minneapolis 9, Minn.

GERSTENBERGER, Edgar J. (*A* 1938) Secy.-Treas., Glendale Supply Co., 1819 W. Glendale Ave., Milwaukee 9, and ● 3824 North 17th St., Milwaukee 6, Wis.

GESSELL, E. T. (*M* 1941) ● W. E. Lewis & Co., 610 Thomas Bldg., Dallas 1, and 2705 E. Amherst, Dallas 5, Texas.

GETSCHOW, Roy M. (*M* 1919) Pres., ● Phillips-Getschow Co., 32 W. Hubbard St., Chicago, and 122 Woodstock, Kenilworth, Ill.

GETTINGER, Edgar W. (*M* 1945) Prod. Design Engr., ● American Furnace Co., 2725 Delmar, St. Louis, and 6627 University Dr., University City 5, Mo.

GEZARI, Zvi (*A* 1941) Research Dir., Industrial Device Corp., 202 East 44th St., New York, and ● 10 West 65th St., New York 23, N. Y.

GHOSE, Khagendra N. (*A* 1938) Cons. Engr., 17 State St., New York, N. Y., and ● 39 Ramkanta Bose St., Bagh Bazar, Calcutta, India.

GHOSH, B. B. (*A* 1944; *J* 1939) Deputy Dir., Civil Engrg., Government of India, Esplanade East, and ● 165A Janak Rd., Calcutta, India.

GIANNINI, Mario C. (*M* 1935) Assoc. Prof. of Mech. Engrg., Dir., Evening Div., College of Engrg., ● New York University, University Heights 53, New York, and 322 Read Ave., Crestwood, N. Y.

GIBBONS, Michael J. (*M* 1914) Gen. Mgr., M. J. Gibbons Supply Co., 601-631 E. Monument Ave., Dayton, and ● 116 W. Thruston Blvd., Dayton 9, Ohio.

GIBBONS, Michael J., Jr. (*A* 1944) Htg. Engr., M. J. Gibbons Supply Co., 601 E. Monument Ave., Dayton 2, and ● 206 Canterbury Dr., Dayton 9, Ohio.

GIBBS, Edward W. (*Life Member*; *M* 1919) Pres., ● The Smith-Gibbs Co., 201 S. Main St., and 39 President Ave., Providence, R. I.

GIBSON, Harry E. (*A* 1945) Owner, ● Gibson Sheet Metal Works, 965 W. Western Ave., and R. R. 1, Muskegon, Mich.

GIESECKE, F. E. (*Life Member*; *M* 1913), (*Presidential Member*), (Pres., 1940; 1st Vice-Pres., 1939; 2nd Vice-Pres., 1938; Council, 1932-41) ● Box 417, New Braunfels, Texas.

GIFFORD, Edmund W. (*M* 1942) Dist. Mgr., ● Himelblau Byfield & Co., 611 N. Broadway, Milwaukee 2, and 4457 N. Stowell Ave., Shorewood 11, Wis.

GIFFORD, Robert L. (*Life Member*; *M* 1908) Pres., Illinois Engineering Co., 21st St. and Racine Ave., Chicago 8, and ● 1231 S. El Molino Ave., Pasadena 5, Calif.

GILBERT, I. Wolfe (*A* 1945) Co-Owner, ● Gilbert & Knight Co., 1300 Broad St., and 223 Pleasant St., Utica, N. Y.

GILBERT, Leslie S. (*M* 1937) Owner, ● Gilbert Engineering Co., 1305 Liberty Bank Bldg., Dallas 1, and 3713 Southwestern Blvd., Dallas 5, Texas.

GILBERT, Thomas (*A* 1940) Sales Engr., Empire Brass Manufacturing Co., Ltd., London and Hamilton, and ● 12 Ontario St. S., St. Catherines, Ont., Canada.

GILBOY, John P. (*M* 1941) Sr. Member, John P. Gilboy Co., and ● 521 Arthur Ave., Scranton, Pa.

GILBOY, William O. (*M* 1945) Engr., Cooney Co., 223 Walton St., and ● 954 Lancaster Ave., Syracuse, N. Y.

GILFRIN, George F. (*M* 1945) Owner, ● Edificio La Nacional 601, P. O. Box 63 Bis. and Explanada 715, Lomas de Chapultepec, Mexico, D. F.

GILLE, Hadar B. (*M* 1930) Cons. Engr., Hugo Theorells Ingeniorsbyra AB, Lagerlofsgatan 8, Stockholm, and ● Svanhildsvagen 19, Nockeby, Sweden.

GILLESPIE, Harvey M. (*A* 1945) Patent Lawyer, ● 105 W. Adams St., Chicago 3, and 1442 Irving Park Rd., Chicago, Ill.

GILLHAM, Walter E. (*M* 1945) Owner, Cons. Htg. & Vent. Engr., ● 401-402 Scarritt Arcade, Kansas City 6, and 6336 Morningside Dr., Kansas City 5, Mo.

GILLILAND, Lesley L. (*A* 1942) Air Cond. Engr., Air Engineers, Inc., 1529 Second Ave., and ● 4904 Seventh Ave., South Birmingham, Ala.

GILMAN, Franklin W. (*M* 1935) Plant Engr., ● Loft Candy Corp., 38-38 Ninth St., Long Island City, N. Y., and The Chatham, 20th and Walnut Sts., Philadelphia, Pa.

GILMORE, John L. (*M* 1944; *A* 1938) Owner, ● John L. Gilmore, 1525 Cochran Ave., 1604 Union St., Brunswick, Ga.

GILMORE, Louis A. (*A* 1940; *J* 1935; *S* 1930) Pres., ● Gilmore & Co., 115 South 11th St., St. Louis 2, and 6633 Kingsbury, St. Louis 5, Mo.

GINI, Aldo (*M* 1933) ● Via Correggio 18, Milano, Italy.

GINN, Tony M. (*M* 1935) Gen. Mgr., ● Tony M. Ginn Co., 214-24 Fifth St. S., Great Falls, Mont.

GINSBERG, Maxwell R. (*A* 1944) Owner, ● American Construction & Engineering Co., 160-16 Jamaica Ave., Jamaica 2, L. I., N. Y.

GINZBURG, Nicola (*M* 1942) Marine Vent. Design Engr., The Pusey & Jones Corp., Front & Popiar St., Wilmington, and ● 5 Maple Ave., Claymont, Del.

GIRGIS, Sobhi (*A* 1945) Engr., ● Carrier Egypt, S. A. E., 37 Kasr El-Nil St., and c/o Sami G. El-Masry, Rd. 15, Maadi, Cairo, Egypt.

GITTLESON, Harold (*A* 1936) Sales Mgr., LaRiviere, Inc., 3715 St. Lawrence Blvd., Montreal 18, and ● 1125 Lajoie Ave., Apt. 6, Outremont, Montreal 8, Que., Canada.

GIVELBER, Samuel H. (*M* 1945) Owner, Mech. Engr., ● Reliance Heating & Air Cond. Co., 7700 Stanton Ave., Cleveland, and 3411 Bradford, Cleveland Heights, Ohio.

GIVIN, Albert W. (*A* 1925) Vice-Pres., ● The Gurney Foundry Co., Ltd., 4 Junction Rd., Toronto, and R. R. 2, Freeman, Ont., Canada.

GLANCE, Alvin C. (*M* 1943) Field Engr., The Anthracite Industries, Inc., 101 Park Ave., New York 17, N. Y., and ● R. D. 2, Allentown, Pa.

GLANZER, Clarence J. (*M* 1944) Vice-Pres., ● Merit Machine Co., Merit Appliances Div., Box 84, Northfield, and Vesta Ave., Northfield Center, Ohio.

GLASS, Robert (*A* 1943) Secy.-Treas., ● Partridge-Holliday, Ltd., 144 Lombard St., Winnipeg, and 85 Claremont Ave., Norwood, Man., Canada.

GLASS, William (*M* 1934) Pres. & Mgr., ● Partridge-Halliday, Ltd., 144 Lombard St., and 605 River Ave., Suite "A," Winnipeg, Man., Canada.

GLOVER, Ralph P. (*A* 1945) Cons. Engr., ● 1024 Superior St., Oak Park, Ill.

GODBOLD, Bernard P. (*A* 1944) Prin. Optg. Engr., Smithsonian Institute, Washington, and ● 1751 Lang Place, N.E., Washington 2, D. C.

GODES, Elliot (*A* 1942; *J* 1939) Engr., ● Anemostat Corporation of America, 10 East 39th St., New York 16, and 925 Westend Ave., New York 25, N. Y.

GODFREY, J. E. (*A* 1946; *J* 1938) Ens., ● Navy Dept., Section Ad 4b, Washington, D. C.

GODFREY, Theodore R. (*A* 1945) Partner, ● F. H. Godfrey Engineering Co., 421 Vance Bldg., Seattle 1, and 5212-18th Ave. N.E., Seattle 5, Wash.

GOEHLER, Elmer E. (*A* 1939) ● 1921 S.E. Grand Ave., Portland, Ore.

GOELLNER, John A. (*A* 1945) Pres., ● Monarch Metal Weather Strip Corp., 6333 Etzel Ave., St. Louis 14, and 4729 Lee Ave., St. Louis 15, Mo.

GOELZ, Arnold H. (*M* 1931) Pres., ● Kroeschell Engineering Co., 215 W. Ontario St., Chicago, and 827 Greenwood Ave., Wilmette, Ill.

GOERG, B. (*M* 1928) ● American Radiator & Standard Sanitary Corp., 675 Bronx River Rd., Yonkers, N. Y.

GOERGENS, Albert G. (*A* 1938) Engr., U. S. War Dept., New War Dept., Washington, D. C., and ● 817 Chalfonte Dr., Alexandria, Va.

GOFF, John A.* (*M* 1939) Dean, ● Towne Scientific School, University of Pennsylvania, Philadelphia 4, and 623 Righters Mill Rd., Penn Valley, Pa.

GOINS, E. H. (*M* 1941) Dist. Repr., ● Warren Webster & Co., 420 Market St., San Francisco 11, and 2353 Laguna St., San Francisco 15, Calif.

GOLDBERG, Moses (*A* 1934) Pres., Electric Motors Corp., 168 Centre St., New York 13, and ● 885 E. Eighth St., Brooklyn 30, N. Y.

GOLDMANN, Philipp (*J* 1942; *S* 1940) ● 238 Fellows Ave., Syracuse 10, N. Y.

GOLDNER, Herman W. (*M* 1944) Pres., ● Herman Goldner Co., Inc., 425 W. Lehigh Ave., and 7415 Euston Rd., Philadelphia, Pa.

GOLDSMITH, F. W. (*M* 1936) Pres., ● The W. Clasmann Co., 513 E. Day Ave., and 629 E. Day Ave., Milwaukee 11, Wis.

GOLL, Willard A. (*A* 1937) Engr., W. A. Goll & Assoc., and ● 4423 Pine St., Omaha, Nebr.

GONZALEZ, Rafael A. (*M* 1936) Chrysler Corp., Airtemp Div., 119 Leo St., Dayton 1, and ● 276 Delaware Ave., Dayton, Ohio.

GOOD, Charles S. (*J* 1941; *S* 1939) Partner, A. C. Good & Sons, 620 Rebecca Ave., and ● 102 Cascade Rd., Pittsburgh 21, Pa.

GOODE, Archie E. (*A* 1945) Mech. Engr., & Draftsman, Robert & Co., Inc., Bona Allen Bldg., Atlanta, and ● 535 E. Lake Dr., Decatur, Ga.

GOODHUE, B. C. (*M* 1945) Application Engr., W. A. Case & Sons Manufacturing Co., 115 N. Geddes St., and ● 610 W. Onondaga St., Syracuse, N. Y.

GOODMAN, C. L. (*A* 1944) Mech. Engr., ● 5715 31st N.E., Seattle, Wash.

GOODMAN, William (*M* 1941) Prof. & Cons. Engr., • Illinois Institute of Technology, 3300 Federal St., Chicago 16, and 5141 Greenwood Ave., Chicago 15, Ill.

GOODRAM, W. E. (*M* 1939; *A* 1936) Partner & Mgr., Goodram Bros., 88 King St. W., Hamilton, and • R. R. 2, Freeman, Ont., Canada.

GOODRICH, Charles F. (*M* 1919) Andrews & Goodrich, Inc., Boston, and • 336 Adams St., Dorchester, Mass.

GOODWIN, Eugene W. (*M* 1936) Mech. Engr., Public Buildings Administration, and • 7024 Hampden Lane, Bethesda 14, Md.

GOODWIN, Frank T. (*A* 1944) Repr., Forest City Foundries Co., 2500 West 27th St., Cleveland 13, Ohio, and • 902 Ballard St. S.E., Grand Rapids 7, Mich.

GOODWIN, Samuel L. (*Life Member*; *M* 1924) Cons. Engr., John & Drew Eberson, 2 West 47th St., New York, N. Y., and • 247 Madison Ave., Hasbrouck Heights, N. J.

GOODWIN, Walter C. (*M* 1945) Negotiation Mgr., Air Cond. Div., • Westinghouse Electric Corp., 150 Pacific Ave., Jersey City, and 57 Elm St., Westfield, N. J.

GOOHS, William E. (*M* 1944) Asst. to Plant Engr., Allison Div., General Motors Corp., Indianapolis, and • 5910 Indianola Ave., Indianapolis 5, Ind.

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GORDON, Colin W. (*A* 1938) Vice-Pres. & Mgr., A. G. Baird, Ltd., 286 Lisgar St., and • 962 Shaw St., Toronto, Ont., Canada.

GORDON, Peter B. (*M* 1944; *A* 1938; *J* 1935) Treas., • Wolff & Munier, Inc., 222 East 41st St., New York 17, N. Y., and 35 Park Ave., Bloomfield, N. J.

GORGEN, Roy E. (*A* 1940) Vice-Pres., The Day Co., 810 Third Ave. N.E., and • 2120 W. Cedar Lake Blvd., Minneapolis, Minn.

GORNSTON, Michael H. (*Life Member*; *A* 1923) Retired, Board of Education, City of New York, and • 111-32-76th Ave., Forest Hills, N. Y.

GORSUCH, Winfield J. (*A* 1944) Assoc. Naval Archt., Norfolk Navy Yard, and • 314 Harrison St., Portsmouth, Va.

GOSS, Matthew H. (*M* 1938) Partner, • M. H. Goss Co., 3409 Ludden St., Detroit 7, and 296 E. Grand Blvd., Detroit, Mich.

GOSSETT, Earl J. (*M* 1923) Pres., • Bell & Gossett Co., 8200 Austin Ave., Morton Grove, and 314 Woodland Ave., Winnetka, Ill.

GOTHARD, William W. (*A* 1936) Gen. Mgr., • Domestic Engineering, 1900 Prairie Ave., Chicago 16, and 1027 Arlington Ave., La Grange, Ill.

GOTSCHALL, Harry C. (*M* 1935) Air Cond. Teacher, Lane Technical High School, 2501 W. Addison St., Chicago 18, and • 2953 Eastwood Ave., Chicago 25, Ill.

GOTTSCHALK, Kenneth A. (*A* 1944) Indus. Sales & Power Engr., Cleveland Electric Illuminating Co., 75 Public Sq., Cleveland 1, and • 3641 West 46th St., Cleveland 2, Ohio.

GOTTSCHALL, Walter C. (*A* 1945) Pres. & Estimator, Industrial Insulation, Inc., Terminal Office Bldg., Pittsburgh 19, and • 530 Hillcrest Pl., Pittsburgh 16, Pa.

GOTTWALD, C. (*A* 1916) Pres., • The Ric-wil Co., 1562 Union Commerce Bldg., Cleveland, and 2225 Stillman Rd., Cleveland Heights, Ohio.

GOULD, Henry E. (*M* 1942) Partner & Mgr., • Natkin & Co., 1800 Baltimore, Kansas City 8, Mo., and 2300 Wyncote Lane, Kansas City 3, Kans.

GOULD, James L. (*J* 1942) Pilot, Air Corps, Columbus, Miss., and • 1701 Poyntz Ave., Manhattan, Kans.

GOULDBOURNE, Thomas H. (*M* 1944) Sales Dir., The Standard & Pochin Bros., Ltd., Evington Valley Rd., and • 39 Stoughton Rd., Leicester, England.

GOULDING, William (*A* 1933) Air Cond. Engr., World Broadcasting System, Inc., 711 Fifth Ave., New York, and • 42 Highview Ave., Tuckahoe, N. Y.

GOUNDIE, Joseph K. (*M* 1938) Sales Engr., Fritch Coal Co., 116 River St., Bethlehem, and • 1426 Walnut St., Allentown, Pa.

GOWDY, Allen C. (*A* 1941; *J* 1939) Mgr., • The Huffman-Wolfe Co., 308 Standard Bldg., Atlanta 3, and 417 Hillside Dr. S.W., Atlanta, Ga.

GRABER, Ernst (*A* 1942; *J* 1936) Engr., Minneapolis-Honeywell Regulator Co., 221 Fourth Ave., New York, and • 215 Hollywood Ave., Douglaston, L. I., N. Y.

GRABMAN, Henry B. (*J* 1942; *S* 1938) 1st Lt., 0-393335, • H & S Co., 95th Engr., Regt., A. P. O. 350, c/o Postmaster, New York, N. Y., and 355 E. Spring St., Zelienople, Pa.

GRAHAM, F. D. (*M* 1940) Local Mgr., • York Corp., 422 Richards Bldg., New Orleans, and 420 Betz Pl., New Orleans 20, La.

GRAHAM, William A. (*A* 1945) Prod. Engr., Reid Hayden, Inc., 1238 W. Morehead, and • 1638 Garden Terrace, Charlotte, N. C.

GRAHAM, William D. (*M* 1929; *A* 1925; *J* 1923) Dist. Mgr., • The Trane Co., Piedmont Bldg., Rm. 407, Greensboro, N. C.

GRANDSTAFF, C. L. (*M* 1944) Engr., The C. A. Olsen Manufacturing Co., and • 192 Longfellow Ave., Elyria, Ohio.

GRANKE, Arnold A. (*M* 1939) Sales Engr., Speakman Co., 816 Tatnall St., Wilmington 99, and • 28 West 41st St., Wilmington 220, Del.

GRANSTON, Ray O. (*A* 1939; *J* 1935; *S* 1930) Partner, • University Plumbing & Heating Co., 3941 University Way, Seattle, and 1533 East 76th St., Seattle 5, Wash.

GRANT, Ralph A. (*M* 1945) Engr., • McCrea Equipment Co., 516 Second St. N.W., Washington, and 5050 First St. N.W., Washington 1, D. C.

GRANT, Walter A.* (*M* 1943; *A* 1933; *J* 1929) Dir. of Appl. Engrg., Carrier Corp., Syracuse, and • Woods End, R. D. 1, Camillus, N. Y.

GRANT, Walter H., Jr. (*M* 1940) Mfrs. Repr., • Warren Webster & Co., 209 Vincent Bldg., New Orleans 12, and 5632 Elysian Fields Ave., New Orleans 19, La.

GRATIOT, Jules (*A* 1944) Vice-Pres. in charge of Sales, Air Maze Corp., 5200 Harvard Blvd., Cleveland, and • 3312 Braemar Rd., Shaker Heights 20, Ohio.

GRAVES, Charles S. (*A* 1945) Asst. Steam Ht. & Water Service Engr., • A. T. & S. F. Railway Co., General Office Bldg., Rm. 409, and 1112 Jewell St., Topeka, Kans.

GRAVES, Clarence C. (*M* 1943) Partner, • Graves & Graves, 3047 Sheffield Ave., Chicago 14, and 4041 N. Spaulding Ave., Chicago 18, Ill.

GRAY, Earle W. (*M* 1938; *A* 1934) Mgr., Indus. & Comm'l. Sales, • Oklahoma Gas & Electric Co., Third and Harvey Sts., and 2125 Northwest 18th St., Oklahoma City, Okla.

GRAY, Emerson G. (*J* 1944) Htg. Engr., Gray Engineering Co., P. O. Box 264, and • 1612 Johnson St., High Point, N. C.

GRAY, Everett W. (*M* 1936) Dist. Mgr., • The Trane Co., 1900 Euclid Ave., Cleveland 15, and 17545 Madison Ave., Lakewood 7, Ohio.

GRAY, Frank L., Jr. (*J* 1945) Engr., • Carrier Corp., 20 N. Wacker Dr., Chicago 6, and 5528 Maryland Ave., Chicago 37, Ill.

GRAY, G. A. (*M* 1924) Branch Mgr., • C. A. Dunham Co., Ltd., 45 Rideau St., and 114 Belmont Ave., Ottawa, Canada.

GRAY, Hamilton E. (*A* 1943; *J* 1940) Partner, Gray Engineering Co., Box 264, and • 108 Hillcrest Dr., High Point, N. C.

GRAY, William E. (*M* 1922) Partner, • Gray Engrg. Co., Box 264, and 718 W. Fariss Ave., High Point, N. C.

GREAR, Walter W. (*A* 1943) Engr., Freyn Bros., Inc., 1028 N. Illinois St., and • R. R. 16, Box 631-5, Indianapolis, Ind.

GREEN, Donald P. (*J* 1945) Supt., John E. Green Plumbing & Heating Co., Inc., and • 10415 Lincoln Dr., Huntington Woods, Mich.

GREEN, Edward J. (*A* 1945) Secy.-Treas., John E. Green Plumbing & Heating Co., Inc., Highland Park, and • 181 Tennyson Ave., Highland Park 3, Mich.

GREEN, Everett W. (*A* 1946; *J* 1938) Jr. Partner, Green Furnace & Plumbing Co., and • 2747 North 48th, Lincoln 4, Nebr.

GREEN, William C. (*Life Member*; *M* 1906) Retired, 707 Race St., Rm. 317, Cincinnati 2, and • 244 Erkenbrecher Ave., Cincinnati 29, Ohio.

GREENBURG, Leonard, M. D.* (*M* 1932) Exec. Dir., Div. of Industrial Hygiene, • New York State Department of Labor, 80 Centre St., New York 13, and 44 West 77th St., New York, N. Y.

GREENLAND, Sidney F. (*M* 1934) Tech. Htg. & Vent. Consultant, J. Brockhouse & Co., Ltd., West Bromwich, Staffs., and • 75 Grestone Ave., Handsworth Wood, Birmingham 20, England.

GREENWOOD, L. D. (*A* 1945; *J* 1943) • 3104 Main St., Rm. 201, Houston 4, and 3650 Locke Lane, Houston 6, Texas.

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GREGORY, Henry D. (*M* 1945) Htg. Dept., Hendrie & Bolthoff Manufacturing & Supply Co., 1635–17th St., P. O. Box 5110, Denver 17, and • 1042 S. High St., Denver 9, Colo.

GREGORY, Paul E. (*A* 1944) Sales Engr., Horne-Wilson, Inc., 163 Peters St. S.W., and • 2054 Chelsea Circle N.E., Atlanta, Ga.

GREILING, Winford W. (*M* 1942) Mech. Engr., The Austin Co., 16112 Euclid Ave., Cleveland, and • 1557 East 256th St., Cleveland 17, Ohio.

GREILSAMER, Philippe (*J* 1945) Engr., Air Cond. & Refrig., 122 East 42nd St., New York 22, and • Hotel Pierre, 2 East 61st St., New York 21, N. Y.

GREINER, Maurice C. (*M* 1945) Owner, Hess-Greiner & Polland, 1700 S. Main St., Los Angeles 15, and • 1559 Club View Dr., Los Angeles 24, Calif.

GREISS, Philip G. (*M* 1937) Mech. Engr., Voorhees, Walker, Foley & Smith, 101 Park Ave., New York, N. Y., and • 189 Walnut Ave., Bogota, N. J.

GRIEST, Kermit C. (*A* 1940; *J* 1936) C.W/O, Commander Service Force, • Pacific Fleet, F.P.O., San Francisco, Calif., and 38 Greenlee Rd., Pittsburgh 10, Pa.

GRIEWISCH, Alfred H. (*A* 1938) Pres., • Bayley Heating Supply Co., 2045 W. St. Paul Ave., Milwaukee 3, and 2557 North 47th St., Milwaukee 10, Wis.

GRIFFIN, Charles J. (*M* 1940) Lt. (j.g.), and • 8231-S. Loomis Blvd., Chicago, Ill.

GRIFFIN, DeWitt C. (*M* 1943) Owner, • DeWitt C. Griffin & Assocs., 717 Lloyd Bldg., Seattle 1, and 6521–32nd N.W., Seattle 7, Wash.

GRIFFITH, Claude A. (*A* 1938) Htg. & Vent. Engr., Griffith Air Conditioning Service, and • 16 Altamont Terrace, Cumberland, Md.

GRIFFITH, H. T. (*M* 1938) Lt., U. S. N. R., • 105th Naval Construction Bn., Fleet Post Office, San Francisco, Calif., and 1909 Third Ave. W., Seattle, Wash.

GRIFFITH, Joseph B. (*A* 1942; *J* 1938) Works Mgr., Bryant Heater Co., Tyler, Texas Plant, and • 902 Whitaker, Tyler, Texas.

GRIGG, James L. (*A* 1945) Asst. Mgr., Sheet Metal Dept., American Sheet Metal Co., 2713 Colley Ave., Norfolk, and • 531 Fisherman Rd., Norfolk 3, Va.

GRIMES, Fenner M. (*A* 1942; *J* 1935) Mech. Engr., War Dept., Office of the Chief of Engrs., Engrg. Div., Specifications Sect., Washington 25, D. C., and • 849 S. Ivy St., Arlington, Va.

GRISWELL, H. D. (*A* 1944) Vice-Pres., • The Hosmer Products Corp., 819 Massachusetts Ave., Indianapolis 4, and 1131 N. Delaware St., Indianapolis, Ind.

GRISWOLD, Arthur S. (*M* 1945) Staff Engr., Central Htg. Dept., • The Detroit Edison Co., 2000 Second Ave., Detroit, and 5015 Pon Valley, Bloomfield Hills, Mich.

GRISWOLD, Edgar A. (*A* 1945) Cons. & Design Engr., and • 6641 Pollard St., Los Angeles 42, Calif.

GRITSCHKE, Elmer R. (*M* 1940) Cons. Engr., • E. R. Gritschke, 123 W. Madison St., Chicago 2, and 4132 Gregory Ave., Wilmette, Ill.

GRITTON, Earl V. (*M* 1944) Htg. & Vent. Engr., Williams & Richardson Co., 204 Dooly Bldg., Salt Lake City 1, and • 2470 South 15th East St., Salt Lake City, Utah.

GROOM, John W., Jr. (*A* 1943) Design Engr., and • 1 Riverside Ave., Merrick, L. I., N. Y.

GROOT, Harry W. (*M* 1937) Mech. Engr., Supvsr. of Shipbuilding, Evansville, Ind., and • 3728 N. Western Pkwy., Louisville, Ky.

GROSS, Herbert A. (*M* 1945) Dist. Mgr., • Surface Combustion, 535 S. Seventh St., and 3705 Dupont Ave. S., Minneapolis, Minn.

GROSS, Lester (*A* 1946; *J* 1942) Chief Engr., • General Installation Co., 2234 Olive St., and 1368 Shawmut Pl., St. Louis, Mo.

GROSS, Lyman C. (*M* 1931) Mech. Engr., Magney, Tusler & Setter, 202 Foshay Tower, Minneapolis, and • 5324 Oaklawn Ave., Minneapolis 10, Minn.

GROSS, Morris H. (*M* 1944) Co-Owner, Heating, Ventilating & Air Conditioning, 8725 Puritan, Detroit 21, and • 18434 Marlowe, Detroit 19, Mich.

GROSS, Victor L. (*A* 1944) Sales Engr., • Taylor Forge & Pipe Works, 50 Church St., New York 7, and 281 Storer Ave., New Rochelle, N. Y.

GROSSENBACHER, Henry E. (*A* 1938) Pres., • Grossenbacher Furnace Co., 9416 W. Milton Ave., and 9741 Lackland Rd., Overland, Mo.

GROSSMAN, F. Arthur (*A* 1942; *J* 1938; *S* 1937) Sales Engr., • Servel, Inc., 122 S. Michigan Ave., Chicago 3, Ill., and 867 S. Harlan Ave., Evansville, Ind.

GROSSMAN, Harry E. (*A* 1943) Sales Engr., Minneapolis-Honeywell Regulator Co., 5060 Wayne Ave., Philadelphia 44, and • 213 Parham Rd., Springfield, Pa.

GROSSMAN, Ralph (*A* 1943) Combus. Sales Engr., Volcano, Ltd., 1106 Beaver Hall Hill, and • 4130 Decarie Blvd., Apt. 10, Montreal 28, Que., Canada.

GROSSMANN, Harry A. (*M* 1931) Owner, H. A. Grossmann Co., 3138 Cass Ave., St. Louis 6, and • 16 Huntleigh Downs, Route 5, Kirkwood 22, Mo.

GROSVOLD, F. E. (*M* 1944) Owner, • F. E. Grosvold Heating & Plumbing, 417 Wisconsin St., and 606 Dodge St., Eau Claire, Wis.

GROTH, Don R. (*A* 1945) Dist. Mgr., • U. S. Air Conditioning Corp., 342 Key Bldg., Oklahoma City 2, and 227 Northwest 23rd St., Oklahoma City 3, Okla.

GROVES, Samuel A. (*A* 1940; *J* 1935) Asst. to Pres., United Carr Fastener Co., 31 Ames St., Cambridge, and • 15 Sylvan Rd., Wellesley Hills 82, Mass.

GRUTZMACHER, Robert L. (*M* 1945) Vice-Pres., • Natural Gas Equipment, Inc., 540 Petroleum Bldg., Los Angeles 15, and 2121 N. Foothill Blvd., Altadena, Calif.

GSCHWIND, Jean F. (*M* 1945) Dvlp. Engr., J. O. Ross Engineering Corp., 350 Madison Ave., New York, and • 720 Grand St., Mamaroneck, N. Y.

GUEST, P. L., Jr. (*A* 1939) Owner, • P. L. Guest Sales Co., 311 Piedmont Bldg., and 716 Dover Rd., Greensboro, N. C.

GUEST, Ross Burton (*M* 1945; *A* 1940) Owner & Mgr., • The Guest Sheet Metal Works, 827-37 Dryades St., New Orleans 13, and 1028 City Park Ave., New Orleans, La.

GUILBERT, Stanley R. (*A* 1940) Air Cond. Engr., The Riester & Thesmacher Co., 1526 West 25th St., Cleveland 13, and • R. F. D. 3, Chagrin Falls, Ohio.

GULER, George D. (*A* 1937) Sales Mgr., Air Cond. Controls Div., • Minneapolis-Honeywell Regulator Co., 2753 Fourth Ave. S., and 2622 West 49th St., Minneapolis, Minn.

GUMAER, P. Wilcox (*M* 1937) Engr., The Barrett Div., Allied Chemical & Dye Corp., 40 Rector St., New York, N. Y., and • 25 Garden St., West Englewood, N. J.

GUNZEL, Rudolph M. (*M* 1942) Sales Repr., • R. M. Gunzel & Co., 320 Crocker St., Los Angeles 13, and 375 La Mirada Ave., San Marino 9, Calif.

GURNEY, E. Holt (*M* 1929). (*Presidential Member*), (Pres., 1938; 1st Vice-Pres., 1937; 2nd Vice-Pres., 1936; Council, 1931-39) Pres., • The Gurney Foundry Co., Ltd., 4 Junction Rd., Toronto 9, and 347 Walmer Rd., Toronto, Ont., Canada.

GURNEY, Edward R. (*A* 1940; *J* 1937) Mech. Supt., • Electric Steels, Ltd., Cap de La Madeleine, Que., Canada.

GUSTAFSON, Carl A. (*M* 1938) Sales Engr., • The Powers Regulator Co., 2720 Greenview Ave., and 6231 Fairfield Ave., Chicago, Ill.

GUTKNECHT, Fritz (*M* 1940) Chief Engr., • Blattmann-Weeser Sheet Metal Works, Inc., 1001 Toulouse St., New Orleans, La., and R. F. D. 2, Gulfport, Miss.

H

HAAS, Emil, Jr. (*M* 1945; *A* 1944) Partner, • Natkin & Co., 1800 Baltimore Ave., Kansas City 8, and 5526 Crestwood Dr., Kansas City, Mo.

HAAS, Samuel L. (*M* 1923) Pres.-Treas., • Advance Heating & Air Conditioning Corp., 117-119 N. Desplaines St., Chicago 6, and 4300 Lake Shore Dr., Chicago 13, Ill.

HABASHI, Fawzi (*S* 1945) Student, Architectural Dept., • Fouad 1st University, Faculty of Engineering, and 3 Khouzam St., Shoubre, Cairo, Egypt.

HACH, Edward C.* (*M* 1939) Dist. Repr., Air Conditioning Div., Westinghouse Electric Corp., 1503 Gulf Bldg., Pittsburgh 19, and • 221 Broadmoor Ave., Mt. Lebanon, Pittsburgh 16, Pa.

HACKETT, Frank C. (*A* 1940) Southern Dist. Mgr., • Bell & Gossett Co., 607 Barr Bldg., Washington D. C., and 3017 N. Glebe Rd., Arlington, Va.

HADEN, G. Nelson (*M* 1934; *A* 1928; *J* 1922) Chairman & Managing Dir., • G. N. Haden & Sons, Ltd., 19-29 Woburn Pl., London, W. C. 1, and 36 Wildwood Rd., London, N. W. 11, England.

HADEN, William N. (*Life Member*; *M* 1902) Retired Chairman, and • Arnolds Hill, Trowbridge, Wiltshire, England.

HADJISKY, Joseph N. (*M* 1930) Cons. Engr., • 744 Bates St., Birmingham, Mich.

HADLEY, Lawrence G. (*J* 1945) Apprentice Engr., Young, Austen & Young, Ltd., 19 Buckingham St., London, W. C. 1, and • 1 Salcombe Wave, Ruislip Manor, Middlesex, England.

HAGAN, William V. (*M* 1938; *A* 1933; *J* 1926) Owner, V. J. Hagan Co., 506 Pearl St., and • 7 Black stone Ave., Sioux City, Iowa.

HAGEDON, Charles H. (*M* 1919) Partner, • S. E. Fenstermaker & Co., 937 Architects & Builders Bldg., Indianapolis 4, and 945 West 58th St., Indianapolis 5, Ind.

HAGEN, George M. (*J* 1944) Vice-Pres., • Hagen & Co., Halifax, Ltd., 89 Hollis St., and 18 Coburg Rd., Halifax, Nova Scotia, Canada.

HAGEN, Roscoe T. (*A* 1945) Maintenance Engr., • The Perfect Circle Co., 552 S. Washington St., and 53 S. Perry St., Hagerstown, Ind.

HAGUE, William S. (*A* 1944) Mgr., • Crane Co., P. O. Box 876, and R. 1, Box 495, Indianapolis, Ind.

HAHN, Roy F. (*A* 1941; *J* 1936) Engr., Advanced Refrigeration, Inc., and • 936 Argonne Ave. N.E., Atlanta, Ga.

HAINES, J. E. (*M* 1940) Mgr., Commercial Control Sales, • Minneapolis-Honeywell Regulator Co., and 2119 S. Humboldt Ave., Minneapolis, Minn.

HAINES, John J. (*M* 1915) Pres., • The Haines Co., 1931 W. Lake St., Chicago 12, and 623–17th Ave., Maywood, Ill.

HAITMANEK, Louis M. (*A* 1938) Htg. & Vtg. Engr., and • 217 Rose St., Newark 3, N. J.

HAJEK, William J. (*M* 1945) • P. O. Box 405, Knoxville, Tenn.

HAKES, Leon M. (*M* 1932; *J* 1929) Resident Repr., • Warren Webster & Co., 210 Reynolds Arcade Bldg., Rochester 4, and 144 Inglewood Dr., Rochester 11, N. Y.

HALE, Frederick J. (*M* 1936) Mgr., • Empire Sheet Metal Works, Ltd., 1606 W. First Ave., Vancouver, and 995 Mathers Ave. W., Vancouver, B. C., Canada.

HALEY, Harry S.* (*M* 1914) Cons. Engr. and Partner, • Leland & Haley, 58 Sutter St., San Francisco 4, and 735–21st Ave., San Francisco, Calif.

HALEY, Robert T. (*A* 1943) Mgr., Heating Dept., • Minneapolis Gas Light Co., 739 Marquette Ave., and 5340 Beard Ave. S., Minneaplis, Minn.

HALL, Elmo (*M* 1944) Owner, • Empire Gas & Equipment Co., 1198 Stout St., and 1449 Ash St., Denver, Colo.

HALL, George (*A* 1937) Lt. (s.g.), U. S. N.–R., • Hyland, Hall & Co., 218 N. Bassett St., Madison 3, Wis.

HALL, John A. (*M* 1943) Sales Engr., • A. L. Vanderhoof, Inc., 233 Hanna Bldg., Cleveland 15, and 11030 Granger Rd., Garfield Heights, Ohio.

HALL, John R. (*M* 1937; *J* 1932) Vice-Pres., • c/o Visioneering Co., Inc., 4700 Prospect Ave., Cleveland 3, and 17700 Fries Ave., Lakewood 7, Ohio.

HALL, Mora S. (*M* 1934) Engr., • William Bornstein & Son, 2209 Channing Pl. N.E., Washington, D. C., and 5004–42nd Ave., Myattsville, Md.

HALL, Norman H. (*A* 1945) Supvsr., • East Ohio Gas Co., 1405 E. Sixth St., Cleveland 14, and 147 Beachview Rd., Willoughby, Ohio.

HALLENBECK, C. W. (*M* 1944) Dist. Engr., • T. C. Heyward, 1408 Independence Bldg., Charlotte, and 2157 Norton Rd., Charlotte 4, N. C.

HALT, Howard R. (*M* 1943) Mgr., Htg. Div., Thos. J. Sheehan Co., 2233 Olive St., St. Louis 3, and 8729 Annetta Ave., St. Louis 15, Mo.

HAMACHER, K. F. (*M* 1938) Partner, • Hamacher & Williams, 2540 W. Wells St., Milwaukee 3, and 4387 S. Austin St., Milwaukee 7, Wis.

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HAMIG, Louis L. (*M* 1941; *A* 1940; *J* 1935) Partner, • Ferris & Hamig, 516 Fullerton Bldg., St. Louis, and 7530 Buckingham Dr., Clayton 5, Mo.

HAMILTON, Howard S. (*A* 1940) Owner, • Air Comfort Co., 1330 N. Franklin Pl., Milwaukee 2, Wis.

HAMILTON, M. S. (*M* 1942) Southeastern Sales Mgr., Air Cond. Controls Div., Minneapolis-Honeywell Regulator Co., 4-5 N. Rhodes Center, and • 3372 Walters Ct., N. E., Atlanta, Ga.

HAMLET, F. Aylmer (*M* 1944; *A* 1936) Branch Office Mgr., • C. A. Dunham Co., Ltd., Dominion Sq. Bldg., Rm. 832, 1010 St. Catherine St. W., Montreal 2, and Grove Hotel, Beaconsfield, Pointe Claire, P. Q., Canada.

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HANBURGER, Fred W. (*M* 1938) Cons. Engr., • 252 West 76th St., New York, N. Y.

HAND, William L. (*M* 1944) Pres., Cons. Mech. Engr., • W. L. Hand Engineering Service, 4102 S. LaSalle St., and 8832 Dante Ave., Chicago, Ill.

HANLEIN, Joseph H. (*M* 1937) Vice-Pres., • Wilberding Co., Inc., 1822 Eye St. N.W., Washington 6, and 5420 Connecticut Ave. N.W., Washington, D. C.

HANLEY, Edward V. (*A* 1933) Pres., • S. V. Hanley Co., 1653 N. Farwell Ave., Milwaukee 2, and 844 E. Birch Ave., Whitefish Bay 11, Wis.

HANLEY, T. F., Jr. (*M* 1933) Pres., Hanley & Co., 1503 S. Michigan Ave., and • 1640 East 50th St., Chicago, Ill.

HANNAN, Lloyd F. (*A* 1945) Owner, • Ace Sheet Metal Works, 444 Clementina St., San Francisco 3, and 1343–32nd Ave., San Francisco 22, Calif.

HANNIGAN, William (*M* 1940) Bldg. Supt., • Acacia Mutual Life Insurance Co., 51 Louisiana Ave. N.W., Washington, D. C., and Route 2, Silver Spring, Md.

HANSEN, John S. (*A* 1945) Testing Engr., • Sears Roebuck & Co., Dept. 817, 925 S. Homan Ave., Chicago 7, and 115 South Blvd., Oak Park, Ill.

HANSEN, Willis A. (*M* 1943) Design Engr., G. M. Richards & Assoc., 333 Kearny St., San Francisco, and • 1440 California St., Berkeley 3, Calif.

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HANSON, Lars C. (*M* 1945) Asst. Dir. of Dvlpt., Carrier Corp., S. Geddes St., Syracuse 1, and • 282 Robineau Rd., Syracuse 4, N. Y.

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HANTHORN, Walter (*A* 1942; *J* 1939) Engr., Kleenair Furnace Co., 5329 N.E. Sandy Blvd., and • 2946 Northeast 54th Ave., Portland 13, Ore.

HAPPERFIELD, G. J. (*M* 1943) Managing Dir., • Chandos Engineering Co., Ltd., 10 Chandos St., Cavendish Sq., London, and 468 Staines Rd., Twickenham, Middlesex, England.

HARBERGER, G. L. (*A* 1939) Mgr., Boiler & Htg. Dept., Eastern Foundry Co., Boyertown, and • 641 King St., Pottstown, Pa.

HARBIN, Frank, Jr. (*M* 1941) Engr., • Home Furnace Co., 280 E. Sixth St., and 181 West 21st St., Holland, Mich.

HARBORDT, Otto E. (*A* 1936) Treas., • U. S. Supply Co., 1315 West 12th St., and 4600 Mill Creek Pkwy., Kansas City, Mo.

HARDEN, J. Clinton (*M* 1938) Engr., Round Oak Co., and • 106 Courtland St., Dowagiac, Mich.

HARDER, Ervin P. (*A* 1944) Product Application Engr., Herman Nelson Corp., St. Louis, and • 1502 Collins Ave., Richmond Heights 17, Mo.

HARDIN, James T. (*M* 1943) Mgr. of Heating Dept., • Roland M. Cotton Co., Inc., 1720 E. Tenth St., Indianapolis 1, and 1204 West 29th St., Indianapolis, Ind.

HARDING, Edward R. (*M* 1936) State Sales Engr. & Mgr., • Kewanee Boiler Corp., P. O. Box 536, 1003 Jefferson Std. Bldg., Greensboro, and Guilford College, N. C.

HARDING, Louis A.* (*Life Member*; *M* 1911), (*Presidential Member*), (Pres., 1930; 1st Vice-Pres., 1929; 2nd Vice-Pres., 1928; Council, 1922-31) Cons. Engr., and • 85 Cleveland Ave., Buffalo 9, N. Y.

HARDING, Walter (*M* 1941) Supt. Engr., Air Ministry, N.E. Wing, Bush House, Kingsway, and • Danesfield House, 33 Bute Gardens, Wallington, Surrey, England.

HARE, W. Almon (*M* 1941) Prop., • Hare Engineering Co., 155 W. Congress St., Detroit 26, Mich., and 833 Kildare Rd., Windsor, Ont., Canada.

HARMAN, Ralph H. (*A* 1945) Archt., • 2 W. Columbia St., and 330 Broadmoor Blvd. N., Springfield, Ohio.

HARMONAY, W. L. (*M* 1944; *A* 1935) Pres., • Michael Harmonay Corp., 124 Elm St., Yonkers 2, and 1360 Midland Ave., Bronxville, N. Y.

HARRIGAN, Edward M. (*M* 1915) Pres. & Gen. Mgr., • Harrigan & Reid Co., 1365 Bagley Ave., and 7450 LaSalle Blvd., Detroit, Mich.

HARRIGAN, Edward R. (*M* 1939) Lt., U. S. Navy, Navy Yard, New York, N. Y., and • 19 Erwin Pl., Caldwell, N. J.

HARRIGAN, John E. (*A* 1945) Plant Facilities Engr., U. S. Time Corp., 31 Cherry Ave., Waterbury, and • 1081 Huntingdon Ave., Waterbury 46, Conn.

HARRINGTON, David W.* (*A* 1943) Sales Engr., • Dallas Air Conditioning Co., 2809 Canton St., and 1549 El Campo, Dallas, Texas.

HARRINGTON, Elliott D.* (*M* 1932; *A* 1930) Mgr., Induction Motor Div., • General Electric Co., 1 River Rd., Schenectady, and The Hawthorne, R. D. 2, Altamont, N. Y.

HARRINGTON, Jeremiah P. (*A* 1944) Mech. Engr., U. S. Engineers, and • 1515 Mirasol, San Marino, Calif.

HARRINGTON, Larry J. (*M* 1941) Owner, • Larry Harrington Co., 1971 N.W. Lovejoy St., Portland 9, Ore.

HARRIS, Herbert J. (*A* 1945) Pres., • Marion Oil & Service Corp., 222 N. Fifth Ave., Mt. Vernon, and 7 Croydon Rd., Yonkers, N. Y.

HARRIS, Jesse B. (*M* 1918) Mgr., • Jesse B. Harris Co., 702 Wesley Temple Bldg., Minneapolis 2, and 3620 Colfax Ave. S., Minneapolis 8, Minn.

HARRIS, Warren S.* (*M* 1942) Asst. Prof. of Special Research, • University of Illinois, 801 W. Green St., and 802 N. Broadway, Urbana, Ill.

HARRISON, Paul N. (*A* 1945) Owner, • Paul N. Harrison Co., 50 Hawthorne, San Francisco, and 2301 Ivy Dr., Oakland, Calif.

HARRISON, William Z. (*A* 1944) Owner, • Harrison & Co., 210 S.W. Temple St., Salt Lake City 1, and 1127 E. South Temple St., Salt Lake City 2, Utah.

HART, Harry M.* (*Life Member*; *M* 1912), (*Presidential Member*), (Pres., 1916; 1st Vice-Pres., 1915; Council, 1914-17) Pres., • L. H. Prentice Co., 1048 Van Buren St., Chicago 7, and 3730 Lakeshore Dr., Chicago, Ill.

HART, John H. (*M* 1942) Lt. Comdr., U. S. N. R., Navy Dept., Bureau of Ships, Washington, D. C., and • 1506 Mount Eagle Pl., Alexandria, Va.

HART, Stanley (*M* 1938) Pres., • Tuttle & Bailey, Inc., and 60 Chatham Rd., New Britain, Conn.

HART, Theodore S. (*M* 1938) Partner, Brenton & Hart, 6 Beacon St., Boston, and • 530 Lincoln St., New Britain, Conn.

HART, Winston W. (*A* 1945) Jr. Engr., Carrier Corp., and • 768 Ostrom Ave., Syracuse, N. Y.

HARTIN, William R., Jr. (*A* 1944; *J* 1935) Vice-Pres.-Secy., 2123 Green St., Columbia 41, and • 2744 Thenholm Rd., Columbia 32, S. C.

HARTMAN, John M. (*M* 1927) Engr., • Kewanee Boiler Corp., and 618 Elliott St., Kewanee, Ill.

HARTMANN, Charles W. (*A* 1944) Sales Engr., A. F. Hinrichsen, Inc., 50 Church St., New York 7, and • 551–72nd St., Brooklyn, N. Y.

HARTON, A. J. (*A* 1935) Sales Engrg., St. Joseph Light & Power Co., 514 Francis, and • 730 E. Hyde Park Ave., St. Joseph, Mo.

HARTSFIELD, Charles G. (*M* 1944) Vice-Pres. and Gen. Mgr., • Atlantic Engineering Co., 730 S. Elm St., Greensboro, N. C.

HARTSOOK, Granville S., Jr. (*A* 1939) Owner, • Hartsook Plumbing & Heating Co., 21 S. Royal Ave., P. O. Box 361, and Front Royal, Va.

HARTWEIN, C. E. (*M* 1933) • St. Louis County Gas Co., 231 W. Lockwood Ave., Webster Groves 19, and 135 Peeke Ave., Kirkwood 22, Mo.

HARTWELL, Joseph C. (*Life Member*; *M* 1922) Pres., • Hartwell Co., Inc., 87 Weybosset St., and 16 Freeman Pkwy., Providence, R. I.

HARTWIG, Walter J. (*A* 1945) Mech. Engr., Service Command Engineer, Federal Bldg., Omaha 1, and • 4516 William St., Omaha 6, Nebr.

HARVEY, A. D. (*A* 1928; *J* 1925) Vice-Pres., • The Dorr Co., Inc., 570 Lexington Ave., New York 22, N. Y.

HARVEY, Ernest W. (*A* 1943) Application Engr., • Westinghouse Electric Corp., 1014 Fairfax Bldg., and 210 East 72nd St., Kansas City, Mo.

HARVEY, John W. (*M* 1942) Managing Dir., Sterigas Construction Co., Ltd., 2 Gorst Rd., London, N. W. 10, and • 64 Crowshott Ave., Stanmore, Middlesex, England.

HARVEY, Lyle C. (*M* 1928) Pres., • The Bryant Heater Co., 17825 St. Clair, Cleveland 10, and 2666 Leighton Rd., Cleveland, Ohio.

HASHAGEN, John B. (*M* 1930) Prod. Mgr., of Food Products, 1750 N. Ashland Ave., and • 4236 N. Mozart St., Chicago, Ill.

HASTINGS, Addison (*M* 1942) Asst. Secy., Sales Engr., Burnham-Boiler Corp., and • Lewis Rd., Irvington, N. Y.

HATCH, George (*A* 1941) Htg. & Vent., Engr., • 3527 Carnation Ave., Los Angeles 26, Calif.

HATCH, O. J. (*A* 1941) Mgr., • Clare Bros. Western, Ltd., 179 Notre Dame Ave. E., and 26 Rosemount Apts., 351 River Ave., Winnipeg, Man., Canada.

HATTERSLEY, Eugene H. (*M* 1943) Member of Firm, • A. Hattersley & Sons, 212 E. Main St., Ft. Wayne 2, Ind.

HATTIS, Robert E. (*M* 1926) Cons. Engr., • 26 E. Jackson Blvd., Chicago 4, and 1454 Fargo Ave., Chicago 26, Ill.

HAUAN, Merlin J. (*M* 1933) Cons. Engr., • 3412 16th St. S., Seattle, Wash.

HAUER, Fred W. (*A* 1938) Owner, • Fred Hauer & Co., 315 Elmhurst Ave., Peoria 4, Ill.

HAUF, Joseph C., Jr. (*M* 1944) Exec. Sales Engr., Geroter May Corp., Box 86, Baltimore, and • 4204 Tuscany Court, Baltimore 10, Md.

HAUS, Irvin J. (*A* 1937; *J* 1935) Supt. of Maintenance, • Nash Kelvinator Corp., Seaman Body Plant, 3880 N. Richards St., Milwaukee 1, and 2211 North 51st St., Milwaukee 8, Wis.

HAUSMAN, L. M. (*M* 1945) Pres., • L. M. Hausman & Co., P. O. Box 1729, Manila, P. I.

HAUSS, Charles F.* (*Charter Member*; *Life Member*) Retired, and • Ball Heights, California, Ky.

HAUSSER, Fred L. (*A* 1945) Partner, Ward & Hausser, 17 Seyms St., Hartford 5, and • 178 Four Mile Rd., West Hartford 7, Conn.

HAWES, Harold D. (*J* 1942; *S* 1940) Lt., U. S. Army; and ● 454 Toilsome Hill Rd., Bridgeport, Conn.

HAWISHER, Harold H. (*A* 1938) Mech. Engr., ● Automatic Heating & Engineering Co., 416-418 N. Main St., and 411 S. Jameson Ave., Lima, Ohio.

HAWK, Charles A., Jr. (*A* 1944) Mgr. Engrg. Service Dept., A. M. Byers Co., Clark Bldg., Pittsburgh; and ● 604 Pennridge Rd., Chatham Village, Pittsburgh 11, Pa.

HAYES, James J. (*M* 1920) Vice-Pres., ● Stannard Power Equipment Co., 53 W. Jackson Blvd., Chicago 4; and 2515 East 77th St., Chicago 49, Ill.

HAYES, Joseph G. (*Life Member*; *M* 1908) Pres. & Engr., ● Hayes Brothers, Inc., 236 W. Vermont St., Indianapolis 4, and 2849 N. Capital Ave., Indianapolis 8, Ind.

HAYMAN, A. Eugene, Jr. (*A* 1941; *J* 1935; *S* 1930) Sun Oil Co., Marcus Hook, Pa., and ● 2715 Washington St., Wilmington 239, Del.

HAYNES, Charles V. (*Life Member*; *M* 1917); (*Presidential Member*), (Pres., 1934; 1st Vice-Pres., 1933; 2nd Vice-Pres., 1932; Council, 1926-29; 1932-35) P. O Box 26, Ardmore, Mont. Co., Pa., and ● P. O. Box E, Woodford Sta., Portland 5, Maine.

HAYS, Herbert G. (*M* 1945) Chief Engr., ● Armstrong Furnace Co., 1639 Olentangy River Rd., and 1066 Chelsea Ave., Columbus, Ohio.

HAZLETT, T. Lyle, M. D. (*M* 1938) Medical Dir., ● Westinghouse Electric Corp., East Pittsburgh, Pa.

HEACOCK, Grosvenor W. (*A* 1945) Mgr., Htg. Specialties Div., ● O'Brien Specialty Co., 205 Harrison St., Syracuse 2, and 1105 E. Genesee St., Syracuse, N. Y.

HEAGERTY, William H. (*A* 1940) Sales Engr., ● Chandler Bldg., and 5100 N. Capitol St., Washington, D. C.

HEAPHY, J. Arthur (*A* 1945) Pres. & Owner, ● D. J. Heaphy & Son, 133 N. Geddes St., Syracuse 4, and 101 Wellesley Rd., Syracuse, N. Y.

HEATH, George A. (*A* 1944) Sales Engr., ● P. O. Box 207, and 24 W. Marble St., Mechanicsburg, Pa.

HEATH, William R. (*M* 1931) Asst. Chief Engr., ● Buffalo Forge Co., 490 Broadway, and 119 Wingate Ave., Buffalo, N. Y.

HEAVEN, Lewis P. (*M* 1943) Owner, ● Heaven Engineering Co., 1529 Wyandotte, Kansas City 8, and 2942 Bales Ave., Kansas City 3, Mo.

HEBLEY, Henry F. (*M* 1934) Dir. of Research, ● Pittsburgh Coal Co., P. O. Box 146, Pittsburgh 30, and 216 Jefferson Dr., Mt. Lebanon, Pittsburgh 16, Pa.

HECKEL, Edmund P. (*M* 1918) ● E. P. Heckel & Assoc., 100 W. Monroe St., Chicago, and 314 Cuttriss Pl., Park Ridge, Ill.

HECKEL, Edmund P., Jr. (*J* 1941) ● Buffalo Forge Co., 490 Broadway, Buffalo 4, and 226 Linwood Ave., Buffalo 9, N. Y.

HECKERT, Willard H. (*A* 1944) Htg. & Fuel Oil Div. Mgr., American Lubricants, Inc., 1575 Clinton St., Buffalo, and ● 616 Kennison Pkwy., R. D. 3, Hamburg, N. Y.

HEDEEN, Laurel E. (*M* 1942; *A* 1941; *J* 1938) Cons. Engr., ● Todd, Hedeen & Assocs., 180 W. First St., and 1421 Bertch, Waterloo, Iowa.

HEDGES, H. Berkley (*M* 1919) Mgr., Indus. Sales, ● John J. Nesbitt, Inc., State Rd. and Rhawn St., Holmesburg, Philadelphia 36, and 114 Waverly Rd., Wyncote, Pa.

HEDLEY, Park S. (*M* 1923) Partner, ● Park S. Hedley Co., 45 Allen St., Buffalo 2, and 31 Westgate Rd., Kenmore 17, N. Y.

HEEBNER, Walter M. (*M* 1922) Sales Engr., Warren Webster & Co., 20 Washington Pl., Newark 2, and ● 282 Highwood St., Teaneck, N. J.

HEGELE, Richard H. (*A* 1945) Owner & Mgr., H. W. Hegele Co., 231 N. Main St., and ● 338 E. Court, Urbana, Ohio.

HEIBEL, Walter E. (*M* 1917) Dist. Mgr., ● Aerofin Corp., 11 West 42nd St., New York 18, N. Y., and 12 Richmond Dr., Old Greenwich, Conn.

HEILMAN, Russell H.* (*M* 1923) Sr. Fellow, ● Mellon Institute, 4400 Fifth Ave., Pittsburgh 13, and 2303 Beechwood Blvd., Pittsburgh 17, Pa.

HEINDEL, Richard A. (*M* 1943) Sr. Engr., U. S. Navy Dept., Washington, D. C., and ● 709 S. Ivy St., Arlington, Va.

HEINKEL, Charles E. (*J* 1938) Branch Mgr., ● The Trane Co., 426 Postal Bldg., Portland 4, and 1620 E. Burnside, Portland 14, Ore.

HEISINGER, Frederick W. (*A* 1945) Mgr., Htg. & Vent. Div., ● Benedict & Benedict, 1433 E. Colorado St., Pasadena, and 3375 Gale Ave., Long Beach 6, Calif.

HEISTERKAMP, Herbert W. (*A* 1940; *J* 1937) Mgr. Dehumidifier Div., ● Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, and 175 East 207th St., Euclid 19, Ohio.

HELBURN, I. B. (*M* 1929; *J* 1927) Jr. Asso., ● Wyman Engineering, 1306 Chamber of Commerce Bldg., Cincinnati 2, and 781 Clinton Springs, Cincinnati 29, Ohio.

HELFINSTINE, Roy J. (*M* 1943) Mech. Engr., ● Illinois State Geological Survey, 214 Geological Survey Laboratory, Urbana, and 55 E. Armory, Champaign, Ill.

HELLEND, Albert E. (*M* 1944) Lt., Engrg. Office, U. S. N. R., U. S. S. Kline, A. P. D. 120, c/o Fleet Post Office, San Francisco, Calif., and ● 4549 N. Magnolia Ave., Chicago, Ill.

HELLER, Joseph A. (*A* 1938) ● Air Conditioning Utilities Co., 8 West 40th St., New York 18, and 150 West 79th St., New York 24, N. Y.

HELLMERS, Charles C., Jr. (*A* 1942; *J* 1937) Piping Supvsr., Stone & Webster Engineering Corp., Clinton Engineer Works, P. O. Box 1751, Knoxville, and ● 184 Ash, Rock Gardens, Alcoa, Tenn.

HELLSTROM, John (*A* 1929) Vice-Pres., ● American Air Filter Co., Inc., 215 Central Ave., and 423 Lightfoot Rd., Louisville, Ky.

HELMS, Loyd J. (*M* 1945) Mech. Engr., English & Lauer, Inc., 1978 S. Los Angeles St., Los Angeles 11, and ● 633 Berendo, Los Angeles 5, Calif.

HELSTROM, Clifford W. (*M* 1938) Sales Mgr. & Vice-Pres., Globe Machinery & Supply Co., E. First and Court Ave., and ● 1614 Thompson Ave., Des Moines 16, Iowa.

HELSTROM, Herman G. (*M* 1936) Northwest Sales Mgr., William Brothers Boiler & Manufacturing Co., 1057 Tenth Ave. S.E., and ● 4608 Arden Ave. S., Minneapolis 10, Minn.

HELWICK, Numa J. (*M* 1940) Partner, ● American Heating & Plumbing Co., 829 Baronne St., and 809 Greenwood Dr., New Orleans, La.

HELWIG, E. Cornelius (*A* 1945) Asst. to Pres., ● Tanner & Co., 409 S. Pennsylvania St. Indianapolis 9, and 4162 N. Meridian St., Indianapolis 8, Ind.

HEMEON, Wesley C. L. (*M* 1944) Vent. Engr., ● Industrial Hygiene Foundation, Mellon Institute, 4400 Fifth Ave., and 4111 Allequippa St., Pittsburgh, Pa.

HEMINGWAY, John C., Jr. (*A* 1943) Northern Calif. Distributor, H. C. Little Burner Co., Inc., Second and Lincoln St., San Rafael, and ● Box 62, 101 Magnolia Ave., Larkspur, Calif.

HENDERSON, Alexander S. (*J* 1940; *S* 1938) Design Engr., ● Air Control Equipment Pty., Ltd., P. O. Box 53, Waterloo, and 7 Elizabeth St., Sydney, N. S. W., Australia.

HENDRICKSON, Harold M. (*M* 1934) Air Conditioning Engr., ● Safeway Stores, Inc., P. O. Box 660, Oakland 4, and 2517-99th Ave., Oakland 3, Calif.

HENDRICKSON, W. B. (*A* 1940) ● Elmer, N. J.

HENDRIKSEN, Leonard (*A* 1938) Owner, ● Hendriksen Sheet Metal & Heating Service, 1919 N. Vernon Ave., Flint 6, Mich.

HENDRIX, Thomas K. (*S* 1944) Ens., ● U. S. S. L. S. T. 267, c/o Fleet Post Office, San Francisco, Calif., and 571 W. Prairie, Decatur, Ill.

HENEGAR, Barnie V. (*A* 1944) Owner, ● Southland Supply Co., 2400 S. Harwood, Dallas 1, and 4565 Belclair St., Dallas, Texas.

HENENFELD, Henryk (*M* 1944) Air Cond. Engr., ● A. S. E. A. Electric, Ltd., Yaffi Bldg., Goa St., Fort, Bombay, India.

HENION, Hudson D. (*A* 1923) Sales Mgr., ● C. A. Dunham Co., Ltd., 1523 Davenport Rd., Toronto 4, and 45 Ridge Dr., Toronto, Ont., Canada.

HENNESSY, William J. (*M* 1938) Air Cond. Operation, U. S. Rubber Co., Munition Div., Des Moines Ordnance Plant, and ● 1238-47th St., Des Moines, Iowa.

HENRY, Alexander S., Jr. (*M* 1930) Mech. Engr., R. K. O., 1270 Sixth Ave., and ● 300 Central Park W., New York 24, N. Y.

HENRY, Ernest C. (*M* 1938) Owner, E. C. Henry Co., 1317 S. Water St., and ● 1115 Park Ave., Bay City, Mich.

HENRY, Ray W. (*A* 1945) W. B. Young Supply Co., 208 Delaware St., Kansas City 6, Mo., and ● 2652 Minnesota Ave., Kansas City 2, Kans.

HENSON, Wendell L. (*A* 1944) Htg. & Vtg. Engr., ● Hughes Heating Co., 670 Madison Ave., and 354 Prescott, Memphis, Tenn.

HENZE, Arthur L. (*M* 1945) Mech. Engr., ● Walter Scholer, 1114 State St., Lafayette, and 616 Hillcrest Rd., West Lafayette, Ind.

HENZEL, John H. (*M* 1945) Mech. Engr., Arthur G. McKee & Co., 2300 Chester Ave., Cleveland, and ● 10804 Elmwood Ave., Garfield Heights, Ohio.

HEPBURN, E. M. (*A* 1940) Branch Mgr., ● Empire Brass Manufacturing Co., Ltd., 74 Princess St., and 954 McMillan Ave., Winnipeg, Man., Canada.

HERBERT, James S. (*A* 1946; *J* 1940) Asst. Engr., ● Blue Ridge Glass Corp., and 1325 Catawba St., Kingsport, Tenn.

HERBERT, Richard M. (*A* 1946; *J* 1938) Lt., U. S. Army Air Corps., Lubbock Army Flying School, and ● 2314 Sixth St., Lubbock, Texas.

HERING, Alfred (*M* 1935) Pres., Hering Heating Co., Inc., and ● 39 West 71st St., New York 23, N. Y.

HERKIMER, Harold (*A* 1945)* Design Engr., Raisler Corp., 129 Amsterdam Ave., New York 23, and ● 140 Riverside Dr., New York 24, N. Y.

HERKIMER, Herbert (*M* 1945) Cost Appraiser, Maintenance & Repair Dept., War Shipping Administration, 39 Broadway, and ● 140 Riverside Dr., New York, N. Y.

HERMAN, Harry H. (*M* 1944) Owner, ● Harry H. Herman, 1228 California St., and 300 Cherry St., Denver, Colo.

HERMAN, Nell B. (*A* 1943; *J* 1937; *S* 1936) Lt., U. S. Navy, U. S. S. Vinton A. K. A. 83, F. P. O., San Francisco, Calif., and ● 4217 Garfield Ave. S., Minneapolis 9, Minn.

HERMANN, Harold N. (*M* 1942) Pres., ● Carl J. Kiefer Assoc., Inc., 701 Enquirer Bldg., Cincinnati 2, and Vernon Manor, Cincinnati 19, Ohio.

HERO, George A., Jr. (*M* 1940) Comdr., U. S. N. R., ● C. A. S. U. 64, Fleet Post Office, San Francisco, Calif., and P. O. Box 84, Gretna, La.

HERRE, H. A. (*J* 1941; *S* 1940) Lt. (j.g.) ● Fleet Post Office, San Francisco, Calif.

HERRICK, Marvin M. (*M* 1945) Application Engr., ● Johnson Service Co., 507 E. Michigan St., Milwaukee, and 118 Wilson Ave., Waukesha, Wis.

HERRING, Edgar (*Life Member*; *M* 1919) Chairman and Governing Dir., ● J. Jeffreys & Co., St. George's House, 195-203 Waterloo Rd., London, S. E. 1, and "Kenia," Keswick Rd., Putney, London, S. W., England.

HERRMAN, Donald D. (*A* 1945) Chief Engr., Hartzell Propeller Fan Co., and ● 1407 Nicklin Ave., Piqua, Ohio.

HERSH, Franklin C. (*M* 1939; *A* 1937) Tech. Consultant, Pennsylvania Power & Light Co., 901 Hamilton St., and ● 317 South 16th St., Allentown, Pa.

HERSHEY, Albert E.* (*M* 1940) Research Engr., Westinghouse Electric & Manufacturing Co., East Pittsburgh, and ● 10806 Frankstown Rd., Pittsburgh 21, Pa.

HERSKE, Arthur R. (*M* 1926) Pres., Au-Temp Co., Corp., 521 Fifth Ave., New York 17, and ● 630 Gramatan Ave., Apt. 6H, Mt. Vernon, N. Y.

HERTY, Frank B. (*M* 1933) ● 814 Poinsettia Rd., Columbia 50, S. C.

HERTZLER, John R. (*M* 1936; *J* 1928) Vice-Pres. and Gen. Sales Mgr., York Corp., and ● 863 S. George St., York, Pa.

HESS, Armin E. (*A* 1945) Branch Mgr., ● General Controls Co., M & M Bldg., Rm. 928-A, Houston 2, and 1600 Louisiana St., Houston, Texas.

HESS, Arthur J. (*M* 1937) Member-Owner, ● Hess-Greiner & Polland, 1700 S. Main St., Los Angeles 15, and 897 Linda Vista Ave., Pasadena 2, Calif.

HESSELSCHWERDT, August L., Jr. (*M* 1940; *J* 1937) Asst. Prof., Mech. Engrg., ● Massachusetts Institute of Technology, Cambridge 39, and 28 Hillcrest Rd., Milton 86, Mass.

HESSLER, Lester W. (*M* 1936) Mgr., The Trane Co., 1835 N. Third St., and ● 6034 N. Bayridge Ave., Milwaukee, Wis.

HESTER, Thomas J. (*M* 1919) Pres., ● Hester Bradley Co., 2835 Washington Ave., St. Louis 3, and 4406 McPherson Ave., St. Louis 8, Mo.

HEWETT, John B. (*M* 1937; *A* 1935) Sales Mgr., Anemostat Corp. of America, 10 East 39th St., and ● 64 Buena Vista Dr., Dobbs Ferry, N. Y.

HEYMSFIELD, Herbert R. (*M* 1941) Automatic Htg. Instructor, Brooklyn Technical High School, and ● 478 Third St., Brooklyn 15, N. Y.

HEYSE, Harold S. (*A* 1944) Mgr. & Pres., ● Heyse Sheet Metal Works, P. O. Box 677, and 1033 North Institute, Colorado Springs, Colo.

HEYWOOD, W. J. T., Jr. (*M* 1945) Consult. Engr., H. B. van Zelm, 11 Asylum St., Hartford, and ● 14 Hartt Lane, Newington, Conn.

HICKEY, C. B. (*M* 1945) Sales Engr., Hendrie & Bolthoff Mfg. & Supply Co., 1635-17th St., P. O. Box 5510, Denver 17, and ● 4849 West 32nd Ave., Denver 12, Colo.

HICKEY, Daniel W. (*A* 1931) Pres., ● D. W. Hickey & Co., Inc., 1841 University Ave., St. Paul 4, Minn.

HICKMAN, Herbert V. (*M* 1945; *A* 1938) Htg. Engr. & Gen. Mgr., ● Neil H. Peterson Co., 1129 Folsom St., San Francisco 1, and 11 Palm Ave., San Rafael, Calif.

HIERS, Charles R. (*M* 1929; *J* 1927) Modine Mfg. Co., 101 Park Ave., New York 17, and ● 19 Westminster Rd., Great Neck, N. Y.

HIGH, A. K. (*A* 1945) Owner, ● Automatic Heating Service Co., 421 W. Marshall St., Norristown, and Washington Sq. Gardens, Norristown R. D. 3, Pa.

HIGH, John M. (*M* 1940; *A* 1938) Mgr., Insulation Div., ● The Ruberoid Co., 500 Fifth Ave., New York, N. Y.

HILDER, Frederick L. (*M* 1937) Chief Engr., Electric Furnace Man. Inc., Fourth and Furnace Sts., Emmaus, and ● 2604 Walnut St., Allentown, Pa.

HILL, Edward, Jr. (*A* 1942; *J* 1939) Capt., ● U. S. Army, 0306714, 2836 E. P. D. Co., A. P. O. 350, c/o Postmaster, New York, N. Y., and 102 Washington St., Greenwood, Ind.

HILL, E. Vernon, M. D.* (*Life Member*; *M* 1914; *A* 1912) (*Presidential Member*), (Pres., 1920; 1st Vice-Pres., 1919; 2nd Vice-Pres., 1918; Council, 1915-21) Pres., E. Vernon Hill, and ● 6826 W. Highland Ave., Chicago 31, Ill.

HILL, Fred M. (*M* 1944) Estimator, Htg., Vent. & Air Cond., and ● 225 East Ave. 39, Los Angeles 31, Calif.

HILL, Harold H. (*M* 1935) Branch Mgr., ● American Blower Corp., 438 Woodward Bldg., Washington 5, and 3002 Rodman St. N.W., Washington 8, D. C.

HILL, H. Gordon (*M* 1943; *J* 1938) Lt. Engr., Royal Canadian Naval Volunteer Reserve, Naval Service Hqs., Ottawa, and ● 175 Clendenan Ave., Toronto, Ont., Canada.

HILL, Jared A. (*M* 1938) Dom. Utilization Engr., Pacific Gas & Electric Co., 245 Market St., San Francisco, and ● 715 Laurel Ave., Burlingame, Calif.

HILL, N. Edward (*M* 1944) Chief Designer, L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., Milwaukee 7, and ● 842 North 54th St., Milwaukee 13, Wisc.

HILL, Vaughn H. (*M* 1943; *J* 1938) Indus. Hygiene Engr., Michigan Dept. of Health, Lansing 4, and ● 2111 Colvin Court, Lansing 10, Mich.

HILLARD, Henry F. (*M* 1945) Chief Engr., Distribution Div., ● Mountain Fuel Supply Co., 36 S. State St., and 125 South 13th East St., Salt Lake City, Utah.

HILLEN, William G. (*M* 1945; *A* 1944) Asst. Sales Mgr., Carrier Corp., and ● 120 Sherbourne Rd., Syracuse, N. Y.

HILLS, Arthur H. (*M* 1924) Gen. Mgr., Penn Controls, Ltd., 284 King St. W., and ● 21 Nealon Ave., Toronto, Ont., Canada.

HILMER, George O. (*A* 1944) Designing Engr., Marine Dept., Carrier Engineering Corp., 405 Lexington Ave., New York 17, and ● 231 Stuyvesant Ave., Brooklyn 21, N. Y.

HILMER, Ralph H. (*M* 1944) Engr., C. G. Hokanson Co., 8373 Melrose Ave., and ● 6501 Colgate Ave., Los Angeles 36, Calif.

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JAMISON, Ernest K. (*A* 1944) Contract Engr., The Huffman-Wolfe Co., 308 Standard Bldg., and • 3369 Walters Ct. N.E., Atlanta, Ga.

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JONES, Edwin A. (*M* 1919) Staff Engr., Chrysler Corp., Airtemp Div., and ● 1848 Tennyson Ave., Dayton 6, Ohio.

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JONES, Joe T. (*A* 1945) Applications Engr., ● Fulton Sylphon Co., 200 Southern Bldg., Washington 5, and 1428 Saratoga Ave. N.E., Washington 18, D. C.

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JONES, Lawrence K. (*M* 1939) Mgr., Special Test Sect., ● Pittsburgh Testing Laboratory, 1330 Locust St., and 320 S. Aiken Ave., Pittsburgh, Pa.

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JONES, William C. (*M* 1941) Mgr., ● Johnson Service Co., 1931 K St. N.W., Washington 6, D. C., and Triadelphia Rd., R. F. D. 2, Ellicott City, Md.

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JOYCE, James J. (*A* 1941) Branch Mgr., ● The Powers Regulator Co., 135 Luckie St. N.W., Atlanta 3, and 5 Gatewood Rd., Emory, Ga.

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KARLSTEEN, Gustav. H. (*M* 1935) Plant Engr., Dunlop Tire & Rubber Corp., Station B, Buffalo, and ● Box 55, Route 1, Tonawanda, N. Y.

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LAUTZ, Fritz A. (*M* 1936) Maintenance Engr., American Bemberg Corp., Elizabethton, and ●R. F. D. 1, Johnson City, Tenn.

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LAVORGNA, Michael L. (*M* 1941; *A* 1940) Mgr., Milwaukee Sales, ●L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., Milwaukee 7, and 4472 N. Murray Ave., Milwaukee 11, Wis.

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LAWRENCE, Floyd D. (*A* 1938) Field Repr., ●Clarage Fan Co., 500 Fifth Ave., Rm. 1024, New York 18, N. Y., and 60 Gardiner St., Noroton Heights, Conn.

LAWRENCE, John F. (*A* 1945; *J* 1944) Engr., Designer, ●P. O. Box 127, Tomah, and 130 S. Eighth St., LaCrosse, Wis.

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LECUREUX, E. (*A* 1943) Pres. & Mgr., ●Automatic Gas Co., P. O. Box 29, Athens, Ala.

LEDBETTER, Norvell C. (*M* 1944) Partner, Automatic Heating & Service Co., 22 S. Cooper St., Memphis 4, and ●1935 Nelson Ave., Memphis, Tenn.

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LEE, Burton H. (*M* 1945; *A* 1940) Chief Engr., ●Mance Corp., 400 W. Broadway, New York 12, and 50 Mohican Park Ave., Dobbs Ferry, N. Y.

LEE, Carl M. (*A* 1944) Foreman, Air Cond. Dept., Allison Div., General Motors Corp., Plant No. 5, Tibbs and Raymond St., Indianapolis, and ●4506 Farnsworth St., Indianapolis 3, Ind.

LEE, James A. (*M* 1943; *A* 1937) Dir. of Purchasing, Nash Motors Div., Nash Kelvinator Corp., and ●7301 Fifth Ave., Kenosha, Wis.

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LEE, Robert J. (*M* 1941) Lt. (j.g.), U. S. N. R., ●623 Font Blvd., Parkmerced, San Francisco 12, Calif., and 2356–77th Ave., Philadelphia 38, Pa.

LEE, Robert T. (*A* 1946; *J* 1937; *S* 1936) Chief Engr., Charlotte War Products Pool, Box 59, Charlotte 1, N. C., and ●P. O. Box 164, Lynchburg, Va.

LEEK, Charles W. (*M* 1938) Managing Dir., Leek & Co., Ltd., 1111 Homer St., and ●4682 W. Sixth Ave., Vancouver, B. C., Canada.

LEEK, Walter (*Life Member*; *M* 1903) Pres., ●Leek & Co., Ltd., 1111 Homer St., and 4769 W. Second Ave., Vancouver, B. C., Canada.

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LEESE, Milton L. (*A* 1945) Sales Engr., A-1 Industrial Equipment, 1202 Frankford Ave., Philadelphia 25, and ●2044 S. Redfield St., Philadelphia 43, Pa.

LEFEBVRE, Eugene J. (*M* 1937) Engr., Warden King, Ltd., 2104 Bennett Ave., Montreal, and ●378 Wood Ave., Westmount, Que., Canada.

LEFFEL, Paul C. (*A* 1941) Owner, ●The Leffel Co., 3323 Main, Kansas City 2, and 316 East 75th St., Kansas City, Mo.

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LEHMAN, Max G. (*A* 1937) Owner, ●M. G. Lehman, 720 O St., and 3225 W. Pershing Rd., Lincoln, Nebr.

LEIBY, Robert S. (*A* 1944) Owner & Mgr., ●Robert S. Leiby Sheet Metal Co., 212 N. Grant Ave., Columbus, and 6870 Havens-Corners Rd., New Albany, Ohio.

LEICHNITZ, Robert W. (*A* 1944; *J* 1936) Installation Supvsr., Leichnitz-Bibb, Inc., and ●2506 W. Chestnut, Yakima, Wash.

LEILICH, Roger L. (*M* 1922) Pres., ●The Wallace Stebbins Co., 100 S. Charles St., Baltimore 1, and Ruxton, Baltimore Co., Md.

LEINROTH, J. P. (*M* 1929) Gen. Indus. Fuel Repr., ●Public Service Electric & Gas Co., 80 Park Pl., Newark, and 37 The Fairway, Montclair, N. J.

LEITCH, Arthur S. (*M* 1908) Pres. & Managing Dir., The Arthur S. Leitch Co., Ltd., 1123 Bay St., Toronto 5, and ●421 Russel Hill Rd., Toronto 12, Ont., Canada.

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LEVITT, Leroy L. (*A* 1942; *J* 1940) Lt., U.S.N.R., and ● 3326 Gwynns Falls Pkwy., Baltimore, Md.

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LEWIS, Carroll E. (*M* 1930) Vice-Pres., The Perfex Corp., 500 W. Oklahoma, and ● 5108 N. Woodburn Ave., Milwaukee 11, Wis.

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LEWIS, Gelder V. (*M* 1944) Pfc., U. S. M. C., "K" Co., 3rd Bat., 22nd Marines, 6th Div., c/o F. P. O., San Francisco, Calif., and ● c/o John G. Lewis, Plumbing & Heating Co., 412 East 31st St., Kansas City 2, Mo.

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LEWIS, Herald F. (*M* 1940; *A* 1937) Major, Infantry, ● Commanding HQ Co., E. D. C., Governors Island, N. Y.

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LEWIS, S. David (*A* 1945) Mgr., ● Westinghouse Electric Elevator Co., 842 Rockefeller Bldg., Cleveland 13, and 114 Park Lane Villa, Cleveland 6, Ohio.

LEWIS, Samuel R.* (*Life Member*; *M* 1905), (*Presidential Member*). (Pres., 1914; 2nd Vice-Pres., 1910; Board of Governors, 1909-12; Council, 1914-15); Cons. Engr., ● 100 W. Monroe St., Chicago 3, and 4737 Kimbark Ave., Chicago, Ill.

LEWIS, Velma L., Mrs. (*A* 1943) Htg. & Combus. Engrg., ● c/o Major Herald Lewis, Hq. Eastern Defense Command, Governors Island, N. Y.

LEY, Ralph B. (*A* 1940) Chief, Assignment Sect., Corps of Eng. War Dept., 23rd and C Sts. N.W., Washington, D. C., and ● 916 Park Ave., Falls Church, Va.

L'HOMMEDIEU, Curtiss L. (*M* 1944) Htg., Vent. & Fire Protection Engr., ● Foskett & Bishop Co., 76 Blatchley Ave., New Haven, and Rimmon Rd., Woodbridge, Conn.

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LIGHT, John C. (*A* 1938) Major, Ord. Dept., and ● 3004 Alamogordo St., El Paso, Texas.

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LIMBACHER, Howard R. (*M* 1943) Research Engr., ● Ingersoll Steel & Disc Div., Borg-Warner Corp., 763 E. Vine St., and 525 Edgemoor Ave., Kalamazoo, Mich.

LINCOLN, Roland L. (*M* 1935) Mgr., Dust Lab., B. F. Sturtevant Co., Hyde Park 36, Boston, and ● Box 97, Dover, Mass.

LINDSAY, Griffith W. (*M* 1937) Supvsr. of Estimating & Time Study, Chrysler Corp., Airtemp Div., Dayton, and ● 301 N. Garland Ave., Dayton 3, Ohio.

LINDSLEY, Forrest A. (*A* 1943) Sales & Service Engr., Crane Co., 150 Randolph, Detroit 26, and ● 151 University Ave., Ferndale 20, Mich.

LINDSTROM, Alvin L. (*M* 1945) Cons. Mech. Engr., ● 1001 Mortgage Guarantee Bldg., Atlanta 3, and 1272 Stillwood Dr. N.E., Atlanta, Ga.

LINDSTROM, Donald F. (*A* 1946; *J* 1941) Mech. Engr. & Owner, ● 505 Mortgage Guarantee Bldg., Atlanta 3, and 1259 N. Highland Ave. N.E., Atlanta, Ga.

LINGEN, Ralph A. (*A* 1939; *J* 1938) Dist. Mgr., ● American Foundry & Furnace Co., 709 North 11th St., Milwaukee, and 6028 W. Wisconsin Ave., Wauwatosa, Wis.

LINK, Charles H. (*S* 1941) Link Co., 414 N. Blackstone, and ● 607½ W. Franklin St., Jackson, Mich.

LINSENMEYER, F. J. (*M* 1944) ● Hoffman Combustion Engineering Co., 710 Marquette Bldg., and 19020 Warrington Dr., Detroit, Mich.

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LITTLE, Herbert J. (*J* 1945) Asst. Dvlpt. Engr., ● Anglo-Iranian Oil Co., Abadan, Iran.

LITTLE, Kenneth B. (*A* 1943) Owner, ● Kenneth B. Little Co., 832 Temple Bar Bldg., Cincinnati 2, Ohio.

LITTLE, Raymond (*M* 1944; *A* 1943) Gen. Sales Mgr., Equitable Gas Co., 120 Cecil Way, Duquesne Bldg., Pittsburgh 19, and ● 81 Roycroft Ave., Pittsburgh 16, Pa.

LITTLE, Vernon R. (*A* 1944) Chief Steam Engr., Curtiss-Wright Corp., 4500 E. Fifth Ave., and ● 1456 Fair Ave., Columbus, Ohio.

LITTLEFORD, Wallace H. (*M* 1936) Vice-Pres., ● E. J. Febrey & Co., Inc., 616 New York Ave. N.W., Washington, D. C., and 5703-36th Ave., Hyattsville, Md.

LIVAUDAIS, Marcel (*M* 1945) Chief Draftsman & Mech. Engr., Headquarters 8th Naval Dist., 1015 Federal Bldg., New Orleans, and ● 319 Audubon St., New Orleans 18, La.

LIVELY, George P. (*M* 1942) Principal Engr. (N.A.), Bureau of Ships, Navy Dept., and ● 818 S. Pitt St., Alexandria, Va.

LIVERMORE, James N.* (*M* 1939) Mech. Engr., ● The Detroit Edison Co., 2000 Second Ave., Detroit 26, and 5 Hanover Rd., Pleasant Ridge, Mich.

LIX-KLETT, Luis (*A* 1944) Chief Engr. & Dir., ● Carrier Lix-Klett, S. A., Florida 229, Buenos Aires, Argentina, S. A.

LLOYD, Edmund H. (*A* 1943; *J* 1936) Capt., A. U. S. (C.E.), Office, Chief of Engrs., Repairs and Utilities Br., New War Bldg., 21st and Virginia Ave., and ● 2614-39th St. N.W., Washington, D. C.

LLOYD, Edward C. (*M* 1927) Armstrong Cork Co., and ● R. D. 5, Old Philadelphia Pike, Lancaster, Pa.

LOBSTEIN, Melville C. (*M* 1941) Chief Engr., Museum of Science & Industry, 57th St. and Outer Dr., Chicago 37, and ● 1401 S. Highland Ave., Berwyn, Ill.

LOCHER, David E. (*M* 1945) Pres. & Mgr., ● Coolidge Locher Co., Box 949, 224 S. Staples St., and 215 E. Oleander St., Corpus Christi, Texas.

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LOCKE, Robert A. (*M* 1935) Comdr., U. S. N. R., Naval Inspector of Ordnance, Philadelphia 40, and ● 305 Brentford Rd., Haverford, Pa.

LOCKHART, Harold A. (*M* 1944; *A* 1936; *J* 1935) Chief Engr., Bell & Gossett Co., Morton Grove, and ● Box 5385, R. R. 2, Des Plaines, Ill.

LOCKWOOD, James R. (*A* 1944) Supervising Mech. Engr., ● Familian Pipe & Supply Co., 9430 Rayo Ave., South Gate, and 4540 Simpson Ave., North Hollywood, Calif.

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LOFTUS, Robert G. (*A* 1945) Maintenance Engr., Canada Cycle & Motor Co., Ltd., and ● 6 River Lea Rd., Weston, Ont., Canada.

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LONG, David A. (*A* 1943) Sales Engr., Trane Company of Canada, Ltd., 4 Mowat Ave., Toronto, and ● Centre Rd. S., Cooksville, Ont., Canada.

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LONGCOY, Grant B. (*M* 1933) Engr., Joseph Breslove, 1149 Leader Bldg., Cleveland 14, and ● 1215 Ramona Ave., Lakewood 7, Ohio.

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LUTY, Donald J. (*M* 1933) Gen. Mgr., Htg. Div., and Chief Research Engr., ● Gar Wood Industries, Inc., 7924 Riopelle St., Detroit 11, and 30 Colorado Ave., Detroit 3, Mich.

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McCONNER, Charles R. (*A* 1925; *J* 1922) Gen. Sales Mgr., • Clarage Fan Co., Kalamazoo 16F, and 1904 Waite Ave., Kalamazoo 44, Mich.

McCORMACK, Denis (*M* 1933) Pres., • Fox Industries, Inc., 21st and Arch Sts., Philadelphia 3, and The Kenilworth, Alden Park, Gtn., Philadelphia 44, Pa.

McCORMICK, George W., Jr. (*A* 1941) Capt., U. S. M. C., Combat Intelligence Officer, and • 1320 Sheridan Rd., Menominee, Mich.

McCOY, C. E. (*M* 1936) Partner, • Turner-McCoy, 315 W. Second St., Little Rock, Ark.

McCOY, Thomas F. (*M* 1924) Mgr., • The Powers Regulator Co., 125 St. Botolph St., Boston 15, Mass.

McCREA, Joseph B. (*M* 1937) • 2919 Drexel Ave., Detroit, Mich.

McCREA, Lester W. (*M* 1940) Prop., • McCrea Sales Co., 15 East 21st St., Baltimore 18, and 564 W. University Pkwy., Baltimore 10, Md.

McCULLEY, D. E. (*A* 1941) Sales Engr., • 814 South 14th St., Omaha 2, and 5504 Corby St., Omaha 4, Nebr.

McCULLOUGH, Henry G. (*M* 1936) Vice-Pres., S. S. Fretz, Jr., Inc., 1902 Chestnut St., and • 7042 Lincoln Dr., Mt. Airy, Philadelphia, Pa.

McCULLOUGH, John L. (*M* 1939) Engr., Harry Dougherty & Son, Freeport, and • 731 Country Club Dr., Pittsburgh 16, Pa.

McCULLY, Robert E. (*A* 1945) Sales Engr., Bay Cities Asbestos Co., Fifth Ave. and East 12th, Oakland, and • 2619 Eighth Ave., Oakland 6, Calif.

McCUMBER, Burton R. (*M* 1945) Staff Asst., Carrier Corp., Syracuse, and • West Lake Rd., Skaneateles, N. Y.

McCUNE, Laban J. (*M* 1943) Bldg. Supt. & Engr., • Springer Interests, 118 E. Sixth St., Tulsa 3, Okla.

McCUSKER, James P. (*J* 1944; *S* 1940) Student, Catholic University of America, and • 1445 Evarts St. N.E., Washington, D. C.

McDANIELS, James A. (*A* 1944) Air Cond., Allison Div., General Motors Corp., Speedway City, and • 4950 Ford St., Speedway City, Indianapolis 8, Ind.

McDERMOTT, John P. (*A* 1942; *J* 1939) Major, C. E., Asst. Div. Engr., • Alaskan Div., Air Transport Command, A. P. O. 462, Minneapolis, Minn.

McDONALD, Frank L. (*M* 1944) Branch Mgr., • Barber-Colman Co., 816 W. Fifth St., Los Angeles, and 1655 Rose Ave., San Marino, Calif.

McDONALD, Ivan (*A* 1938) Capt., Overseas, and • Minneapolis-Honeywell Regulator Co., 117 Peter St., Toronto, Ont., Canada.

McDONALD, John J. (*M* 1943) Pres., The McDonald Co., 43-47 Lansdowne St., Boston, and • 401 Beacon St., Chestnut Hill 67, Mass.

McDONALD, Thomas (*A* 1931) Vice-Pres., • Minneapolis-Honeywell Regulator Co., 2747 Fourth Ave. S., Minneapolis 8, Minn.

McDONNELL, Everett N. (*M* 1923), (Council, 1940-45) Pres., • McDonnell & Miller, 400 N. Michigan Ave., Chicago 11, and 219 Lake Shore Dr., Chicago, Ill.

McDONNELL, John E. (*A* 1936) Sales Engr., • McDonnell & Miller, 400 N. Michigan Ave., Chicago, and 2299 Lakeside Pl., Highland Park, Ill.

McDOWELL, Harry L. (*J* 1939) Lt. U. S. N. R., • c/o A. & R. Dept., Naval Air Station, Seattle, Wash., and 1123 Belt Line Blvd., Columbia, S. C.

McEAHERN, Edward J. (*A* 1945; *J* 1944) Vice-Pres., William A. Flatt Co., 1031 Santa Fe, Denver, and • 4754 Raleigh St., Denver 12, Colo.

McELGIN, John W.* (*A* 1937; *J* 1931) Engr., J. J. Nesbitt, Inc., Holmesburg, Philadelphia, and • Tennis Ave., Ambler, Pa.

McFADDEN, H. J. (*A* 1944) Sales Engr., • Servel, Inc., 706 Lone Star Gas Bldg., Dallas 1, Texas, and 1920 Northwest 32nd St., Oklahoma City, Okla.

McFARLAN, A. I. (*M* 1942) Pres., A. I. McFarlan Co., Inc., 21 West St., New York 6, N. Y., and • 691 Dorian Rd., Westfield, N. J.

McGEARY, Paul W. (*M* 1944) Mgr., Indiana Dist. Office, • The Trane Co., 635 N. Penn. and 3630 N. Penn, Indianapolis, Ind.

McGEORGE, Richard H. (*M* 1937) Mgr., Htg. & Air Cond. Dept., McCord Radiator & Manufacturing Co., 2587 E. Grand Blvd., and • 14565 Glastonbury Rd., Detroit, Mich.

McGINNIS, F. L. (*M* 1940) Supt., Coley & Petersen, 409 West 21st St., Norfolk, and • 332 N. Henry St., Williamsburg, Va.

McGOVERN, Arthur F. (*A* 1945) Dist. Supvsr. & Engr., • Modine Manufacturing Co., 83 S. High St., Columbus 15, and 362 S. Harding Rd., Columbus 9, Ohio.

McGOWN, Frederick H., Jr. (*J* 1941; *S* 1939) Cpl., U. S. Army, and • c/o The McGown Co., Cooperstown, N. Y.

McGRAIL, Thomas E. (*M* 1926) Local Repr., • Canadian Sirocco Co., Ltd., 63 Sparks St., Ottawa, Ont., Canada.

McGRATH, William L. (*M* 1945) Project Engr., Carrier Corp., Syracuse, and • 113 Carlton Rd., Syracuse 4, N. Y.

McGREW, Joseph A. (*A* 1944) Owner, • J. A. McGrew Supply Co., 2163 Market St., Denver 2, and 2380 Leyden St., Denver 7, Colo.

McGRUER, A. E. (*M* 1945) Supvsr., Stationary Boiler Plants, • Canadian Pacific Railway Co., Rm. 238, Union Station, and 1954 Bloor St., W., Toronto, Ont., Canada.

McGUINESS, C. H. (*A* 1943) Owner, Iowa Kol-Master Stokers, 205 Plymouth Bldg., Des Moines 9, and • 218 Franklin, Des Moines, Iowa.

McGUIRE, James W. (*M* 1945) Owner, • 370 Lexington Ave., New York, and 86-22 Dongan Ave., Elmhurst, L. I., N. Y.

McILVAINE, John H. (*M* 1929) Pres., • McIlvaine Products, Inc., 1516 Callowhill St., Philadelphia 30, and 601 Pembroke Rd., Bryn Mawr, Pa.

McINDOE, James F. (*M* 1939) Mgr., Combustion Equip. Div., Enterprise Engine and Foundry Co., 18th and Florida Sts., San Francisco, and • 1900 46th Ave., San Francisco 16, Calif.

McINTIRE, James F.* (*M* 1915; *A* 1914), (*Presidential Member*), (Pres., 1939; 1st Vice-Pres., 1938; 2nd Vice-Pres., 1937; Council, 1926-28; 1932-40). Retired • 320 Madison Ave., Clearwater, Fla.

McINTOSH, Fabian C. (*M* 1921; *J* 1917), (Council, 1929-31; 1933-35; 1942) Mgr., Pittsburgh Office, • Johnson Service Co., 1238 Brighton Rd., Pittsburgh 12, and 3650 Perrysville Ave., Pittsburgh 14, Pa.

McINTOSH, Robert M. (*M* 1945) Mech. Engr., Harley-Ellington & Day, 1507 Stroh Bldg., Detroit, and • 15624 Cleveland Ave., Allen Park, Mich.

McKAY, Albert W. (*M* 1942) Major, U. S. A., and • 102 S. Kensington St., Arlington, Va.

McKAY, William H. (*A* 1945) Supt. of Stoker Engrg. Dept., • The Winnipeg Supply & Fuel Co., Ltd., 812 Boyd Bldg., Winnipeg, and 465 Scotia St., Kildonan, Man., Canada.

McKEE, James M. (*A* 1945) Engr., Payne Furnace Co., 336 N. Foothill Rd., Beverly Hills, and • 1219A-19th St., Santa Monica, Calif.

McKENZIE, Murdock C. (*M* 1938) Htg. Engr., Southern California Gas Co., 810 S. Flower St., and • 3806 Boyce Ave., Los Angeles 26, Calif.

McKERNAN, Gordon S. (*A* 1942) Secy.-Treas., • Reg. H. Steen, Ltd., 12 Humewood Dr., and 7 Grimthorpe Rd., Toronto 10, Ont., Canada.

McKINNEY, Carl A. (*A* 1939; *J* 1937) Engr., United Gas Corp., United Gas Bldg., and • 2136 Addison Rd., Houston 5, Texas.

McKINNEY, William J. (*M* 1938; *A* 1934) Southern Dist. Mgr., • American Blower Corp., Rm. 714-101 Marietta St. Bldg., Atlanta 3, and 3363 Mathieson Rd. N.E., Atlanta, Ga.

McKINSTRY, Merrill W. (*A* 1945) Supt., W. E. Beggs Co., 234 Ninth N., Seattle, and ● 1825–41st Ave. N., Seattle 2, Wash.

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McLAREN, Frederic S. (*A* 1945) Secy.-Treas., Air Conditioning Engineering Corp., 501 Sinclair Bldg., and ● 3704 Westcliff Rd. S., Fort Worth, Texas.

McLAREN, T. H. (*A* 1938) Asst. Gen. Mgr. & Gen. Sales Mgr., ● The James Morrison Brass Manufacturing Co., Ltd., 276-278 King St. W., Toronto 1, and 65 Beaufort Rd., Toronto 8, Ont., Canada.

McLARNEY, Harry W. (*M* 1933) Indus. Engr., ● Union Electric Co. of Missouri, 315 North 12th Blvd., St. Louis 1, and 807 Hawbrook Rd., Glendale, Mo.

McLEAN, Dermid (*M* 1917) Mech. Engr., ● Snyder & McLean, 2214 Penobscot Bldg., Detroit 26, and 14007 Lauder Ave., Detroit 27, Mich.

McLEISH, William S. (*A* 1932; *J* 1928) Chief Engr., ● The Ric-wil Co., and 3463 New Portage Uniontown Rd., Barberton, Ohio.

McLENEGAN, D. W.* (*M* 1933) Designing Engr., ● General Electric Co., 5 Lawrence St., Bloomfield, and 39 Francisco Ave., Caldwell, N. J.

McLENNAN, Millard C. (*M* 1944) Supvsr. Htg. & Vent. Design, Puget Sound Navy Yard, Bremerton, and ● 1714 Fourth Ave. N., Seattle 9, Wash.

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McLOUTH, Bruce F. (*M* 1936; *J* 1934) ● 3445 Humboldt Ave. S., Minneapolis 8, Minn., and 135 Gunson St., East Lansing, Mich.

McMAHON, Louis A. (*A* 1944) Heating Engr., ● American Furnace Co., 2719 Delmar Blvd., St. Louis 3, and 10 Magnolia, Ladue, Mo.

McMAHON, Thomas W. (*M* 1928) Dist. Mgr., ● American Blower Corp., 1711 Railway Exchange Bldg., St. Louis 1, and 6173 Waterman Ave., St. Louis 12, Mo.

McMASTER, Sherman G. (*A* 1945) Mgr. of Htg. Dept., ● Richards Manufacturing Co., 135 Front Ave. N. W., Grand Rapids 4, and 164 Wallingwood Ave. N.E., Grand Rapids 3, Mich.

McMICHAEL, Chester L. (*M* 1943) Engr., H. H. Wright Co., 1322 Walnut St., Kansas City, Mo., and ● 5519 Aberdeen Rd., Kansas City, Kans.

McMULLEN, C. S. (*A* 1945) Air Cond. Engr., ● The Trane Co., and 715 S. Fourth St., La Crosse, Wis.

McMULLEN, Earl (*M* 1938) Dir. of Research, ● The Eagle-Picher Co., C & Porter Sts., and Olivia Apts., Joplin, Mo.

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McNAMARA, William (*A* 1930) Mgr., ● The Trane Co., 850 Cromwell Ave., and 1355 Como Ave., St. Paul, Minn.

McNAMEE, Earl W. (*M* 1940) Engr., ● B. & J. Jacobs Co., 1729 John St., Cincinnati 14, and 2627 Ocosta Ave., Cincinnati 11, Ohio.

McNEVIN, Joseph E. (*M* 1937) Mgr., ● Colorado Heating Co., 950 Cherokee St., Denver 4, and 1221 Sherman St., Denver 3, Colo.

McPHERSON, William A. (*M* 1929) Chief, Htg.-Vtg. Div., Department of School Building, 26 Norman St., Boston, and ● 86 Dwinnell St., West Roxbury, Mass.

McQUAID, Daniel J. (*M* 1934) Owner, ● Dan J. McQuaid Engineering Service, 1742-46 Arapahoe St., Denver, Colo.

McQUEEN, Boston (*A* 1945) Engr., ● Hall-Neal Furnace Co., 1324 N. Capitol Ave., and 541 Dayton Ave., Indianapolis, Ind.

McQUITTY, Roy M. (*M* 1945) Steam Utilization Engr., ● Union Electric Company of Missouri, 315 North 12th Blvd., St. Louis, and 1219 Highland Terrace, Richmond Heights, Mo.

McRAE, M. W. (*M* 1939) Research Engr., ● Crane Co., 836 S. Michigan Ave., Chicago 5, and 816 Fairview Ave., Park Ridge, Ill.

McWILLIAMS, Joseph W. (*A* 1942) Camera Works, ● Eastman Kodak Co., 333 State St., Rochester 4, and 71 Lyndale Dr., Rochester 11, N. Y.

MEAD, E. A. (*M* 1926) Sales Mgr., ● The Nash Engineering Co., South Norwalk, and 5 Thames St., Norwalk, Conn.

MEAD, H. K. (*A* 1939) Owner, ● 1100 Guardian Bldg., Portland 4, and Jennings Lodge, Ore.

MEAGHER, Arthur T. (*M* 1938) Dir. & Sales Mgr., Plbg. & Htg. Dept., William Stairs, Son & Morrow, Ltd., 174-190 Lower Water St., and ● 83 Seymour St., Halifax, Nova Scotia, Canada.

MEANS, James MacG. (*A* 1944) Owner, ● Mfrs. Agent, 244 Washington St., Rm. 433, Boston, and Off Proctor St., Manchester, Mass.

MEATES, Richard F. (*A* 1944) ● 60-04–153rd St., Flushing, N. Y.

MEDCALF, Lloyd C. (*M* 1944) Cons. Engr., ● 100 Stevens Ave., Mt. Vernon, and 15 McBride Ave., White Plains, N. Y.

MEHAFFEY, Robert V. (*M* 1945) Cons. Engr., ● Robert V. Mehaffey, Associated Engrs., 30 N. LaSalle St., and 10563 S. Prospect Ave., Chicago, Ill.

MEHLEK, John F. (*A* 1945) Dist. Sales Mgr., ● Erie City Iron Works, 520 Union Bldg., 1836 Euclid Ave., Cleveland 15, and 3120 Chadbourne Rd., Shaker Heights 20, Ohio.

MEHNE, Carl A. (*M* 1929) Htg. & Vent Expert, and ● 35 E. Livingston St., Valhalla, N. Y.

MEILLER, Daniel V. (*A* 1941) Sr. Design Engr., ● Public Service Company of Northern Illinois, 1001 S. Taylor Ave., Oak Park, and 1819 South 16th Ave., Maywood, Ill.

MEINHOLTZ, Herbert W. (*M* 1936) Application Engr., York Corp., York, Pa., and ● 1615 N. Springwood Dr., Silver Spring, Md.

MELLON, James T. J. (*Life Member*; *M* 1911), (Council, 1915) ● Mellon Co., 4415-21 Ludlow St., Philadelphia 4, and 431 North 63rd St., Philadelphia, Pa.

MELNICK, Nicholas A. (*M* 1941) Engr., G. M. Simonson, 625 Market St., and ● 279 Fifth Ave., San Francisco 18, Calif.

MELONEY, Edward J. (*M* 1937) Vice-Pres.-Secy., ● Bowers Bros. Co., 2015 Sansom St., Philadelphia 3, and 100 E. Stewart Ave., Lansdowne, Pa.

MELTON, Rupert D. (*A* 1946; *J* 1942) Page & Co., 1001 N. Church St., and ● 117 W. Tenth St., Charlotte, N. C.

MENCH, John G. (*M* 1945) Mech. Engr., James Stewart Corp., 231 S. LaSalle St., Chicago, and ● 915 Elm St., Park Ridge, Ill.

MENDEN, Peter J. (*M* 1935) Cons. Engr., Rm. 503 Janes Bldg., and ● 1232 Hayes Ave., Racine, Wis.

MENDITCH, Barney (*J* 1944) Htg. & Air Cond. Engr., ● American Heating Co., 55 K St. S.E., and 1651 Wisconsin Ave. N.W., Washington, D. C.

MEREDITH, John W. (*M* 1944) Htg. Engr., U. S. Navy, Boston Yard, Charlestown, and ● 1 Sunnyside Rd., Natick, Mass.

MERENS, Seymour H. (*A* 1939) Owner, ● Max Miller & Co., 823 N. California Ave., and 4955 N. Whipple St., Chicago, Ill.

MERGARDT, Albert P. (*A* 1940) Prop., ● American Heating Co., 55 K St. S.E., Washington, D. C., and 3905 N. Fifth St., Arlington, Va.

MERRILL, Carle J. (*M* 1919) Treas., ● C. J. Merrill, Inc., 54 St. John St., and 31 Craigie St., Portland 4, Maine.

MERRILL, Frank A. (*M* 1934) Cons. Engr., ● Office of Hollis French, 210 South St., Boston, and 19 Auburndale Rd., Marblehead, Mass.

MERRYMAN, C. P. (*A* 1945) Owner, ● Merryman Wilson Co., 121 S. First St., and 625 E. Main St., Union City, Tenn.

MERTZ, Walter A. (*M* 1919) Secy., ● The Kehm Corp., 51 E. Grand Ave., and 3753 N. Keeler Ave., Chicago, Ill.

MERWIN, Gile E. (*M* 1924; *J* 1923) Mgr., ● The Trane Co., 5012 Parker St., Omaha 4, Nebr.

METCALF, Ralph H. (*M* 1944) Vice-Pres., ● Kremer Hicks Co., 3974 Delmar Ave., St. Louis, and 101 Jefferson Rd., Webster Groves, Mo.

METZGER, A. F. (*M* 1940) Supvsr. of Steam Utilization Sec., ● Allegheny County Steam Heating Co., 435 Sixth Ave., Pittsburgh, and 3421 Hornie St., Pittsburgh 19, Pa.

METZGER, H. J. (*A* 1937) Pres., ● Wheeler-Blaney Co., 137 E. Water St., and 706 Locust St., Kalamazoo, Mich.

MEYER, E. Walter (*A* 1944) Partner, ● Keystone Sales Co., P. O. Box 4201, Bellevue Sta., Pittsburgh 2, and 6933 Prospect Ave., Ben Avon, Pittsburgh, Pa.

MEYER, Frank L.* (*M* 1932; *J* 1928) Pres., ● The Meyer Furnace Co., Peoria 2, and 9 Cole Court, Peoria 5, Ill.

MEYER, Henry C., Jr.* (*Life Member*; *M* 1898), (Council, 1915-16) Pres., ● Meyer, Strong & Jones, Inc., 101 Park Ave., New York, N. Y., and 25 Highland Ave., Montclair, N. J.

MEYER, John S. (*M* 1945) Sales Promotion Mgr., W. A. Case & Son Manufacturing Co., Buffalo, and ● 38 Crescent Ave., Buffalo 14, N. Y.

MEYER, Karl A. (*M* 1938) Product Engr., Herman Nelson Corp., Moline, and ● 2438–33rd Ave., Rock Island, Ill.

MEYERS, Carl F. (*A* 1942) Engr., Coblentz Equipment Co., 1119 Peach St., and ● 142 East 35th St., Erie, Pa.

MICHAEL, Arthur F. (*M* 1945) Gen. Supvsr. Air Cond., ● Southern California Gas Co., Box 3249 Terminal Annex, Los Angeles 54, and 5266 Dahlia Dr., Los Angeles 41, Calif.

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MICHAEL, Robert K. (*A* 1944) Property Inspector, The Prudential Insurance Company of America, 500 Board of Trade Bldg., Kansas City, Mo., and ● 620 Fairfax St., Denver, Colo.

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MILENER, Eugene D. (*M* 1936) Secy., Indus. & Commercial Gas Section, ● American Gas Assoc., 420 Lexington Ave., New York 17, N. Y.

MILES, James C. (*M* 1943) ● Miles Heating Enterprises, 1836 Euclid Ave., Cleveland 15, and 1972 Ford Dr., Cleveland 6, Ohio.

MILES, Oscar K. (*A* 1945) Mech. Engr., General Engineering Co., 301 West 13th St., and ● 2119 Edwin St., Fort Worth, Texas.

MILEY, Julian J. (*A* 1945) Cons. Engr., Htg. & Air Cond., ● 61 Florence Ave., San Anselmo, Calif.

MILLARD, Junius W. (*M* 1929) Comdr., U. S. Navy, U. S. S. Alcor (AD 34), c/o F. P. O., San Fancisco, Calif., and ● 3516 Valley Dr., Parkfairfax, Alexandria, Va.

MILLER, A. T. (*M* 1938) Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, and ● 125 Godwin Ave., Ridgewood, N. J.

MILLER, Charles A. (*Life Member*; *A* 1917) ● The H. B. Smith Co., Inc., 331 Madison Ave., and 2554 Marion Ave., Bronx, N. Y.

MILLER, Charles W. (*M* 1919; *J* 1908) Pres., ● The Rado Co., 759 N. Milwaukee St., Rm. 204, Milwaukee 2, and R. 1, Box 42, Menomonee Falls, Wis.

MILLER, David C. (*M* 1944) Chief Engr., Buerkel & Co., Inc., 24 Union Park St., Boston, and ● 5 Ellis St., Quincy 69, Mass.

MILLER, Edgar R. (*A* 1935) Chief Engr., ● Winnipeg Cold Storage Co., Ltd., Salter and Jarvis Ave., and P. O. Box 1384, Winnipeg, Man., Canada.

MILLER, Floyd A. (*Life Member*; *M* 1911) Retired, and ● 944 Montrose Ave., Chicago, Ill.

MILLER, Frederick T. J. (*A* 1943) Service Mgr., Hurst Oil Co., 924 West 21st St., Norfolk, and ● 5713 Herbert St., Norfolk 2, Va.

MILLER, George F. (*M* 1936) Sales Engr., ● Geo. F. Miller, 1614 K St. N.W., Washington 6, D. C., and 5608 Grove St., Chevy Chase 15, Md.

MILLER, H. Deane (*A* 1944) Mgr., Air Cond. & Comml. Refrig. Dept., ● General Electric Supply Corp., P. O. Box 1919, Salt Lake City 12, and 1866 Valecrest Ave., Salt Lake City, Utah.

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MILLER, Leo B. (*M* 1926) Sales Exec., ● Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, and 3481 N. Hackett Ave., Milwaukee, Wis.

MILLER, Lorin G.* (*M* 1933), (Council, 1942-44) Prof. & Head Mech. Engrg. Dept., ● Michigan State College, R. E. Olds Hall of Engineering, and 232 University Dr., East Lansing, Mich.

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MILLER, Norton R. (*M* 1944) Sales & Application Engr., The Trane Co., 250 East 42nd St., New York 17, and ● 188-17 Palo Alto Ave., Hollis 7, L. I., N. Y.

MILLER, Robert A.* (*M* 1931) Tech. Sales Engr., ● Pittsburgh Plate Glass Co., 2200 Grant Bldg., Pittsburgh 19, and 1211 Carlisle St., Tarentum, Pa.

MILLER, Robert T. (*A* 1927) Chief Engr., Sales Dept., ● Masonite Corp., 111 W. Washington St., Chicago, Ill.

MILLER, Warren W. (*A* 1945) Canadian Sales Engr., ● Frigidaire Products of Canada, Leaside, and 112 Joicey Blvd., Toronto, Ont., Canada.

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MILLER, William T. (*M* 1938) Prof., Htg. & Vtg., ● Purdue University, and 525 Hayes St., West Lafayette, Ind.

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MILLIKEN, J. H.* (*M* 1923) Repr., ● American Air Filter Co., Inc., 228 N. LaSalle St., Chicago 1, and 1021 Ridge Court, Evanston, Ill.

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MILLS, Doremus L. (*A* 1943) Research & Development Engr., ● Revere Copper & Brass, Inc., and 1407 N. Madison St., Rome, N. Y.

MILLS, D. M. (*A* 1940) Mgr., Houston Div., ● F. J. Evans Engineering Co., Gray and Crawford, Houston 3, and 2726 Robinhood, Houston 5, Texas.

MILLS, Hartzell C. (*A* 1935) Industrial Engr., ● Minneapolis Gas Light Co., 739 Marquette Ave., Minneapolis 2, and 4137–10th Ave. S., Minneapolis 7, Minn.

MILTON, Taylor (*M* 1945) Lt., C. E. C., U. S. N. R., U. S. Naval Air Station, Minneapolis, and ● 1457 Scheffer Ave., St. Paul, Minn.

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MOESEL, F. Albert (*A* 1939) Asst. Mgr., • W. A. Case & Son Manufacturing Co., 31 Main St., Buffalo 3, and 382 Argonne Dr., Kenmore, N. Y.

MOFFAT, Ormond G. (*M* 1940; *A* 1937) Mgr., Special Products Section, Canadian Westinghouse Co., Ltd., Sanford Ave., and • 141 George St., Hamilton, Ont., Canada.

MOHAN, John F. (*A* 1944) Secy.-Treas., • Fox Supply Co., Inc., 2229 Blake St., Denver 2, and 1260 St. Paul St., Denver 6, Colo.

MOHN, H. Leroy (*M*-1937) Cons. Engr., • Fitzgibbons Boiler Co., Inc., and 71 W. Fifth St., Oswego, N. Y.

MOHRFELD, Herbert H. (*A* 1943; *J* 1935) Vice-Pres. & Treas., • C. P. Mohrfeld, Inc., 24 Lees Ave., Collingswood, and 670 Station Ave., Haddonfield, N. J.

MOLER, William H. (*M* 1942) Dist. Mgr., • Governair Corp., 504 Great National Life Bldg., Dallas, and 3928 Lovers Lane, Dallas 5, Texas.

MOLFINO, Philip (*M* 1938) Partner, • Leland & Haley, 58 Sutter St., San Francisco 4, and 125 Clayton St., Apt. 2, San Francisco 17, Calif.

MOLLENBERG, Harold J. (*M* 1936) Pres., Mollenberg-Betz Machine Co., 22 Henry St., Buffalo, and • 111 Saratoga Rd., Snyder 21, N. Y.

MOLONY, James J. (*A* 1944) Steam Engineer, Nebraska Defense Corp., Fremont, and • 106 North 49th, Omaha, Nebr.

MOLONEY, Roger R. (*M* 1937) • 26 Bonner Ave., Manly, Sydney, Australia.

MOLTZ, Bernard S. (*A* 1945) Engrg. Draftsman, Standard Engineering Co., 2129 Eve. St. N.W., and • 1293 Brentwood Rd. N.E., Washington, D. C.

MONAHAN, Maurice B. (*M* 1944) Mfrs. Agent, • 1410 Stanley St., and 5537 Trans Island Ave., Montreal, Que., Canada.

MONGILLO, Alphonse J. (*A* 1945) Prop., • A. J. Mongillo Co., 928 Grand Ave., New Haven 11, and 20 Marvel Rd., New Haven, Conn.

MONICK, Fred R. (*A* 1936) Mgr., • American Radiator & Standard Sanitary Corp., 605 E. Eighth St., and 1114 S. Sixth Ave., Sioux Falls, S. D.

MONIES, Ellis A. (*M* 1944) Htg. Engr., Eastern Pennsylvania Supply Co., 56 South Penna. Ave., Wilkes-Barre, and • Pioneer Ave., Dallas, Pa.

MONTESANO, Frank P. (*M* 1944) Mgr., The Frank A. McBride Co., 158-60 Ward St., and • 117 Genessee Ave., Paterson, N. J.

MONTGOMERY, Edward G. (*A* 1938) Special Repr., Steel Co. of Canada, Ltd., 525 Dominion St., Montreal, and • 20 Finchley Rd., Hampstead, Que., Canada.

MONTGOMERY, J. Russell (*A* 1937) Mgr., Standards & Research • Truscon Steel Co., Albert St., Youngstown, and 296 Granada Ave., Youngstown 4, Ohio.

MOODY, Lawrence E. (*M* 1919) Partner, • Moody & Hutchison, 1420 Walnut St., Philadelphia 2, Pa., and 224 Bellevue Ave., Haddonfield, N. J.

MOON, L. Walter (*M* 1915), (Council, 1933-36) Vice-Pres., St. Louis Industrial Truck Co., 7700 E. Railroad Ave., and • 1137A Hornsby Ave., St. Louis 15, Mo.

MOONEY, Bernard P. (*A* 1945) Owner, • Mooneys, 820 W. Lake, and 4745 France Ave. S., Minneapolis, Minn.

MOONEY, Mark E. (*A* 1945) Dir. of Sales, Applied Refrigeration, Carrier Corp., Syracuse, and • 125 Butternut Dr., De Witt, N. Y.

MOORE, Frank C. (*A* 1938) Mgr., Aerofin Div., Vapor Car Htg. Co. of Canada, Ltd., 67 Yonge St., Rm. 1104, and • 44 Lola Rd., Toronto, Ont., Canada.

MOORE, H. Carlton* (*M* 1935) Engr., Metcalf & Eddy, 1300 Statler Bldg., Boston, and • 145 Beaumont Ave., Newtonville 60, Mass.

MOORE, H. Lee (*M* 1919), (Council, 1927-28) Engr., • 431 Fulton Bldg., Pittsburgh 22, and 7065 Flaccus Rd., Ben Avon, Pittsburgh 2, Pa.

MOORE, Henry W. (*M* 1935) Mgr., Air Cond. Dept., 20th Century Refrigeration Co., 1630 Walnut St., Cincinnati 10, and • 3164 Queen City Ave., Cincinnati 11, Ohio.

MOORE, Herbert S. (*A* 1923) Dist. Repr., Iron Fireman Manufacturing Co. of Canada, Ltd., 602 King St. W., and • 107 Clendenan Ave., Toronto, Ont., Canada.

MOORE, MacDonell (*A* 1940) Pres., Gen. Mgr., • The Moore Fuel Corp., 23 Rose St., Danbury, and R. F. D. 2, Bethel, Conn.

MOORE, R. Edwin (*M* 1944; *A* 1928) Vice-Pres., • Bell & Gossett Co., 8200 N. Austin Ave., Morton Grove, and 425 Merrill Ave., Park Ridge, Ill.

MOORE, Robert E. (*A* 1943) Archt., • Moody & Moore, 216 Graham Ave., Winnipeg, Man., Canada.

MOORE, Roscoe S. (*M* 1944) Precipitron Engr., • Westinghouse Electric Corp., 20 N. Wacker Dr., Chicago 6, and 226 N. Humphrey St., Oak Park, Ill.

MOORE, Wesley R. (*M* 1937) Regional Mgr., • Minneapolis-Honeywell Regulator Co., 5005 Euclid Ave., Cleveland, and 14211 Ashwood Rd., Shaker Heights, Ohio.

MORAN, Edward V., Jr. (*J* 1945) Engr., Carrier Corp., 300 S. Geddes St., Syracuse, and • 115 Oakland Ave., Tuckahoe, N. Y.

MORAWECK, Alvin H., Jr. (*A* 1946; *J* 1941) Capt. Cavalry, O. R. C., Ft. Knox, Ky., and • 36 Woodland Rd., Maplewood, N. J.

MOREHOUSE, H. P. (*M* 1933) Gen. Htg. & Air Cond. Repr., Public Service Electric & Gas Co., 80 Park Pl., Newark, and • 30 Timber Acres Rd., Summit, N. J.

MOREHOUSE, J. Stanley (*M* 1938) Dean of Engrg., • Villanova College, Villanova, and 102 Llandoff Rd., Upper Darby, Pa.

MORGAN, Arthur S. (*M* 1938) Mgr., Fess Oil Burners of Canada, Ltd., 85 King St. W., and • 156 Glenmanor Dr., Toronto, Ont., Canada.

MORGAN, Edwin H., Jr. (*A* 1946; *J* 1942) Chief Draftsman, American Cyanamid Co., S. Cherry St., and • 180 North St., Wallingford, Conn.

MORGAN, George R. (*M* 1945) Partner & Dept. Head., • J. E. Sirrine & Co., and 306 Overbrook Rd., Greenville, S. C.

MORGAN, Robert C. (*Life Member*; *M* 1915) Chairman of the Board, • Stewart A. Jellett Co., 1200 Locust St., Philadelphia, and 314 W. Seymour St., Philadelphia 44, Pa.

MORGAN, Robert L. (*A* 1945) Owner, • Morgan Appliance Co., 706 West Gray, and 607 West Gray, Houston, Texas.

MORGAN, Robert W. (*M* 1938) Asst. Chief Engr., • Bridgeport Thermostat Co., 1225 Connecticut Ave., Bridgeport, and R. F. D. 1, Norwalk, Conn.

MORIARTY, John M. (*M* 1937) Retired, and • 1616 Baldwin Ave., Arcadia, Calif.

MORIN, A. R. (*A* 1938) Co-Owner, M. & V. Supply Co., 911 N. Walnut St., Oklahoma City 4, and • 2115 Sherman, Oklahoma City, Okla.

MORIN, Romeo P. (*A* 1944) In Charge of Ventilation, Gibbs & Cox (Branch Office), Todd Hoboken Shipyard, Hoboken, N. J., and • 3337 89th St., Jackson Heights, N. Y.

MORRIS, C. Raymond (*M* 1938) Pres., • Power & Heating Equipment Sales, Inc., 56 N. Spring Garden Ave., Nutley, N. J.

MORRIS, Edward J. (*M* 1942) Mgr., • Morris Engineering Co., 813 N. Calvert St., Baltimore 2, and 3414 Gwynns Fall Pkway., Baltimore 16, Md.

MORRISON, Chester B. (*M* 1931) Managing Dir., • York-Shipley, Ltd., North Circular Rd., Hendon, London N.W. 2, and 60 Harley St., London S.W. 1, England.

MORRISON, Frank (*M* 1943) • Hall-Neal Furnace Co., 1324 N. Capitol, Indianapolis, and Cloverdale, Ind.

MORRISON, W. Bruce (*A* 1942; *J* 1939) Cons. Engr., Bedell Bldg., Portland 4, and • 1805 Northeast 27th Ave., Portland 12, Ore.

MORRISON, W. L. (*A*-1943) U. S. N. R., • U. S. S. Jubilant A-M 255, F. P. O., New York, N. Y., and 3422-16th, Great Bend, Kans.

MORRO, John J. (*M* 1940) Engr. & Gen. Mgr., Paragon Oil Burner Corp., Brooklyn, and • 130 East 40th St., New York, N. Y.

MORROW, Albert P. (*A* 1944) Engr., • Wm. S. Scull Co., Front & Federal Sts., Camden, N. J., and 237 Maple Ave., North Hills, Mont. Co., Pa.

MORROW, J. DeWitt (*A* 1938) Pres. & Mgr., • The Warren Co., Inc., 614 Walker Ave., Houston 2, and 5503 LaBranch, Houston, Texas.

MORSE, Clark T. (*Life Member*; *M* 1913) Pres., • American Blower Corp., P. O. Box 58, Roosevelt Park Annex, Detroit 32, and Hillcrest Dr., Bloomfield Hills, Mich.

MORSE, Elwin F. (*J* 1944) Field Engr., • B. F. Sturtevant Co., Div. of Westinghouse Electric Corp., 933 Leader Bldg., Cleveland 14, and 15126 Clifton Blvd., Lakewood 7, Ohio.

MORSE, Louis S., Jr. (*M* 1938; *J* 1936) Lt., U. S. N. R., and • Lone Pine Rd., Bloomfield Hills, Mich.

MORSE, Robert D. (*M* 1936) • 414 Vance Bldg., Seattle 1, and 4316 East 43rd St., Seattle 5, Wash.

MORTON, Harold S. (*M* 1931) Col., Ordnance Dept., Amm. Div., Office Chief of Ord., Pentagon Bldg., Washington, D. C., and • 7331 Piney Branch Rd., Takoma Park, Md.

MORTON, Paul S. (*A* 1943; *J* 1939) Htg., Plbg., Elec. Sales & Engrg., • 609 Bangor Rd., Lawrence, Mich.

MOSES, Walter B., Jr. (*A* 1946; *J* 1940; *S* 1936) Sales Engr., • 7836 Saint Charles Ave., New Orleans 18, La.

MOSHER, Clarence H. (*A* 1919) Owner, • C. H. Mosher, 423 Ashland Ave., Buffalo 13, N. Y.

MOSHER, Harold A. (*M* 1945) Mech. Engr., • Eastman Kodak Co., Kodak Park Works, Rochester 4, and 287 Winona Blvd., Rochester 5, N. Y.

MOSS, Alfred W. (*A* 1945; *J* 1943) • Walter Woods, Ltd., Winnipeg, and 26 Ferndale Ave., Norwood, Manitoba, Canada.

MOTZ, O. W. (*M* 1932) Cons. Engr., • 920 E. McMillan St., Cincinnati 6, and 2605 Briarcliffe Ave., Cincinnati 13, Ohio.

MOULD, Harry W., Jr. (*J* 1941) Engr., Fedders Mfg. Co., Inc., 57 Tonawanda St., Buffalo, and • 44 Allegany Ave., Kenmore, N. Y.

MOUNTFORD, Ralph E. (*M* 1945) Asst. Plant Mgr., S. Smith & Sons (London), Ltd., Cricklewood, London, N.W. 2, and • 67 Ferncroft Ave., Eastcote, Middlesex, England.

MOULSDALE, Thomas DeWitt (*M* 1943) Htg. Engr., Lehigh Valley Oil Co., 1200 Walnut St., Allentown, and • 818 Coleman, Easton, Pa.

MOYNAN, Joseph S. (*A* 1945) Engr., New Orleans Public Service, Inc., 317 Baronne St., and • 8211 Spruce St., New Orleans, La.

MUCKLE, James M. (*M* 1939) • Anderson Meyer & Co., Ltd., 570 Lexington Ave., New York, and 34-12 Murray St., Flushing, N. Y.

MUEHLIG, Elmer J. (*M* 1945) Constr. Engr., • F. W. Woolworth Co., 800 U. S. National Bank Bldg., Denver 2, and 1930 Holly St., Denver 7, Colo.

MUELLER, Ervin J. (*A* 1944) Sales Engr., Sullivan Valve & Engineering Co., 910 S. Arizona St., and • 1928 Lowell Ave., Butte, Mont.

MUELLER, H. C. (*M* 1936; *A* 1930) Pres. & Gen. Mgr., • The Powers Regulator Co., 2720 Greenview Ave., Chicago 14, and 1277 Forest Glen Dr. N., Winnetka, Ill.

MUELLER, Harold P. (*M* 1936) Pres., • L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., and 511 E. Monrovia Ave., Milwaukee, Wis.

MUELLER, John E. (*M*-1937) Mgr., Comml. Customer Dept., • West Penn Power Co., Box 1736, Pittsburgh 30, and 387 Broadmoor Ave., Pittsburgh 16, Pa.

MUESSIG, James W. (*M* 1938) Sales Engr., • Clarage Fan Co., 333 N. Michigan Ave., Chicago, and 405 Lodge Lane, Lombard, Ill.

MUIRHEID, John G. (*A* 1940; *J* 1937) Vent. Engr., Bureau of Ships, Navy Dept., Washington, D. C., and • 1201 M St. N.W., Washington 5, D. C.

MULHOLLAND, Daniel J. (*A* 1945) Supt., • Consolidated Conditioning Corp., 460 S. Tenth Ave., Mt. Vernon, and 381 Broadway, Dobbs Ferry, N. Y.

MULREY, Maurice D. (*M* 1944) Owner, • T. A. Mulrey & Son, 3161 N. Illinois St., Indianapolis 8, Ind.

MUNIER, Leon L. (*M* 1919; *J* 1915) Pres., • Wolff & Munier, Inc., 222 East 41st St., New York 17, and 63 Columbia Ave., Hartsdale, N. Y.

MUNKELT, Frederick H. (*M* 1938) Vice-Pres., • W. B. Connor Engrg. Corp., 114 East 32nd St., New York 16, and 317 East 17th St., Brooklyn 26, N. Y.

MURDOCK, Charles E. (*M* 1944) Plant Mgr., Rheem Manufacturing Co., Stokermatic Div., 1415 South State, Salt Lake City 4, and • 1254 E. Third S., Salt Lake City 2, Utah.

MURDOCH, John P. (*M* 1937) Pres., • John P. Murdoch Co., 3630 Haverford Ave., Philadelphia 4, and 204 Glenn Rd., Ardmore, Pa.

MURHARD, Erroll A. (*M*-1939) Partner, Muirhead & Murhard Co., 801 S.W. Stark St., Portland 5, and • 2805 S.W. Greenview Court, Portland 1, Ore.

MURPHREE, Robert L. (*A* 1940; *J* 1936) Engr. & Owner, Rogers Plumbing & Heating Co., 2127 Eighth St., and • P. O. Box 864, Tuscaloosa, Ala.

MURPHY, Charles G. (*M* 1942) Foundry Engr., Wright Aeronautical, Lockland, and • 25100 Treadwell Ave., Euclid, Ohio.

MURPHY, Daniel C. (*A* 1940) Sales Repr., • C. A. Dunham Co., 214 Old Colony Bldg., Des Moines 9, and 3900 Grand Ave., Des Moines, Iowa.

MURPHY, Delacour I. (*M*-1941) E. C. Cooley Co., 625 Market St., San Francisco, and • 3027 Millsbrae Ave., Oakland 5, Calif.

MURPHY, Edward T.* (*Life Member*; *M* 1915) Sr. Vice-Pres., • Carrier Corp., 300 S. Geddes St., Syracuse 1, and 1055 James St., Syracuse, N. Y.

MURPHY, Eugene F. (*A* 1944; *J* 1942) Instructor in Mech. Engrg., Dept. of Mech. Engrg., University of California, Berkeley 4, and • Staff Engr., Committee on Prosthetic Devices, Rm. 355, N.W. Technological Institute, Evanston, Ill.

MURPHY, Howard C.* (*M* 1923) Vice-Pres., • American Air Filter Co., Inc., 215 Central Ave., and 495 Lightfoot Rd., Louisville, Ky.

MURPHY, Joseph R. (*M* 1934; *A* 1925) Vice-Pres., • Taco Heaters, Inc., 342 Madison Ave., New York, N. Y., and Terrace Ave., Riverside, Conn.

MURPHY, William W. (*M* 1930) Treas., • W. W. Murphy Co., 424 Worthington St., and 25 Mansfield St., Springfield, Mass.

MURRAY, G. F. J. (*M* 1944) Tech. Mgr., British Doby Stokers, Ltd., 80 Grosvenor St., London W. 1, and • 8 Homewood Rd., St. Albans, Herts., England.

MURRAY, H. G. S. (*M* 1942; *A* 1941; *J* 1936) Sales Engr., Canadian Comstock Co., Ltd., 206 Laird Dr., Leaside, and • 28 Rosedale Heights Dr., Toronto 5, Ont., Canada.

MURRAY, Thomas F. (*M* 1923) Sr. Htg. & Ven. Engr., • N. Y. S. Dept of Public Works, State Office Bldg., Albany, N. Y.

MURRAY, William A. (*M* 1944) Pres., William A. Murray, Inc., 967 Farmington Ave., West Hartford, and • 43 Fairlee Rd., West Hartford 7, Conn.

MURRAY, William W., Jr. (*M* 1944) Executive Vice-Pres., • Almirall & Co., Inc., 53 Park Place, New York 7, and 2 Thornwood Lane, Roslyn Heights, L. I., N. Y.

MURSINNA, Gilbert P. (*A* 1939) Owner, • Gilbert P. Mursinna, 640 Tafel St., Cincinnati 14, and 3657 Boudinot Ave., Cincinnati, Ohio.

MUSE, Mayland H. (*A* 1944) Mgr. & Owner, • Southern Welding Co., Box 454, and 1010 Melrose Ave., Oakland Gardens, Johnson City, Tenn.

MUSGRAVE, Merrill N. (*M* 1944; *A* 1935) Owner, • Merrill N. Musgrave & Co., 2019 Third Ave., Seattle 1, and 1610 E. Boston Terrace, Seattle 2, Wash.

MUSSER, John M. (*M* 1942) Mech. Engr., U. S. Engineers, Hanford Engineer Works, P. O. Box 550, Pasco, and • 1314 Kimball, Richland, Wash.

MUTH, Arnold J. (*M* 1944) Electrical Engr., Albert Kahn Associated Architects & Engineers, Inc., New Center Bldg., Rm. 1008, Detroit, and • 868 W. Bethune Ave., Detroit 2, Mich.

MYER, Haydn (*A* 1940) Pres., • Haydn Myer Co., Inc., 2224 Comer Bldg., Birmingham 3, and 1411 Avon Circle, Redmont Park, Birmingham 5, Ala.

MYERS, George W. F. (*M* 1930; *A* 1928; *J* 1923) Owner, Myers Engineering Equipment Co., 3800 W. Pine Blvd., St. Louis 8, and • 476 Pasadena Ave., Webster Groves 19, Mo.

MYLER, William M., Jr.* (*M* 1937) Chief Engr., Janitrol Engrg. Dept., • Surface Combustion Corp., P. O. Box 267, Columbus 16, and 1545 Grenoble Rd., Columbus, Ohio.

MYRICK, James W. H. (*M* 1944) Engr. & Owner • New England Air Conditioning Co., Boston Globe Bldg., Rm. 431, 244 Washington St., Boston 8, and South Duxbury, Mass.

MYTINGER, Kenneth L. (*M* 1943) Gen. Management, • Mytinger Air Conditioning Co., Inc., 101 Park Ave., New York 17, and 38-19 50th St., Long Island City, N. Y.

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NACHMAN, George P. (*M* 1938) Treas., • The Spohn Heating & Ventilating Co., 1775 East 45th St., Cleveland 3, and 2887 Falmouth Rd., Cleveland 2, Ohio.

NAMAN, Israel A. (*J* 1940) Mech. Engr., Raymond L. Jenkins, 4316 Blossom, Houston, and • 402 Avondale Ave., Houston 6, Texas.

NAPIER, Charles E. (*A* 1945) Sales Engr., • Consolidated Engines & Machinery Co., Ltd., 200 Bay St., and 144 St. George St., Toronto, Ont., Canada.

NAROWETZ, Louis L., Jr. (*M* 1929; *A* 1912) Pres., Narowetz Heating & Ventilating Co., 1722 Washington Blvd., Chicago 12, and • 945 Michigan, Wilmette, Ill.

NARVEY, S. (*J* 1943) Flt./Sgt., • No. 8 Repair Depot, R. C. A. F., Winnipeg 6, and 451 St. Johns Ave., Wp. 6, Manitoba, Canada.

NASS, Arthur F. (*M* 1927) Pres., • McGinness, Smith & McGinness Co., 527 First Ave., Pittsburgh 19, and 29 Elmhurst Rd., Pittsburgh 20, Pa.

NATHANSON, Max (*A* 1943) Owner & Gen. Mgr., • Canadian Armature Works, 6595 St. Urbain St., and 5457 Victoria Ave., Montreal, Que., Canada.

NATION, Oslin (*A* 1944; *J* 1942) Mfrs. Repr., • 9702 El Patio Dr., Dallas 18, Texas.

NATKIN, Alfred J. (*M* 1944) Partner & Office Mgr., • Natkin & Co., 1418 San Jacinto, and 2327 Quenby, Houston, Texas.

NAYLOR, James F., Jr. (*M* 1944) Dist. Mgr., • The Trane Co., 3930 Lindell Blvd., St. Louis 8, 302 E. Argonne Dr., Kirkwood 22, Mo.

NEAL, James P. (*A* 1939) Capt., U. S. Army, Cincinnati Ordnance Dist., 230 E. Ninth St., and • 1428 Herschel Ave., Cincinnati 8, Ohio.

NEARINGBURG, Arthur (*A* 1938) Sales Engr., • Sheldons, Ltd., 1221 Bay St., and 130 Floyd Ave., Toronto, Ont., Canada.

NEE, Raymond M. (*M* 1936) Power Engr., • American Cyanamid Co., 30 Rockefeller Plaza, New York 20, and 5 East 82nd St., New York 28, N. Y.

NEILANS, John L. (*J* 1943) Draftsman, Trane Company of Canada, Ltd., 4 Mowat Ave., and • 330 Woodmount Ave., Toronto, Ont., Canada.

NEIMAN, Charles H., Jr. (*M* 1945) Asst. Dir. of Engrg., York-Shipley, Inc., Jessop Pl. and P. R. R., and • 330 W. Jackson St., York, Pa.

NELSON, Arthur W.* (*M* 1944) Major, C. E., U. S. A., Hdqtrs. Military Govt., U. S. A. F. I. K., Engineer, Army Administration Sec., A. P. O. 235, c/o P. M., San Francisco, Calif., and • 12 Sylvan Rd., Sharon, Mass.

NELSON, Axel A. (*A* 1942) • R. 4, Northcrest Sub Div., North Kansas City, Mo.

NELSON, D. W.* (*M* 1928) Assoc. Prof. Mech. Engrg., • College of Engineering, University of Wisconsin, Mech. Engrg. Bldg., and 3906 Council Crest, Madison, Wis.

NELSON, George O. (*M* 1923) Engr., • Carstens Bros., Ackley, Iowa.

NELSON, Harold M. (*M* 1937) Pres., • H. M. Nelson & Co., Inc., 1223 Connecticut Ave., Washington 6, D. C., and Falls Church, Va.

NELSON, Herman W. (*Life Member*; *M* 1909) Chairman of the Board, • The Herman Nelson Corp., 1824 Third Ave., Moline, Ill.

NELSON, James A., Jr. (*A* 1945) Partner, • James A. Nelson Co., 1375 Howard St., San Francisco 3, and 143 Tuscaloosa Ave., Menlo Park, Calif.

NELSON, L. K. (*M* 1940) Assoc., • James M. Todd, 217 N. Peters, New Orleans 16, and 2502 Palmer Ave., New Orleans, La.

NELSON, Norris T. (*J* 1944) Designer, Htg., Vent. & Air Cond., Bethlehem Steel Co., 20th and Illinois, San Francisco, and • 2421 Buchanan St., San Francisco 15, Calif.

NELSON, Raymond A. (*A* 1945) U. S. Air Conditioning Corp., 2101 N.E. Kennedy St., Minneapolis, and • 3810 Upton Ave. S., Minneapolis 10, Minn.

NELSON, Richard H. (*A* 1933; *J* 1928) Pres. & Gen. Mgr., The Herman Nelson Corp., 1824 Third Ave., and • 1303–30th St., Moline, Ill.

NELSON, Roy O. (*M* 1938) Asst. Sales Mgr., Marsh Heating Equipment Co., 2120 Southport Ave., Chicago 14, and • 5636 N. Bernard St., Chicago 45, Ill.

NELSON, Vernon E. (*A* 1945) Mgr., • General Heating Products Co., 3353 University Ave. S.E., and 5226–34th Ave. S., Minneapolis, Minn.

NELSON, Walter L. (*M* 1945; *A* 1944) Chief Mech. Engr., District Government, District Bldg., and • 2225 Quincy St. N.E., Washington, D. C.

NEMEC, Frank L. (*M* 1945) Cons. Engr., Hudson & Grady, 525 Market St., San Francisco, and • 1140 Oxford Rd., Burlingame, Calif.

NESBITT, Albert J.* (*M* 1921) Pres., • John J. Nesbitt, Inc., State Rd. and Rhawn St., Holmesburg, Philadelphia 36, and Rockfield Farm, Tennis Ave. and Welsh Rd., Ambler, Pa.

NESMITH, Oliver E. (*A* 1936) Dir. of Engrg. & Research, Eureka Vacuum Cleaner Co., Bell & Hanna, and • 107 Warner, Bloomington, Ill.

NESS, William H. C. (*M* 1931) Secy. & Gen. Mgr., • Master Fan Corp., 1323 Channing St., Los Angeles 21, and 215 N. Kingsley Dr., Los Angeles, Calif.

NESSELL, C. W. (*M* 1937) Asst. to Vice-Pres., • Minneapolis-Honeywell Regulator Co., 1101 Vermont Ave. N.W., Rm. 405, Washington 5, D. C., and 2120 Forest Glen Rd., Silver Spring, Md.

NEST, Richard E. (*M* 1936) Cons. Engr., General Oil Burner Co., 1623 N. Aisquith St., Baltimore 2, and • 6000 Pinehurst Rd., Baltimore 12, Md.

NEUBAUER, Edwin W. (*M* 1939) Engr., • Campbell Norquist & Co., 1127 S.W. Morrison St., Portland 5, and 4804 N.E. Davis, Portland 15, Ore.

NEVIN, J. F. (*M* 1944) Major, • A. B. R. O. 427, S. O. Refrigeration, Hq. Allied Land Forces, S. E. Asia, Singapore.

NEWBY, Ira P. (*M* 1941) Htg. & Fuel Unit, U. S. Army, 8th Service Command, Santa Fe Bldg., and • 6262 Prospect, Dallas 14, Texas.

NEWMAN, Harold E. (*M* 1938) Owner & Mgr., • J. F. Dickinson Co., 716 Tenth St., and 514 High St., Modesto, Calif.

NEWPORT, Charles F.* (*Life Member*; *M* 1906) Sales Engr., Weil-McLain Co., Michigan City, Ind., and • 10001 Longwood Dr., Chicago 43, Ill.

NEWTON, Albert E. (*A* 1943) Engr., • Hall-Neal Furnace Co., 1324 N. Capitol Ave., and 4515 E. Washington, Apt. 1, Indianapolis, Ind.

NEWTON, Alwin B. (*M* 1938) Chief Engr., Chrysler Corp., Airtemp Div., 1119 Leo St., Dayton 1, and • 905 E. Schantz Ave., Dayton 9, Ohio.

NEWTON, David A. (*M* 1944) Chief Engr., Larkin Coils, Inc., 519 Memorial Dr., and • 55 25th St. N.W., Atlanta, Ga.

NEYHART, Floyd B. (*J* 1945) Mech. Engr., Carrier Corp., 300 S. Geddes St., Syracuse 1, N. Y., and • Greentown, Ind.

NICHOLLS, John M. (*M* 1939) • Robbins Gamwell Corp., 68 West St., and 32 Buel St., Pittsfield, Mass.

NICHOLS, Howard R. (*A* 1943) Engr., Grudem Brothers Co., 2645 University Ave., St. Paul, and • 5416 Park Pl., Minneapolis, Minn.

NICHOLS, Lawrence J. (*A* 1944) Branch Mgr., American Radiator & Standard Sanitary Corp., 73 E. Naghten St., Columbus, and • 2074 Greenway N., Columbus 15, Ohio.

NICHOLSON, Sterling J. (*A* 1941) Pres. • Nicholson, Inc., Box 317, Durham, N. C.

NICKERSON, Albert E. (*A* 1945) Engr., Asbestos Supply Company of Seattle, 321 First S., Seattle 4, and • 2415–31st W., Seattle 99, Wash.

NICKLE, Arthur J. (*A* 1936) Asst. Sales Mgr., • Darling Bros., Ltd., 140 Prince St., Montreal 3, and 4804 Oxford Ave., Montreal 29, Canada.

NICOLL, Scott F. (*M* 1939) Mech. Engr., • York Corp., York, and 1433 First Ave., Elmwood, York, Pa.

NICOLS, J. A. (*M* 1941) Mech. Engr., General Mills, Inc., 406 Hodgson Bldg., Minneapolis, and • Rt. 2, Interlachen Blvd., Minneapolis 10, Minn.

NIEMOELLER, Arthur R. (*A* 1943) Sales Engr., • 5817 Itaska St., St. Louis 9, Mo.

NIESKE, George F. (*M* 1943) Engr., C. N. Flagg & Co., Inc., 79 Griswold St., and • 794 Bee St., Meriden, Conn.

NIESSE, Joe H. (*M* 1938) Indianapolis Branch Mgr., • Ilg Electric Ventilating Co., 836 Architects & Builders Bldg., Indianapolis 4, and 5837 Winthrop Ave., Indianapolis 5, Ind.

NOBBS, Walter W. (*M* 1919) Cons. Engr., 26 Victoria St., London S.W. 1, and • 50 Fairhazel Gardens, London N.W. 6, England.

NOBIS, H. M. (*M* 1914) • 1827 Stanwood Rd., East Cleveland 12, Ohio.

NOBLE, Milner (*M* 1940) Pres., • Aerofin Corp., 410 S. Geddes St., and 142 Clarke, Syracuse, N. Y.

NOLAN, James J., Jr. (*M* 1939) Principal Engr., Repairs & Utilities Br. Office, Chief of Engrs., War Dept., 21st and Virginia Ave., Washington 25, and • 4024 Calvert St. N.W., Washington 7, D. C.

NOLAN, J. J. (*M* 1943) Prop., • J. J. Nolan & Co., 78 Washington Ave., and 1606 Goodbar, Memphis, Tenn.

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NOLL, William F. (*M* 1924) Prop., • Wm. F. Noll, 629 North 27th St., Milwaukee 8, and 4823 W. Townsend St., Milwaukee 10, Wis.

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NORCROSS, Ivan F. (*A* 1945) Mech. Engr., • E. L. E. Co., 124 W. Fourth St., Los Angeles 13, and 2917 Urban Ave., Santa Monica, Calif.

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NORDINE, L. F. (*M* 1914) The Trane Co., 2701 Ontario Rd. N.W., Washington, D. C., and • 412 N. Royal Ave., Front Royal, Va.

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NORMAN, Edward A., Jr. (*M* 1944) Pres., Norman Products Co., Inc., 1150 Chesapeake Ave., and • 1926 Andover Rd., Columbus 8, Ohio.

NORMAN, George C. (*M* 1944) Engr., • Koithan & Johnson, 27 Washington St., Newark 2, and 12 Exeter Rd., Short Hills, N. J.

NORRINGTON, Walter L. (*A* 1943; *J* 1938) Capt., Ordnance Dept., Office of Chief of Ordnance, Washington, D. C., and • 305 Barbara Fritchie, Beverly Plaza, Alexandria, Va.

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NORTH, Clarence P. (*M* 1942) Chief Engr., • Campbell Heating Co., P. O. Box 833, Des Moines 4, and 3614 E. Seventh St., Des Moines 16, Iowa.

NORTH, Sam L. (*A* 1942) Partner, • North Bros., P. O. Box 252, Atlanta 1, Ga.

NORTON, John A. (*M* 1940) Mgr., Htg. Sales Div., Crane, Ltd., 306 Front St. W., Toronto, and • 136 Hanna Rd., Leaside, Ont., Canada.

NORTON, L. Ivan (*A* 1941) Engr., Design Htg. Installations, • Evoready Plumbing & Heating Co., 514 W. Main St., Washington, Iowa.

NOTKIN, James B. (*M* 1944) Mech. Engr., Public Works Dept., Puget Sound Navy Yard, Bremerton, and • 2013 East 63rd St., Seattle 5, Wash.

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NOTTBERG, Gustav (*A* 1933) Vice-Pres., • U. S. Engineering Co., 914 Campbell and 650 West 67th St., Kansas City, Mo.

NOTTBERG, Henry J. (*M* 1919) Pres., • U. S. Engineering Co., 914 Campbell St., and 150 West 54th St., Kansas City, Mo.

NOTTBERG, Henry, Jr. (*A* 1946; *J* 1937) Lt. (j.g.), C. E. C., U. S. N. R., • 130th Naval Construction Battalion, F. P. O., San Francisco, Calif., and 150 West 54th St., Kansas City, Mo.

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NOWITZKY, Herman S. (*A* 1931) Supt. Construction, Wilmer & Vincent Corp., 1501 Broadway, New York 18, N. Y., and • 821 Llewellyn Ave., Norfolk 7, Va.

NOYES, Richard R. (*A* 1946; *J* 1938) Lt. (j.g.), U. S. N. R., c/o Bureau of Aeronautics Repair, U. S. N., Columbus 16, Ohio, and • 127 Pacific Ave., Toronto 9, Ont., Canada.

NUGENT, Arthur W. (*M* 1945) Mgr., Washington Office, • McQuay, Inc., 606 Colorado Bldg., 14th and G Sts. N.W., Washington, D. C., and 1030 Flower Ave., Takoma Park, Md.

NUNLIST, F. J., Jr. (*M* 1943) Asst. Chief Engr., L. J. Mueller Furnace Co., 2005 W. Oklahoma, Milwaukee 7, and • 528 North 19th St., Milwaukee 3, Wis.

NUSBAUM, Lee* (*M* 1915) Owner, • Pennsylvania Engineering Co., 1119-21 N. Howard St., Philadelphia, and 315 Carpenter Lane, Germantown, Philadelphia, Pa.

NUSBAUM, S. Richard (*A* 1944; *J* 1940) Mgr., • Pennsylvania Engineering Co., 1119-21 N. Howard St., Philadelphia, and Wynnewood Park Apts., Wynnewood, Pa.

NUSSBAUM, Otto J. (*A* 1944) Chief Engr., • Kramer Trenton Co., 626 Brunswick Ave., Trenton 5, and 918 Greenwood Ave., Trenton 9, N. J.

NUTTING, A.* (*M* 1940) Chief Engr., • American Air Filter Co., 215 Central Ave., and 4040 Ormond Rd., Louisville, Ky.

NYE, L. Bert., Jr. (*A* 1943; *J* 1936) Staff Engr., Washington Gas Light Co., 1100 H St. N.W., Washington, D. C., and • McLean, Va.

NYLAND, J. A. (*A* 1944) Mgr., • Nyland Sheet Metal Co., 2323 West 10th St., Indianapolis 8, and 6133 Rosslyn Ave., Indianapolis 5, Ind.

NYQUIST, John D. (*J* 1942; *S* 1941) Model Shop Supt., Collins Radio Co., and • 728 Fifth Ave. S.W., Cedar Rapids, Iowa.

NYSTROM, Paul E. (*M* 1945) Archt. & Partner, • Law, Law, Potter & Nystrom, 121 S. Pinckney St., Madison 3, and 360 Kensington Dr., Madison 4, Wis.

O

OAKLEY, LeRoy W. (*M* 1937) Owner, • L. W. Oakley Sales Co., 408 W. Clinch Ave., and 2003 Laurel Ave., Knoxville, Tenn.

OAKS, Orion O. (*M* 1917) Chief Engr., J. B. Pierce Foundation, Raritan, and • 119 Oak Ridge Ave., Summit, N. J.

O'BANNON, Lester S.* (*M* 1928) Research Engr., • Kentucky Agricultural Experiment Station, University of Kentucky, and 123 State St., Lexington, Ky.

OBERG, H. C. (*A* 1933) Mgr., Engrg. Dept., Crane Co., Fifth and Broadway, and • 1362 W. Minnehaha St., St. Paul, Minn.

OBERSCHULTE, Richard H. (*M* 1944; *J* 1938) Sales & Field Mgr., • D. T. Randall & Co., 404 Boulevard Bldg., Detroit 2, and Box 76, Franklin, Mich.

O'BRIEN, Thomas J. (*M* 1943) • T. J. O'Brien Engineering Co., 1030 Exchange Bldg., and 1483 Vance Ave., Memphis, Tenn.

O'CONNELL, Thomas D'Arcy (*A* 1942) Pres., • Thomas O'Connell, Ltd., 1169 Ottawa St., Montreal 3, and 9 McCullough Ave., Outremont 8, Que., Canada.

O'DANIEL, James A. (*M* 1942) Owner, • Maple City Furnace Co., 603 S. Main St., Monmouth, Ill.

O'DANIEL, Presley (*A* 1945) Branch Mgr., • York Corp., 1137 Sterick Bldg., Memphis 3, and 2167 Poplar, Apt. 12, Memphis 4, Tenn.

O'DONNELL, Lawrence D. (*A* 1944) Mgr. & Partner, • McCarthy's Sheet Metal Works, 113 N. Eighth St., and 816 Buntin St., Vincennes, Ind.

O'DOWER, Hugh J. (*A* 1938) Distributor, • Vilter Manufacturing Co., 114 W. Tenth St., Kansas City 6, and 6844 Locust, Kansas City, Mo.

OELGOETZ, J. F. (*M* 1938) Sole Owner, J. F. Oelgoetz Co., 3365 N. High St., and • 279 E. North Broadway, Columbus, Ohio.

OFFEN, Ben (*M* 1928) Owner, • B. Offen & Co., 343 S. Dearborn St., and Sheridan Rd., Chicago, Ill.

OFFNER, Alfred J.* (*M* 1922), (1st Vice-Pres., 1945; 2nd Vice-Pres., 1944; Treas., 1935-38; Council, 1935-45) Cons. Engr., • 139 East 53rd St., New York 22, and 160-15-11th Ave., Beechurst, L. I., N. Y.

O'GORMAN, J. S., Jr. (*A* 1934) Mgr., Detroit Office, • Johnson Service Co., 230 E. Alexandrine Ave., Detroit 1, and 147 Abbey Rd., Birmingham, Mich.

OLD, William H. (*M* 1937) Asst. Mgr., • Glanz & Killian Co., 1761 Forest Ave. W., Detroit 8, and 18245 Devonshire Rd., R. F. D. 5, Birmingham, Mich.

OLDS, Dean (*M* 1944) Chief Engr., Gas & Oil Div., • The Coleman Lamp & Stove Co., 250 N. St. Francis, and 222 N. Bleckley, Wichita, Kans.

OLDS, Stephen T. (*A* 1945) Sales Mgr. & Engr., • William C. Maunz Co., 1397 Jefferson Ave., Buffalo 8, and 400 Elmwood Ave., Buffalo 9, N. Y.

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OLSEN, Carlton F. (*A* 1925; *J* 1920) Engr. & Sales, Kewanee Boiler Corp., 549 W. Washington Blvd., and • 1000 West 100 St., Chicago 43, Ill.

OLSEN, Gustav E. (*M* 1930) Sales Mgr., Fitzgibbons Boiler Co., Inc., 101 Park Ave., New York, and • 68-09 Beach Channel Dr., Arverne, L. I., N. Y.

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OLSON, Barney (*A* 1929) • Barney Olson, Inc., 122 S. Michigan Ave., and 5724 N. Natoma Ave., Chicago, Ill.

OLSON, Erling O. (*A* 1943) Service Engr., Oil Burner Service Co., Inc., 315 West Lake St., Minneapolis, and • 4221 Oakland Ave., Minneapolis 7, Minn.

OLSON, Eugene O. (*M* 1942) • Delavan Engineering Co., 414-12th St., Des Moines 9, and 2520-45th St., Des Moines 10, Iowa.

OLSON, Milton J. (*A* 1941; *J* 1937) Partner, Olson Bros., 2651 St. Mary's Ave., and • 5830 Hickory St., Omaha, Nebr.

OLSON, Nat I. (*A* 1944) Owner & Gen. Mgr., • United Blower Co., Inc., 193 Centre St., New York 13, and 711 Brightwater Court, Brooklyn, N. Y.

OLSSON, Eric V. (*A* 1945) Field Engr., • B. F. Sturtevant Co., 36 Pearl St., Hartford 3, and Main St., Collinsville, Conn.

OLVANY, William J. (*Life Member*; *M* 1912) Pres., • William J. Olvany, Inc., 100 Charles St., New York 14, and 109-40-71st Rd., Forest Hills, L. I., N. Y.

O'NEILL, James W. (*M* 1929; *A* 1927; *J* 1925) Vice-Pres. in charge of Production, Trane Company of Canada, Ltd., 4 Mowat Ave., and • 55 Highview Crescent, Toronto, Ont., Canada.

O'NEILL, John E. (*A* 1945) Sales Engr., Kerby Saunders, Inc., 330 West 42nd St., New York, and • 81 W. Raleigh Ave., Staten Island, N. Y.

O'NEILL, Joseph Francis (*M* 1944) Mech. Engr., • Trane Company of Canada, Ltd., 4 Mowat Ave., Toronto 1, and 44 Aldgate Ave., Toronto 14, Ont., Canada.

OONK, W. J. (*M* 1937) Dist. Mgr., B. F. Sturtevant Co., 1315 Syndicate Trust Bldg., St. Louis 1, and • 4548 Red Bud Ave., St. Louis 15, Mo.

OOSTEN, Louis S. (*A* 1944; *J* 1938) Asst. Chief Engr., Bell & Gossett Co., 8200 N. Austin Ave., Morton Grove, and • 725 Case St., Evanston, Ill.

OPPERMAN, Everett F. (*A* 1944) Mech. Engr., Petroleum Heat & Power Co., Southfield St., and • 788 E. Main St., Stamford, Conn.

ORABELLA, Michael J. (*A* 1944) Pres. & Mgr. of Engrg. Dept., Ohio National Products, Inc., 10014 Carnegie Ave., Cleveland, and • 13304 First Ave., East Cleveland 12, Ohio.

OREAR, Andrew G. (*M* 1942) Pres., • Trade-Wind Motorfans, Inc., 5725 S. Main St., Los Angeles 37, and 1015 E. Raleigh St., Glendale 5, Calif.

O'REAR, L. R. (*M* 1934) Pres., • Midwest Plumbing & Heating Co., 2450 Blake St., and 825 S. Josephine St., Denver, Colo.

OREBAUGH, Howard T. (*M* 1945) Application Engr., • York Corp., 4660 E. Marginal Way, Seattle 4, and 1129 North 77th St., Seattle 3, Wash.

ORGELMAN, George H. (*J* 1942; *S* 1940) Lt., Hq. B. A. S. E. C. T. W. O., U. S. A. S. C. O. M. C., Engr. Section, A. P. O., 928, c/o Postmaster, San Francisco, Calif., and • 10 Pearl St., Danbury, Conn.

ORMISTON, Jack B. (*A* 1940) Owner, • Ormiston Plumbing & Heating Co., 105 Manning Ave., and 287 Mile Square Rd., Yonkers 2, N. Y.

ORMSBY, H. Kingsley, Jr. (*M* 1944) Owner, • Syracuse General Sales Co., 511 E. Raynor Ave., Syracuse 1, and 206 Roycroft Rd., East Syracuse, N. Y.

ORR, Weldon John (*A* 1944) Mech. Engr., James A. Kearns, 1414 Drummond St., and • 3559 St. Famille St., Apt. 5, Montreal, Que., Canada.

OSBERGER, T. L. (*A* 1944) Mgr., • T. L. Osberger Co., 550 Michigan Trust Bldg., Grand Rapids, and 1208 Dunham St., S.E., Grand Rapids 6, Mich.

OSBORN, Wallace J. (*A* 1927) Vice-Pres., Keeney Publishing Co., 1734 Grand Central Terminal Bldg., New York, N. Y., and • 1029 Old Post Rd., Fairfield, Conn.

OSBORNE, G. H. (*M* 1922) Managing Dir., • The Ventilating & Blow Pipe Co., Ltd., 714 Saint Maurice St., Montreal 3, and 4535 Lacombe Ave., Montreal 26, Que., Canada.

OSBORNE, James M. (*A* 1944) Field Engr., • The Herman Nelson Corp., 1829 "M" St. N.W., Washington 6, D. C., and 4607 Drexel Rd., College Park, Md.

OSBORNE, Stanley R. (*M* 1939) Promotion Engr., • Chase Brass & Copper Co., Wartebury 91, and 115 Grove St., Naugatuck, Conn.

O'SHEA, John J. (*A* 1941) Sales Repr., Buffalo Forge Co., 305 Teckwood Dr. N.E., and • 714 Greenview Ave. N.E., Atlanta, Ga.

OSTER, William P. (*M* 1940) Vice-Pres., • Equitable Equipment Co., Inc., 410 Camp St., New Orleans 12, La.

OSTERHAGE, Catherine Sweeney (*A* 1945) Draftsman, E. I. duPont deNemours & Co., Inc., P. O. Box 100, and • 1025 Winslow Ave., Richland, Wash.

OSTERMEIER, Edwin J. (*A* 1945) Dir. of Engrg., • Apex Engineering Co., Suite 200, Engineers Bldg., 205 W. Wacker Dr., Chicago 6, and 4508 Wilson Ave., Downers Grove, Ill.

OSTROM, Eric W. (*M* 1937) Chief Engr., Air Cond. Dept., A/B Svenska Flaktfabriken, Kungsgatan 16, and • John Ericssonsgatan 18, Stockholm, Sweden.

OTIS, Alfred L. (*A* 1944) Dist. Sales Mgr., Clarage Fan Co., 965 Farmington Ave., and • 43 N. Main, West Hartford, Conn.

OTT, M. Earl (*M* 1942) Owner, • Lakeside Metal Service, 23 East 115th St., Chicago 28, and 8635 W. Wabash Ave., Chicago 19, Ill.

OTT, Oran W. (*M* 1925), (Council, 1934-36) Cons. Mech. Engr., • 111 W. Seventh Bldg., Los Angeles, and 1462 Waverly Rd., San Marino, Calif.

OTTO, Robert W. (*M* 1941) Mech. Engr., Toltz King & Day, Inc., 1509 Pioneer Bldg., and • 2147 Carroll Ave., St. Paul, Minn.

OTTS, John G. (*A* 1942) Capt., A. C., c/o • Climatic Hangar Project Office, Elgin Field, Fla., and c/o Dr. W. W. Carson, Ft. Valley, Ga.

OTTUM, Alvin L. (*M* 1945; *A* 1943) Air Conditioning Sales & Engineering, • Radiant Sales & Engineering Co., 344 N. San Vicente Blvd., Los Angeles 36, Calif.

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OUWENEEL, W. A. (*M* 1937) • Siesel Construction Co., 514 E. Ogden Ave., Milwaukee 2, and 5282 N. Bay Ridge, Milwaukee 11, Wis.

OWEN, Charles E. (*M* 1941) Cons. Engr., Mfrs. Agent, • Owen Engineering Service, and 1218 S. Thompson St., Carbondale, Ill.

OWEN, Charles E., Jr. (*J* 1945) Petty Officer, U. S. N. R., 7th Special Battalion, Co. D. 1, Fleet Post Office, San Francisco, Calif., and • 1218 S. Thompson, Carbondale, Ill.

OWEN, Frank W. (*J* 1945) Petty Officer, U. S. N. R. N. C. T. C. Sta., Force, Barracks I-2, Camp Endicott, Davisville, R. I., and • 1218 S. Thompson St., Carbondale, Ill.

OWEN, Jeff Davis (*M* 1937) Piping Engr., Stone & Webster Engineering Corp., Los Angeles, and • 4070 East Blvd., Culver City, Calif.

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PABST, Charles S. (*M* 1934) Pres., Pabst Air Conditioning Corp., 219 Eagle St., Brooklyn 22, and • 8727 98th St., Woodhaven L. I., N. Y.

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PALMER, E. M. (*A* 1944) Asst. Gen. Sales Mgr., • Kewanee Boiler Corp., and 436 S. Chestnut St., Kewanee, Ill.

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PARK, Harold E. (*A* 1938; *J* 1936) Sales Engr., • Shaw-Perkins Manufacturing Co., 1645 Oliver Bldg., Pittsburgh 22, and 106 W. Littlewood St., Etna, Pittsburgh 23, Pa.

PARK, J. Frank (*M* 1937; *A* 1936; *J* 1930) Pres., • Western Air & Refrigeration, Inc., 1819 Glendale Blvd., and 2160 Kenilworth Ave., Los Angeles 26, Calif.

PARK, Nicholas W. (*M* 1936) Mech. Engr., Heintz Manufacturing Co., Front and Olney Ave., Philadelphia, and • 509 Jericho Rd., Abington, Pa.

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PARKER, Richard A. (*A* 1938) Capt., 0-235379, • 217th General Hospital, A. P. O. 887, c/o Postmaster, New York, N. Y., and 220 El Camino Real, Burlingame, Calif.

PARKES, C. H. (*M* 1944) Owner, • Distributor of Heating & Plumbing Equip., 101½ W. Chestnut, P. O. Box 947, and 1108 W. Chestnut, Dodge City, Kans.

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PARSONS, Lawson G. (*A* 1945) Mgr., Wholesale Plbg. & Htg. Dept., • York Corrugating Co., Adams St. and W. M. R. R., and 50 N. Newberry St., York, Pa.

PARSONS, Roger A. (*A* 1942; *J* 1933) Htg. Engr., • Board of Water & Electric Light Commissioners, 114-16 W. Ottawa St., and 2609 Clifton St., Lansing, Mich.

PARTLAN, Robert L. (*A* 1946; *J* 1940) Pres., • Partlan Sheet Metal Works, 14265 Arlington Ave., Detroit 12, and 9102 Pinehurst Ave., Detroit 4, Mich.

PARVIS, Ralph S. (*M* 1938) Supvsr., Diamond Ice & Coal Co., 827 Market St., and • 1208 Delaware Ave., Wilmington 19, Del.

PASEK, Leonard E. (*M* 1944) Product Engr., • Kimberly-Clark Corp., Neenah, Wis., and 20 N. Broadway, White Plains, N. Y.

PASIADIS, Fedon H. (*J* 1945) Junior Project Engr., Engineering Laboratory, Airtemp Div., Chrysler Corp., Dayton 1, and • 618 Grafton Ave., Dayton 6, Ohio.

PASTEUR, Hugh W. (*M* 1943) Export Mgr., • J. E. Hall, Ltd., Dartford, and Fairseat House, Fairseat, Kent, England.

PASTOR, John C. (*M* 1938) Design Engr., • Rm. 818 Graham Bldg., Jacksonville 2, and E. 1091 Talbot Ave., Jacksonville 5, Fla.

PATERSON, Rollin O. (*A* 1945) Estimator, Fisher Body Div., General Motors Corp., Detroit, and • 169 Ardmore, Ferndale 20, Mich.

PATORNO, Sullivan A. S. (*M* 1923) Cons. Engr., • 101 Park Ave., New York, N. Y.

PATRICK, George P. (*J* 1945) Jr. Engr., • Carrier Corp., S. Geddes St., Syracuse, N. Y.

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PATTEN, J. Elverton (*M* 1945) Chief Engr., B. Offen & Co., 343 S. Dearborn St., Chicago 4, and 338 N. Stone Ave., La Grange, Ill.

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PATTERSON, G. P. (*M* 1939) Vice-Pres., • W. B. Haggerty, Inc., 805 Morgan St., P. O. Box 2971, Tampa 1, Fla.

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PATTERSON, James C. (*M* 1945) Dir. of Sales, Indus. Htg., Carrier Corp., and • 206 Robineau Rd., Syracuse, N. Y.

PATTERSON, Joseph C., Jr. (*M* 1944) Application Engr., • La-Del Conveyor & Manufacturing Co., 410 Washington Bldg., 15th and G, Washington 5, D. C., and 2414 Taylor Ave., Alexandria, Va.

PATTERSON, Ralph A. (*A* 1943) Gen. Sales Mgr., • Bell & Gossett Co., 8200 N. Austin Ave., Morton Grove, and 1213 Mulford St., Evanston, Ill.

PAUL, Dean W. (*A* 1944) N. Y. Mgr., Sterling Electric Motors, Inc., 41 Park Row, New York 7, and • 140 Cabrini Blvd., New York 33, N. Y.

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PAULDING, Lewis G. (*M* 1944) Supvsr., • Long Island Lighting Co., 250 Old Country Rd., Mineola, L. I., and Box 1033, Brightwaters, N. Y.

PAULEY, Robert D. (*A* 1944; *J* 1940) Development Engr., • Weyerhaeuser Timber Co., and 1503-25th Ave., Longview, Wash.

PAULSON, Owen D. (*A* 1945) Engr., General Air Conditioning & Htg. Co., 2001 Peralta St., Oakland, and • 1804 Nason St., Alameda, Calif.

PAVEY, Charles A. (*M* 1937) Dist. Mgr., • B. F. Sturtevant Co., 812 Michigan Bldg., Detroit 26, and 15778 Asbury Park, Detroit 27, Mich.

PAWKETT, Lawrence S. (*A* 1938) Mech. Engr., • L. S. Pawkett & Co., 810 Insurance Bldg., San Antonio 5, and 1007 W. Magnolia Ave., San Antonio 1, Texas.

PAYNE, Raymond L. (*A* 1944) Partner, • Air Flow Heating Co., 1409 Platte St., Denver 2, and 5706 W. Colfax, Denver 15, Colo.

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PEARSON, Fred L. (*M* 1942) Cons. Engr., • 1131 Majestic Bldg., Detroit, and 354 Glendale, Highland Park, Mich.

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ROSS, Otto C. (*M* 1944) Dist. Mgr., • American Air Filter Co., and Prop., Ross Power Equipment Co., 31 E. Georgia St., Indianapolis 4, and 615 East 44th St., Indianapolis 5, Ind.

ROSS, Roderick (*M* 1937) Principal, • Roderick Ross & Traill, 4 St. James' Bldg., 123 William St., Melbourne, C1, and 5 Burns St., Elwood, Melbourne S.3, Australia.

ROSS, Sidney S. (*A* 1945) Air Cond. Engr., Dwight D. Kimball, 1728 Grand Central Terminal Bldg., New York 17, and • 923 Walton Ave., New York 52, N. Y.

ROSSITER, I. J. (*A* 1939) Major, Air Corp. (0318742), Officers Mail Section, Lowry Field, Denver, and • 1546 Hudson St., Denver 7, Colo.

ROTH, Charles F. (*A* 1930) Pres., • International Exposition Co., Grand Central Palace, 480 Lexington Ave., New York 17, and 141 East 36th St., New York, N. Y.

ROTH, Harold R. (*M* 1935) (Council, 1945) Dist. Mgr., • Canadian Sirocco Co., Ltd., 57 Bloor St. W., and 5 Castleview, Toronto, Ont., Canada.

ROTH, Herman P. (*M* 1945) Design Engr., Air Research Manufacturing Co., 9851 Sepulveda Blvd., Los Angeles 45, and • 1906 Dufour Ave., Redondo Beach, Calif.

ROTHMAN, S. C.* (*M* 1936) Capt., U. S. Army, • Industrial Hygiene Engr., 2nd Service Command; 90 Church St., Room 1313, New York, N.Y.

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ROTTMAYER, Samuel I. (*A* 1933; *J* 1928) Mech. Engr., • 100 W. Monroe St., Chicago 3, and 8830 S. Laflin St., Chicago, Ill.

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ROWLEY, Frank B.* (*M* 1918) (*Presidential Member*) (Pres., 1932; 1st Vice-Pres., 1931; 2nd Vice-Pres., 1930; Council, 1927-33) Prof., University of Minnesota, and • Excelsior, Minn.

ROY, Ernest W. (*M* 1945) Pres., • Roy Equipment Co., Inc., 311 W. Church Ave., and 4638 Chambliss Ave., Knoxville, Tenn.

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ROY, Raul (*A* 1945) Sales Engr., • Sarco Canada, Ltd., 85 Richmond W., Room 326, and 105 Merrell E., Toronto, Ont., Canada.

ROYCE, Robert F. (*S* 1945) Student, University of Michigan, and • 328 E. Ann St., Ann Arbor, Mich.

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RUCHTE, Carl O. (*A* 1943) Gen. Mgr., • Home Insulation Co., 910 S. Saddle Creek Rd., Omaha 6, and 5035 Bedford Ave., Omaha 4, Nebr.

RUDD, Dann J. (*M* 1937) Asst. Mech. Engr., New York City Board of Education, 49 Flatbush Ave. Ext., Brooklyn, and • 580 Deer Park Ave., Babylon, L. I., N. Y.

RUDOLPH, R. R. (*M* 1943) Cons. Engr., • 723 Central Bldg., Assoc., General Engineering Co., • 725 Central Bldg., Seattle 4, and 6712–18th Ave. S.W., Seattle 6, Wash.

RUEMMELE, Albert M. (*A* 1943; *J* 1938) 1st Lt., • Ord. Dept., Hqs. P. R. Dept., O of Asst. C of S, G-4, San Juan, Puerto Rico, and Ashley, N. D.

RUFF, Adolph G. (*M* 1935) Supt. of Power, U. S. Playing Card Co., Park Ave., Norwood, and • 3824 Woodford Rd., Cincinnati, Ohio.

RUFF, Alonzo W. (*M* 1944) Vice-Pres., York-Shipley, Inc., Jessop Pl. & P. R. R., and 538 W. Springettsbury Ave., York, Pa.

RUFF, DeWitt C. (*M* 1922) Treas., • Healy-Ruff Co., 2255 University Ave., St. Paul 4, and 2211 St. Clair Ave., St. Paul 5, Minn.

RUGART, Karl (*A* 1924) Mfrs. Repr., • 26 South 20th St., Philadelphia 3, and 612 Bryn Mawr Ave., Penn Valley, Narberth P. O., Pa.

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RUMSEY, John L. (*M* 1941) Chief Engr., • Macnsons, 151 Tehama St., San Francisco 6, and Box 1025, Belvedere, Marin Co., Calif.

RUNGE, Edwin A. (*A* 1944) Chief Engr., Lee Engineering Co., Union National Bank Bldg., Youngstown, and • 429 Maplewood Ave., Struthers, Ohio.

RUNGE, Nathan P. (*M* 1945) Sales Engr., • Garden City Fan Co., 332 S. Michigan Ave., Chicago, and 346 Greenleaf Ave., Wilmette, Ill.

RUNNINGS, Henry M. (*M* 1943) Registered Engr., • 4729 Vermont Ave., Detroit 8, Mich.

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RUSSELL, James R. (*A* 1945) Chief Designer, Meadows, Chritoph & Misener, 62 Richmond St. W., and • 45 DeVere Gardens, Toronto, Ont., Canada.

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RUSSELL, W. A. (*M* 1943) Chief Engr. & Sales Mgr., Skinner Heating & Ventilating Co., Inc., 1948 N. Ninth St., St. Louis 6, and • 7918 Kingsbury Blvd., Clayton 5, Mo.

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RYAN, Joseph B. (*M* 1938) Engr., U. S. Engineering Co., 914 Campbell St., and • 3860 Charlotte Ave., Kansas City, Mo.

RYAN, William F. (*M* 1940; *A* 1939; *J* 1933) Chief Engr., The Salina Supply Co., 302-304 N. Santa Fe, and • 310 W. Republic, Salina, Kans.

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SAAR, Laird F. (*A* 1945) Owner, • Saar Heating & Engineering Co., 206½ W. Yakima Ave., Yakima, Wash.

SABIN, Edward R. (*M* 1919) Pres., • E. R. Sabin & Co., 4710 Market St., Philadelphia, Pa., and 205 Page Ave., Allenhurst, N. J.

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SACHS, Sam (*J* 1940) Mech. Engr., National Advisory Committee for Aeronautics, Bldg. 1229, Langley Field, and • 117 Barksdale Rd., Hampton, Va.

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SADLER, C. Boone (*M* 1928) Civil Engr., Public Works Office, 11th Naval Dist., San Diego, and • 4844 Del Mar Ave., San Diego 7, Calif.

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SALTER, Stanton W. (*M* 1942) Sales Engr., Engineering Equipment Co., Ltd., New Birk's Bldg., Rm. 910, Montreal, and • 153 Hillcrest Ave., Montreal W., Que., Canada.

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SANDERS, William L. (*A* 1945) Partner, • Midwest Plumbing & Heating Co., 6243 Prospect, Kansas City, and 103rd Blueridge, R. R. 1, Hickman Mills, Mo.

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SANFORD, Harold L. (*A* 1945) Co-Partner, • Sans Co., 57-59 Winsor St., and 69 Townline Rd., Jamestown, N. Y.

SANFORD, Sterling* (*M* 1930) Htg., Vtg. & Air Cond. Engr., • The Detroit Edison Co., 2000 Second Ave., Detroit 26, and 279 Hillcrest Rd., Grosse Pointe Farms 30, Mich.

SANGER, Ernest (*A* 1944) Mfrs. Agent, Watts Regulator Co., and • 1045 Wayburn Ave., Detroit 30, Mich.

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SAPP, Charles L. (*M* 1945; *A* 1936) Secy.-Mgr., Farquhar Furnace Co., 150 Owens Ave., and • 620 N. Walnut St., Wilmington, Ohio.

SATTERLEE, H. A. (*A* 1944; *J* 1940) Partner, • The Schulte Plbg. & Htg. Co., 526 Joplin St., and 1006 N. Sergt., Joplin, Mo.

SAUNDERS, Claude V. (*M* 1946) Mech. Engr., Louis C. Kingscott & Assocs., P. O. Box 671, Kalamazoo 99, and 906 Hazard Ave., Kalamazoo 19, Mich.

SAUNDERS, Douglas G. (*A* 1943) Estimator & Supvr., • John Plaxton Co., Ltd., 244 Main St., and 696 Warsaw Ave., Winnipeg, Man., Canada.

SAUNDERS, Lawrence P. (*M* 1933), (Treas., 1944-45; Council, 1941-45) Chief Engr., Research Engrg., • Harrison Radiator Div., General Motors Corp., and Tuscarora Club, Lockport, N. Y.

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SAURWEIN, Geo. K. (*M* 1938) Sales Engr., • Lombard Governor Corp., Ashland, and 639 Boston Post Rd., Weston 93, Mass.

SAVAGE, Sidney W. (*A* 1944) Engr., Gilson Manufacturing Co., Ltd., Guelph, and • 486 Clendenan Ave., Toronto, Ont., Canada.

SAVOIE, J. L. Roland (*A* 1944) • Codere, Ltd., 18 Wellington N., and 22-B Aberdeen, Sherbrooke, Que., Canada.

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SCHAARSCHMIDT, Karl F. (*A* 1945) Htg. Contr., • The Burns Heating Co., 1042 Holden Ave., Detroit 2, and 5121 W. Chicago Blvd., Detroit 4, Mich.

SCHAD, Clifford A. (*A* 1938; *J* 1937) Engr., United States Air Conditioning Corp., 2101 N.E. Kennedy St., and • 4425-43rd Ave. S., Minneapolis, Minn.

SCHAEFER, Armand (*A* 1945) Dist. Mgr., Iron Fireman Manufacturing Co., 429 S. Ashland Ave., and • 1429 Farragut, Chicago, Ill.

SCHAFER, Harry C. (*M* 1937) Sales Mgr., Iroquois Gas Corp., 45 Church St., Buffalo, and • 197 Union St., Hamburg, N. Y.

SCHAMPEL, Howard B. (*A* 1943) Design Engr., Minnesota Mining & Mfg. Co., 900 Fauquier Ave., and • 2122 Knapp St., St. Paul 8, Minn.

SCHARMER, George A. (*A* 1945) Special Supvsr., U. S. Navy Yard; New York, Bayonne Annex X-17, Bayonne, and ● 175 Main St., New Milford, N. J.

SCHAUER, Robert A. (*A* 1944; *J* 1942) Air Conditioning Engr., ● J. C. Penney Corp., Construction Dept., 330 West 34th St., New York, N. Y., and *2615 Indiana Way N.E., Canton, Ohio.*

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SCHERGER, Fred J. (*A* 1942) Sales Engr., ● Mundet Cork Corp., 335 W. Jefferson Ave., Detroit 26, and 3754 Kipling, Berkley, Mich.

SCHERNBECK, Fred H. (*A* 1930) Salesman, ● William Bros. Boiler & Mfg. Co., 1057–10th Ave. S.E., and 5045 Portland Ave., Minneapolis, Minn.

SCHICK, Karl W. (*A* 1946) Sales Mgr., Railway Controls Div., ● Minneapolis-Honeywell Regulator Co., 433 E. Erie, Chicago 11, and 5739 N. Francisco, Chicago 45, Ill.

SCHIWETZ, D. P. (*M* 1945) Branch Mgr., ● York Corp., Box 1007, Charlotte 1, and 825 Romany Rd., Charlotte 3, N. C.

SCHLENK, Hugo, Jr. (*M* 1945) Engr., ● Northwest Paper Co., and 214 Chestnut St., Cloquet, Minn.

SCHLICHTING, Walter G. (*M* 1932) Mgr., Air Conditioning Dept., Clarage Fan Co., 619 Porter St., Kalamazoo 16, and ● 224 S. Fletcher Ave., Kalamazoo 51, Mich.

SCHLICK, Paul F. (*A* 1940) Col., C. A. C., U. S. Army, ● Camp Haan, 216th Coast Artillery, San Francisco, Calif., and 1410 Edgcumbe Rd., St. Paul, Minn.

SCHMIDT, Alexander F. (*M* 1943) Designer, Air Cond., Engineering Dept., Ansco, Charles St., and ● 174 Crary Ave., Binghamton, N. Y.

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SCHMITT, William C. (*J* 1945; *S* 1944) Cpl., U. S. Army, Mech. Engr., Flesch & Schmitt, Inc., 118 Brown St., Rochester, and ● 590 Beach Ave., Rochester 12, N. Y.

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SCHOEN, Daniel D. (*J* 1942) Mech. Engr., Triangle Sheet Metal Works, 17 Meadow St., Brooklyn, and ● 272 S. Broadway, Yonkers 5, N. Y.

SCHOENIJAHN, Robert P. (*M* 1919) Owner, ● 305 Industrial Trust Bldg., Wilmington 7, and 719 Nottingham Rd., Wilmington 56, Del.

SCHOEPFLIN, P. H. (*M* 1920) Pres., ● Niagara Blower Co., 6 East 45th St., New York, and 9 Broadmoor Rd., Scarsdale, N. Y.

SCHOERNER, Rudolph T. (*M* 1943) Chief Engr., ● Taco Heaters, Inc., 123 South St., Providence, and 48 Fairway Dr., Oaklawn, R. I.

SCHOTT, Frederick C. (*A* 1945) Co-Owner, ● Atlas-Butler Furnaces, Inc., 243 N. Fifth St., Columbus 15, and 160 W. Royal Forest Blvd., Columbus 2, Ohio.

SCHREIBER, Herbert W., (*A* 1937) Branch Mgr. ● Johnson Service Co., 507 E. Michigan St., Milwaukee 2, and 6187 Washington Circle, Milwaukee 13, Wis.

SCHREIBER, W. F. (*M* 1945) Owner, ● Schreiber Inst. of Refrig., 902 Westport Rd., Kansas City 2, Mo., and 6200 Terry Dale Rd., Merriam, Kans.

SCHROEDER, Delman E. (*A* 1943) Surveyor to The American Bureau of Shipping, Rockefeller Bldg., Cleveland 13, and ● 1531 Duglas St., Ames, Iowa.

SCHROEDER, William R. (*A* 1939) Pvt., 345th Infantry, Camp McKain, Miss., and ● 201 Kedzie W., Evanston, Ill.

SCHROTH, August H. (*Life Member*; *M* 1911) N. Y. Mgr., Columbia Radiator Co., and ● 167 N. Grove St., East Orange, N. J.

SCHUBERT, Arno G. (*M* 1939) Asst. Prof., Mech. Engrg., Rensselaer Polytechnic Institute, Troy, and ● 1301 Broadway, Watervliet, N. Y.

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SCHULMEISTER, Walter A. (*M* 1944) Sales Mgr., American District Steam Co., North Tonawanda, and ● R. D. 1, Niagara Falls, N. Y.

SCHULTZ, Albert W. (*M* 1936) Grinnell Co., Inc., 240 Seventh Ave. S., Minneapolis 15, and ● 5204 France Ave. S., Minneapolis 10, Minn.

SCHULTZ, Stewart F. (*A* 1943) Engr., Htg. Vtg., Works Fisher Body Div., General Motors Corp., Engr. Dept., 11-135 G. M. Bldg., Detroit, and ● 16250 Cheyenne, Detroit 27, Mich.

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SHELDON, Nelson E. (*M* 1927) Branch Mgr., • Carrier Corp., 820 Reynolds Arcade Bldg., Rochester 4, and 41 Lanark Crescent, Rochester 9, N. Y.

SHELDON, Robert W. (*A* 1942) Heating Engr., • The Lake Shore Gas Co., Ashtabula, and 485 East 238th St., Euclid, Ohio.

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SIMPSON, Walter B. (*A* 1945) Div. Mgr., • A. M. Byers Co., 1270 Arcade Blvd., St. Louis 1, and 7747 Kingsbury, Clayton 5, Mo.

SIMPSON, William K. (*M* 1919) M. S. Little Mfg. Co., Hartford, and • 21 Sands St., Waterbury, Conn.

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SNOW, Edward E. (*M* 1945) Chief Designer & Engr., Mellor Bromley & Co., Ltd., St. Saviours Rd., and • 32 Spencefield Lane, Evington, Leicester, England.

SNYDER, Edwin F., Jr. (*A* 1946; *J* 1940) Htg. Engr., Minneapolis-Honeywell Regulator Co., Minneapolis, and • R. 3, Box 286-E, Excelsior, Minn.

SNYDER, Edwin J. (*J* 1944; *S* 1943) 1st. Lt., U. S. A., • 0553657, 3118th Engr., F. F. Plat., Fort Lewis, Wash., and 3903 Dolfield Ave., Baltimore, Md.

SNYDER, Jay W. (*M* 1917) Member of Firm, • Snyder & McLean, 2214 Penobscot Bldg., Detroit, and 15810 Asbury Park, Detroit 27, Mich.

SNYDER, L. R. (*A* 1944) Owner, • L. R. Snyder Co., 2625 University Blvd., Dallas 5, and 7034 Lakewood Blvd., Dallas 14, Texas.

SNYMAN, G. C. (*A* 1941) Mgr., Overseas Div., • Celotex Corp., 120 S. LaSalle St., Chicago 3, and Evanshire Hotel, Evanston, Ill.

SOBEL, Frank (*S* 1939) Pvt., • 813 M. P. Co., A. P. O. 928, San Francisco, Calif., and 115 Post Ave., New York, N. Y.

SOCKWELL, Charles, Jr. (*A* 1942) Partner, Sockwell Co., 156 Rogers St. N.E., and • 2562 Boulevard Dr. N.E., Atlanta, Ga.

SOCKWELL, Charles, Sr. (*M* 1942) Partner, • Sockwell Co., 156 Rogers St. N.E., Atlanta, and • 5447 Roswell Rd., Dunwoody, Ga.

SOCKWELL, Tyrus R. (*A* 1942) Partner, Sockwell Co., 156 Rogers St. N.E., and • 372 Morgan Pl. S.E., Atlanta, Ga.

SODEMANN, William C. B. (*Life Member; M* 1919) Pres., • Sodemann Heat & Power Co., 2306 Delmar Blvd., St. Louis, and 7542 Teasdale Ave., University City, Mo.

SOLSTAD, Lester L. (*A* 1945; *J* 1936) Engr., Austin Sheet Metal Works, 5109 W. Chicago Ave., Chicago 51, and • 1649 N. Oak Park Ave., Chicago 35, Ill.

SOMERS, William S. (*M* 1938) Vice-Pres.-Chief Engr., Lamneck Products, Inc., 1025 Lamneck St., and • 113 Kenwood Dr., Middletown, Ohio.

SOMMERFIELD, Sumner S. (*A* 1941; *J* 1936) • 4918 George St., Chicago 41, Ill.

SOMMERS, William J. (*M* 1937) Dist. Repr., • Ilg Electric Ventilating Co., 505 Delaware Ave., Buffalo 2, and 235 Hartford Ave., Kenmore 17, N. Y.

SOPER, Horace A. (*M* 1916) Pres., • American Foundry & Furnace Co., Washington at McClun, and 1122 E. Monroe St., Bloomington, Ill.

SORMANE, Walter (*A* 1944) Gen. Sales Mgr., Conco Engineering Works, and • 912-13th Ave., Mendota, Ill.

SOULE, John P. (*J* 1942) U. S. N. R., and • Gordon & Stewart Rds., Essex Fells, N. J.

SOULE, Lawrence C.* (*Life Member M* 1908) Secy. & Cons. Engr., Aerofin Corp., Syracuse, N.Y., and • Gordon & Stewart Rds., Essex Fells, N. J.

SPAAN, John H., Jr. (*M* 1945) Mgr., J. M. O'Connor Co., 1633 N.W. Fifth St., Oklahoma City, and • 2649 N.W. 11th St., Oklahoma City 7, Okla.

SPALL, Edward C. (*A* 1939) Vice-Pres., Perfex Controls, Ltd., Toronto, and • 291 Windermere Ave., Toronto 3, Ont., Canada.

SPARKS, James D. (*M* 1944; *A* 1937) Northwest Repr., • Ilg Electric Ventilating Co., 7331 W. Green Lake Way, Seattle 3, Wash.

SPECKMAN, Charles H. (*Life Member; M* 1918) Consulting Htg. & Vtg. Engr., • 382 Philadelphia Bourse Bldg., Philadelphia 6, and 1217 S. Fourth St., Philadelphia, Pa.

SPEER, Julian (*A* 1944) Mgr., • Julian Speer Co., 101 North High St., and 101 N. Ridge Rd., Columbus, Ohio.

SPELLER, Frank N.* (*Life Member; M* 1908) Metallurgical Consultant, • 6411 Darlington Rd., Pittsburgh 17, Pa.

SPENCE, Morton R. (*A* 1942; *J* 1934) Vice-Pres., • Rundle & Spence Mfg. Co., 445 N. Fourth St., and 709 E. Lexington Blvd., Milwaukee, Wis.

SPENCE, Robert A. (*A* 1946; *J* 1937) • 33 Barnard Rd., Belmont 78, Mass.

SPENCER, Charles H., (*M* 1944) Factory Agent and Sales & Service Engr., • 1421 S. Main St., Salt Lake City 4, Utah.

SPENCER, Roland M. (*M* 1945; *A* 1940; *J* 1938) Chicago Distr. Mgr., • The Powers Regulator Co., 2720 Greenview Ave., Chicago 14, and 7315 Lunt Ave., Chicago 31, Ill.

SPENCER, Warner E. (*A* 1938) Repr., • National Radiator Co., Inc., 220 Delaware Ave., Buffalo 2, and 212 Bidwell Pkwy., Buffalo, N. Y.

SPIELMANN, Gordon P. (*A* 1931; *J* 1923) Owner, • Harrison-Spielmann Co., 480 Milwaukee Ave., Chicago 10, and 730 N. Prospect Ave., Park Ridge, Ill.

SPIETH, Benjamin (*M* 1941) Chief Engr., • Modine Manufacturing Co., and 400 Harvey Dr., Racine, Wis.

SPITZ, Nathaniel (*J* 1945) Head of Sales & Installation, • City Coal Co., 410 Bank St., and 300 Ocean Ave., New London, Conn.

SPITZLEY, Joseph H. (*J* 1943) Vice-Pres., • R. L. Spitzley Heating Co., 1200 Fort St. W., Detroit 26, Mich.

SPITZLEY, R. L. (*M* 1920) Pres. & Gen. Mgr., • R. L. Spitzley Heating Co., 1200 W. Fort St., Detroit 26, and 26 Renaud Rd., Grosse Pointe Shores 30, Mich.

SPITZLEY, Ray L. (*A* 1944) Inspector of Refineries Class B, General Petroleum Corp., Torrance, and • 11542 Rochester, West Los Angeles 25, Calif.

SPOERR, Frank F. (*A* 1942; *J* 1937) • 140-19 Queens Blvd., Jamaica 2, L. I., N. Y.

SPOFFORTH, Walter (*M* 1930) Chief of Mech. Services, Federal Penitentiary, McNeil Island, and • 615 N. Ainsworth, Tacoma 6, Wash.

SPRIGGS, Carl U. (*A* 1945) Asst. Sales Mgr., Carrier Corp., Syracuse, and • 101 Hampshire Rd., Syracuse 3, N. Y.

SPRINGER, J. J. (*A* 1945) Owner, • Oneida Supply Co., 157-159 Cedar St., and 506 Broad St., Oneida, N. Y.

SPROTT, John I. (*A* 1941) Sales, • Ray Oil Burner Co., 401 Bernal Ave., and 70 Crestlake Dr., San Francisco, Calif.

SPROULL, Howard E. (*M* 1920) Div. Sales Mgr., • American Blower Co., 1005-6 American Bldg., Cincinnati 2, and 3588 Raymer Dr., Cincinnati 8, Ohio.

SPURGEON, Joseph H. (*M* 1924) Mfrs. Agent, • Spurgeon Co., 5050 Joy Rd., Detroit 4, and 17215 Pennington Dr., Detroit 21, Mich.

SPURLOCK, Benjamin H., Jr. (*M* 1944) Dept. of Mech. Engrg., University of Colorado, Boulder, Colo.

SPURNEY, Felix E. (*A* 1938) Engr., Luther and Wood, 1427 Eye St. N.W., Washington, D. C., and • 28 W. Baltimore St., Kensington, Md.

STACEY, Alfred E., Jr.* (*M* 1914) (Council, 1941-44) Capt., U. S. Navy, and • 35 Wootton Rd., Essex Fells, N. J.

STACK, Arthur E. (*A* 1935) Lt. Comdr., U. S. N. R., and • 7911 Chicago Ave., Silver Spring, Md.

STACY, Stanley C. (*M* 1931) Mech. Engr., • Board of Education, 13 S. Fitzhugh St., Rochester 4, and 111 Valley Rd., Rochester 10, N. Y.

STAFFORD, J. Fuller (*A* 1938) Owner, 519 N. Snelling Ave., St. Paul, and • 4545-18th Ave. S., Minneapolis, Minn.

STAFFORD, Thomas D. (*A* 1937) Vice-Pres.-Mgr., • Alexander-Stafford Corp., 1 Ionia Ave. N.W., and 954 Ogden Ave. S.E., Grand Rapids, Mich.

STAHL, Walter A. (*M* 1938) Gen. Maintenance Mgr., Montgomery Ward & Co., 619 W. Chicago Ave., Chicago, and • 2504 Harrison St., Evanston, Ill.

STAINS, W. A. (*M* 1945) Chief Engr., • Wurdeman & Becket, A. I. A., 3757 Wilshire Blvd., Room 201, Los Angeles 5, and 521 E. Valencia Ave., Burbank, Calif.

STALB, J. G. (*A* 1943) Reg. Mgr., Indus. Heating, Carrier Corp., 405 Lexington Ave., New York 17, and • R. F. D. No. 2, Ridgefield, Conn.

STAMBERGER, Robert F. (*A* 1944) Estimator & Supt., The Stamberger Co., 3115 Mayfield Rd., and • 2289 Lamberton Rd., Cleveland Heights, Ohio.

STAMMER, E. L. (*Life Member; M* 1919) Retired, • 7046 Winona, St. Louis 11, Mo.

STANDRING, Ronald A. (*A* 1942; *J* 1938) Htg. Engr., • Gurney Foundry Co., Ltd., P. O. Box 277, and 2358 Leclaire Ave., Maisonneuve, Montreal, Que., Canada.

STANGER, R. B. (*M* 1920) Prop., • Robinson & Stanger, Empire Bldg., Pittsburgh 22, and Middle Rd., Glenshaw, Pa.

STANGER, William F. (*M* 1945) Sales Engr., Robinson & Stanger, 1005 Empire Bldg., Pittsburgh 22, and • 337 S. Millvale Ave., Pittsburgh 24, Pa.

STANLEY, Robert L. (*M* 1938) Field and Research Engr., Naco Manufacturing Co., 7631 Roseberry Ave., Huntington Park, and • 3828 Udell Court St., Los Angeles 27, Calif.

STEARNS, Ellis J., Jr. (*M* 1944) Engr., National Bureau of Standards, Washington, D. C., and • 933 Parker St., Falls Church, Va.

STECKHAN, Louis (*M* 1941) Sales Engr., • Crane Co., 30 South 16th St., St. Louis 3, and 3240 Liberty St., St. Louis 11, Mo.

STEEL, R. Justin (*A* 1938) Lt., U. S. N. R., • Bureau of Ships, Navy Dept., Washington, D. C. and 346 S. College Ave., Newark, Del.

STEELE, J. B. (*M* 1932) Chief Operating Engr., Winnipeg School District No. 1, Ellen and William Ave., and • 184 Waterloo St., R.H., Winnipeg, Man., Canada.

STEEVES, Donald R. (*A* 1944) Mgr., Equipment Div., Farquhar Robertson, Ltd., 1124 Beaver Hall Hill, and • 5400 Queen Mary Rd., Apt. 29, Montreal, Que., Canada.

STEFFNER, Edward F. (*A* 1937; *J* 1934) Research Engr., Viking Air Conditioning Corp., 5601 Walworth, Cleveland, and • 1429 East 133rd St., E. Cleveland Ohio.

STEGGALL, Howard B. (*M* 1942; *A* 1934) United States Radiator Corp., 1500 United Artists Bldg., Detroit, and • Concord Rd., Bloomfield Hills, Mich.

STEIN, Herman W. (*M* 1944) • A. C. Nielsen Co., 2101 Howard St., Chicago, and 708 Roger Ave., Kenilworth, Ill.

STEIN, Jerome (*J* 1942; *S* 1940) Secy., Torrington Supply Co., Inc., 125 Maple St., and • 756 Waterville St., Waterbury, Conn.

STEINHORST, Theodore F. (*M* 1919) Pres., • Emil Steinhorst & Sons, Inc., 612-616 South St., Utica 3, N. Y.

STEINKE, Bernard J. (*J* 1940; *S* 1937) Mech. Engr., • U. S. Rubber Co., Maple St., Naugatuck, and Peach Orchard Rd., Waterbury 82, Conn.

STEINMETZ, C. W. A. (*M* 1934) Mgr., Newark Office, • American Blower Corp., 1060 Broad St., Newark 2, and 50 Oakwood Ave., Bogota, N. J.

STELL, William H., Jr. (*M* 1944) Owner, • Arnold R. Kamman Co., 493 Franklin St., Buffalo 2, and 648 Clay Ave., Rochester 13, N. Y.

STEMPEL, Edward H. (*M* 1942) Chief Design Engr., Young Radiator Co., and • 2321 Webster St., Racine, Wis.

STENCEL, Reinhold A. (*M* 1938) Chief Engr., • Canadian Ice Machine Co., Ltd., 65 Villiers St., and 224 Inglewood Dr., Toronto, Ont., Canada.

STENGEL, Frank J. (*A* 1935) Capt., • 1326 Engr. G. S. Regt., Camp Claiborne, La., and 39 Walnut Ave., Millburn, N. J.

STEPHAN, Charles O. (*A* 1945) Owner, Stephan & Lambert, 419 N. Santa Fe, and • 1925 Greenwood, Pueblo, Colo.

STEPHENSON, James R. (*A* 1940) • J. R. Stephenson & Co., 278 Main St., and 647 Strathcona St., Winnipeg, Man., Canada.

STEPHENSON, Kiser A. (*M* 1941) Pres., • Stephenson Co., 14 Harris St. N.W., Atlanta 3, and 923 St. Charles Ave. N.E., Atlanta, Ga.

STERMER, Clarence J. (*M* 1936) Engr., Crane Co., 836 S. Michigan Ave., and • 7839 Clyde Ave., Chicago, Ill.

STERN, Edward J. (*A* 1941) Dist. Mgr., • Armstrong Cork Co., 701 Burt Bldg., Dallas 1, and 223 E. Davis St., Dallas 8, Texas.

STERN, Lester M. (*J* 1945; *S* 1944) Ensign, U. S. N. R., • ASST. Engr. Officer, U.S.S.Sproston (DD 577), F. P. O., San Francisco, Calif.

STERNBERG, Edwin (*A* 1932; *J* 1931) Design Engr., Fred Moesel Assoc., 101 Park Ave., and • 16 East 98th St., New York, N. Y.

STERNE, C. M. (*M* 1943; *A* 1934) Lt. Comdr., U. S. N. R., Retired, and • 50-23 23rd St., Long Island City 1, N. Y.

STERNER, Douglas S. (*M* 1941; *A* 1940; *J* 1936) Major, U. S. Army, and • 1801 Pebrican St., Cheyenne, Wyo.

STERRY, H. Lee (*A* 1945) Dir. of Business Research, • Carrier Corp., and 321 Clairmonte Ave., Syracuse, N. Y.

STETSON, Lawrence R. (*M* 1913) Engr., • The McMurrer Co., 303 Congress St., Boston, and 35 Bradfield Ave., Roslindale, Mass.

STEVENS, Earl Knights (*A* 1940) • International Exposition Co., 480 Lexington Ave., New York 17, N. Y.

STEVENS, Harry L. (*M* 1934; *A* 1927; *J* 1924) Pres., • M. M. Stevens Co., 108-110 W. Sherman St., and 320 West 20th Ave., Hutchinson, Kan.

STEVENS, Howard R. (*M* 1941) Owner, Stevens Heating & Supply Co., 225 E. Fourth St., and • P. O. Box 182, Reno, Nev.

STEVENS, Joseph W. (*M* 1944) Sales Mgr., Cyclotherm Corp., Oswego, and • 303 Maple Rd., Syracuse 9, N. Y.

STEVENS, Judson E. (*A* 1941) Engr. in Charge, • National Coal Co., 318 Spokane St., P. O. Box 318, and 520 Vassar St., P. O. Box 2542, Reno, Nev.

STEVENS, Kenneth M. (*A* 1943; *J* 1936) Br. Mgr., The Powers Regulator Co., 409 East 13th St., Kansas City 6, and • 1214 East 82nd Terrace, Kansas City, Mo.

STEVENS, Stanley A. (*A* 1945) Engr., • Matthew Hall & Co., Ltd., 28 Dorset Sq., London N. W. 1, and 30A Falloden Way, Golders Green, London N. W. 11, England.

STEVENS, Wayne H. (*A* 1939) Engr., • Shellenberger, Gregg & Co., 2211 N. Prospect Ave., Milwaukee 2, and 2501 E. Stratford Ct., Milwaukee 11, Wisc.

STEVENS, William B. (*A* 1944) Htg. Engr., • Stevens Furnace Co., 79 S. LaSalle St., and 1219 New York St., Aurora, Ill.

STEVENS, William R. (*A* 1934) Partner, • L. E. Stevens Co., 626 Broadway, Cincinnati 2, Ohio, and 30 Chalfonte Ct., Ft. Thomas, Ky.

STEVENSON, Mel J. (*M* 1935) Cons. Mech. Engr., Pyle National Industrial Multivent Division, plus private practice, • Apt. 1004, 2970 Sheridan Rd., Chicago 14, Ill.

STEVENSON, W. W. (*M* 1928) Steam Htg. Engr., • Allegheny County Steam Heating Co., 435 Sixth Ave., Pittsburgh 19, and 1125 Lancaster Ave., Pittsburgh 18, Pa.

STEWART, Charles W. (*M* 1919; *A* 1918) Vice-Pres.-Sales, • Clark Mfg. Co., 1830 East 38th St., Cleveland, Ohio.

STEWART, Clement W. (*M* 1944) Marine Specialist, Ilg Electric Ventilating Co., 15 Park Row, 1108, New York 7, N. Y., and • 828 Fifth Ave., River Edge, N. J.

STEWART, Duncan J.* (*M* 1936; *A* 1930) Vice-Pres. & Gen. Mgr., • Barber-Colman Co., Drawer 99, and R. R. 4, Rockford, Ill.

STEWART, Iril A. (*A* 1945) Mech. Engr., Office of the Service Command Engineer, Hq. Ninth Service Command, Ft. Douglas, and • 1621 Princeton, Salt Lake City 5, Utah.

STEWART, James P. (*A* 1940; *J* 1937) Engr., Carrier Corp., 300 Geddes St., and • 224 Greenwood Pl., Syracuse 10, N. Y.

STEWART, John L., Jr. (*A* 1944) Gen. Mgr., • Calif-Fresno Air Conditioning Co., P. O. Box 527, and 1416 San Pablo, Fresno, Calif.

STEWART, John N. (*A* 1939) Plan Examiner, District of Columbia, 102 District Bldg., and • 6124-32nd Pl. N.W., Washington 15, D. C.

STEWART, Ralph M. (*A* 1945) Mgr., • Bryant Gas Heating Co., 3020 E. Franklin Ave., Minneapolis 6, and 26 West 22nd St., Minneapolis 4, Minn.

STEWART, Wesley O. (*M* 1944; *A* 1938) Branch Mgr., • Johnson Service Co., 153 West Ave., Los Angeles 31, and 4100½ Los Feliz Blvd., Los Angeles, Calif.

STICKLE, Fred A. (*A* 1945) Pres. & Gen. Mgr., Stickle Steam Specialties Co., 2215 Valley Ave., Indianapolis 1, and • 2339 Broadway St., Indianapolis 5, Ind.

STICKRATH, Kenneth J. (*A* 1945) Owner, • K. J. Stickrath, 316 Shaw Ave., and 2821 Capital St., McKeesport, Pa.

STILES, Gordon S. (*A* 1941; *J* 1936) Field Engr., Chrysler Airtemp Sales Corp., 1032 Irwin-Keasler Bldg., Dallas, and • Rt. 4, Box 140, Bryan, Texas.

STILLER, F. W. (*A* 1945) • 4501 S. Aldrich, Minneapolis 9, Minn.

STILPHEN, Norman E. (*M* 1945) Owner, • Stilphen Engineering Co., 26 Roberts St., and 253 Main St., Sanford, Maine.

STILWELL, R. F. (*M* 1944) Fuel Engr., • c/o North American Coal Corp. 1425 Terminal Tower, Cleveland 13, and 12500 Clifton Blvd., Lakewood, Ohio.

STIMSON, DePark (*M* 1945) Application Engr., The Bahnson Co., and • Idlewilde Dr., Winston-Salem, N. C.

STITES, Richard, Jr. (*A* 1943; *J* 1937) Sales Engr., • Coon Devisser Co., 2051 W. Lafayette, Detroit 16, and 17537 Hartwell, Detroit 21, Mich.

ST. LAURENT, Guy (*A* 1942) Htg. Contr., • Hector Groulx, Engrg., 1638 Notre Dame St. W., and 8381 Drolet St., Montreal, Que., Canada.

STOBBE, Gustav C. (*A* 1944) Gen. Supt., Becker-Seidel-Clark Co., 4500 Euclid Ave., Cleveland 3, and • 1805 Tampa Ave., Cleveland 9, Ohio.

STOCK, Charles S. (*M* 1936) Gen. Sales Mgr., The Herman Nelson Corp., 1824 Third Ave., Moline, Ill., and • R. R. 1, Bettendorf, Iowa.

STOCKWELL, William R. (*Life Member*; *M* 1903; *J* 1901) Gen. Mgr., Mfg. Div., • Weil-McLain Co., Michigan City, Ind.

STOFFER, Glen H. (*M* 1945) Branch Dealer Mgr., Carrier Corp., 542 Buhl Bldg., Detroit 26, and • 14384 Prevost Ave., Detroit 27, Mich.

STOKES, Alvin D. (*M* 1936) Riggs Distler & Co., Inc., 216 N. Calvert St., Baltimore, and • 424 Winston Ave., Baltimore 12, Md.

STOLKER, Edward A. (*A* 1945) Owner, • Stolker Engineering Co., 4201 Brown St., Philadelphia 4, and 2601 Parkway 30, Philadelphia, Pa.

STONE, A. A. (*A* 1943) Branch Mgr., • Mundet Cork Corp., 505 Southland Life Annex, Dallas 1, and 5307 Monticello Ave., Dallas 6, Texas.

STONER, William C. (*A* 1944) • Moore-Stoner Co., 500 Stewart Ave. S.W., Atlanta, Ga.

STOREY, J. H. (*M* 1944) Pres., • John Tweddle, Ltd., 80 Jean Talon St. W., Montreal 14, and 5201 Decarie Blvd., Montreal, P. Q., Canada.

STORMS, Robert M. (*M* 1936) Htg., Plbg., Vent., Storms & Lowe, 6359 Yucca St., Los Angeles 28, and • 354 W. Wilson Ave., Glendale 3, Calif.

STOTESBURY, Bernard (*M* 1942) Partner, • Ottawa Plumbing & Heating Contractors, 111 Third Ave., and 118 Gilmour St., Ottawa, Ont., Canada.

STOTT, Douglas A. (*A* 1940) Vice-Pres., Mgr., • Canadian Powers Regulator Co., Ltd., 195 Spadina Ave., Toronto 2-B, and 30 MacNaughton Rd., Toronto 12, Leaside, Ont., Canada.

STOTT, F. W. (*M* 1938) Branch Sales Mgr., C. A. Dunham Co., Ltd., 1139 Bay St., Toronto, and • Thomas St., Oakville, Ont., Canada.

STOUT, Arthur G. (*M* 1943) Asst. Chief Engr., Holabird & Root, 333 N. Michigan Ave., Chicago, and • 105 East Ave., Park Ridge, Ill.

STOVER, Rolland S. (*A* 1944) Owner, • Engineering Equipment Co., P. O. Box 354, and 207 N. Eighth St., Marshalltown, Iowa.

STRACHAN, George W. (*M* 1945) Project Engr., Dravo Corp., 300 Penn Ave., and • 1848 Hillsdale Ave., Pittsburgh 16, Pa.

STRAIN, A. James (*A* 1942) Gen. Mgr., • Ruud Manufacturing Co., 474-476 Bathurst St., Toronto 2-B, and 376 St. Clements Ave., Toronto 12, Ont., Canada.

STRAND, Charles A. (*A* 1940) Supt., • Bruce Wigle Plumbing Co., 9117 Hamilton Ave., Detroit 2, and 6533 Barlum Ave., Detroit 10, Mich.

STRAUCH, Paul C. (*M* 1945; *A* 1934) Mgr., Htg. Div., • Sears-Piou & Co., 814 S. Vandeventer, St. Louis 10, and 7309-F Burrwood Dr., Normandy 21, Mo.

STREATER, Edward C. (*A* 1939) Mgr., • E. C. Streater Co., Spring Park, and Mound, Minn.

STREATER, Walter A. (*M* 1945) Dist., Repr., • Modine Manufacturing Co., 152 Nassau St. N.W., Atlanta 3, and 151 Coventry Rd., Decatur, Ga.

STREVELL, R. P. (*M* 1934) Pres.-Treas., • The William R. Hogg Co., Inc., 900 Fourth Ave., Asbury Park, and Victor Place and State Highway, Neptune, N. J.

STROCK, Clifford (*M* 1937; *A* 1929) Editor, • Heating & Ventilating, 148 Lafayette St., New York 13, and Box 756, Amityville, L. I., N. Y.

STROMGREN, Sven G. (*M* 1938) Managing Dir., • Magnussons Mekaniska Verstad, Regeringsgatan 109, Stockholm, Sweden.

STRONG, Henry G. (*M* 1945) Dir. of Sales, Transportation, Air Cond. & Refrig., • Carrier Corp., Syracuse, and 133 Edwards Dr., Fayetteville, N. Y.

STRONG, Stewart S. (*A* 1945) Sales Engr., • Parsons Engineering Corp., Box 226, Cuyahoga Falls, and 2916 Lee Rd., Silver Lake, Ohio.

STROTHER, William E. (*M* 1944; *A* 1941) Major, Corps of Engineers, Post Engineer, Hq. 4th Serv. Command, and • 997 Burns Dr., S.W., Atlanta, Ga.

STROUSE, Bernard H. (*M* 1945) Owner, • S. B. & B. H. Strouse, 307 Guarantee Trust Bldg., and 129 N. Bartram Ave., Atlantic City, N. J.

STROUSE, Sherman W. (*A* 1938) • Trane Co., 493 Franklin St., Buffalo 2, and 95 Mayville Ave., Tonawanda, N. Y.

STRUNIN, Jay (*A* 1939; *J* 1933) Owner, • Jay Strunin Plbg. & Htg. Contr., 408 Second Ave., New York 10, and 217 Ocean Ave., Brooklyn, N. Y.

STUART, Milton C.* (*M* 1935) Prof. of Mech. Engrg., • Lehigh University, Bethlehem, Pa.

STUART, Ralph A. (*M* 1943) Cons. Engr., Opera House Bldg., and • 432 North 6¼ St., Terre Haute, Ind.

STUART, W. W. (*A* 1940) Owner, • Stuart Supply, 417 Ninth St., Des Moines 9, and 1920 Pleasant St., Des Moines, Iowa.

STURM, William (*A* 1944; *J* 1940) Mech. Engr., • Ellerbe & Co., 1021-E First National Bank Bldg., St. Paul 1, and 60 Inner Dr., St. Paul 5, Minn.

SUDDERTH, Leo, Jr. (*A* 1942; *J* 1936) Partner, • Garrard, Sudderth & Seat, 313 Bona Allen Bldg., Atlanta 3, and 3047 Piedmont Rd. N.E., Atlanta, Ga.

SULLIVAN, John R. (*A* 1944) Foreman, U. S. Naval Air Material Centre, Navy Yard, Philadelphia, and • 31-B Stephen Court, Milbourne, Pa.

SULLIVAN, Thomas J. (*J* 1943; *S* 1942) Lt. (j.g.), U. S. N. R., Administrative Aide to Shop Supt., • U. S. Navy Yard, and 48 Monument Walk, Apt. 1C, Brooklyn 1, N. Y.

SULLIVAN, T. J. (*M* 1940) Pres., Sullivan Valve & Engineering Co., 910 S. Arizona St., and • 1205 W. Park St., Butte, Mont.

SULLIVAN, William H., Jr. (*M* 1944) Vice-Pres., • W. H. Sullivan Co., Inc., P. O. Box 232, and 308 W. Greenway South, Greensboro, N. C.

SULLIVAN, William H. (*M* 1943) Pres., • W. H. Sullivan Co., Inc., P. O. Box 232, and 200 E. Greenway North, Greensboro, N. C.

SUMMERVILLE, E. A. (*A* 1944) Pres., • Bixley Inc., 610 Walnut Bldg., and 5701 Waterbury Circle, Des Moines 12, Iowa.

SUNDERLAND, Richard P. (*A* 1938) Partner, • General Meters & Controls Co., 205 W. Wacker Dr., Chicago, and 1250 Cherry St., Winnetka, Ill.

SUPPLE, Graeme B. (*M* 1934) Indiana Dist. Mgr., • American Blower Corp., 625 Architects & Builders Bldg., and 420 East 58th St., Indianapolis, Ind.

SUTCH, Harry C. (*A* 1940) Capt., • Umatilla Ordnance Depot, Ordnance, Ore.

SUTCLIFFE, A. G. (*M* 1922; *A* 1918) Chief Engr., Ilg Electric Ventilating Co., 2850 N. Crawford Ave., Chicago, and • 432 S. Delphia Ave., Park Ridge, Ill.

SUTFIN, George V. (*M* 1942; *A* 1937) Branch Mgr., • American Blower Corp., 1005-6 American Bldg., Cincinnati 2, and 3270 Hildreth Ave., Cincinnati 11, Ohio.

SUTTER, E. E. (*A* 1936) Sales Engr., Mueller Brass Co., Port Huron, Mich., and • 6705 Sixth St. N.W., Washington, D. C.

SWAIN, Douglas S. (*A* 1945; *J* 1941) Sales Engr., Trane Company of Canada, Ltd., 365 Hargrave St., and • 1186 Downing St., Winnipeg, Man., Canada.

SWAIN, Wilbur A. (*A* 1944) Sales Engr., Jenkins Bros., 80 White St., New York 13, and • 90 Evergreen Pl., East Orange, N. J.

SWAIN, William L. (*M* 1939) Dir., • Messrs. Young, Austen & Young, Ltd., 19 Buckingham St., London W.C. 2, and 35 Uphill Rd., Mill Hill, London N. W. 7, England.

SWALLOW, J. H. (*A* 1945) Branch Mgr., St. Louis Factory, • Bryant Heater Co., 494 Arcade Bldg., St. Louis 1, and 845 N. Kirkwood Rd., Kirkwood 22, Mo.

SWANEY, Carroll R. (*M* 1929; *J* 1921) Mfrs. Agent, • C. R. Swaney Co., 335 Newbury St., Boston 16, and 61 Morse Rd., Newtonville, Mass.

SWANSON, Earl C. (*A* 1935) Vice-Pres., • Andersen Corp., Bayport, Minn.

SWANSON, Nils W. (*A* 1936) Sales Engr., McDonnell & Miller, 400 N. Michigan Ave., Chicago 11, and • 2746 Morse Ave., Chicago 45, Ill.

SWART, Harvey G. (*A* 1944) Salesman & Engr., • The Trane Co., 2326 S. Michigan Ave., Chicago 16, and 525 Fairview Ave., Park Ridge, Ill.

SWASKI, Kenneth J. (*A* 1945) MMR 1/c, Mat. & Eng. Bldg. 142, Box 10, N. O. B., Norfolk, Va., and • Rt. 1, Sweet Home, Ore.

SWATS, William F. (*A* 1945) Asst. Supt. of Bldgs., • Vanderbilt University, Dept. of Bldgs., Nashville 4, and 3304 Wimbleton Rd., Nashville 5, Tenn.

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WOOD, Alfred W. (*A* 1941; *J* 1938) F/Lt., R.C.A.F., Overseas, Station Armament Officer, Linton-on-Ouse, Yorkshire, and • 36 Dufferin St., Ingersoll, Ontario, Canada.

WOOD, Charles F. (*M* 1937) Air Cond. Mgr., Prod. Development & Application Dept., Frigidaire Div., General Motors Sales, Corp., 300 Taylor St., Dayton 1, and • R. R. 1, Spring Valley, Ohio.

WOODALL, Raymond G. (*A* 1945) Air Cond. Dept., General Electric Co., Red Rock Bldg., Atlanta, and • 1324 East 37th St., Savannah, Ga.

WOODBRIDGE, William L. (*A* 1944) Owner, • Woodbridge Plumbing & Heating, R. F. D. 3, 610 Farr St., Milford, Mich.

WOODGER, Herbert W. (*M* 1939) Htg. & Vtg. Engr., • General Electric Co., 100 Woodlawn Ave., Pittsfield, and "Pineacres," East St., Lenox, Mass.

WOODHOUSE, Graham D. (*A* 1938) Vice-Pres., Dowagiac Steel Furnace Co., and • 204 Indiana Ave., Dowagiac, Mich.

WOODMAN, Lawrence E. (*M* 1934) Owner, • Woodman Engineering Co., 203 E. Capitol, and 925 Adams, Jefferson City, Mo.

WOODRUFF, Henry S. (*M* 1945) Product Engr., Carrier Corp., Syracuse, and • 24 Fenner St., Cazenovia, N. Y.

WOODS, Baldwin M., Dr. (*M* 1937), (Council, 1942-45) Dir. of University Extension, University of California, and • 249 The Uplands, Berkeley 5, Calif.

WOODSON, Edward G. (*M* 1944) Mgr., Memphis Dist., • The Trane Co., 705 Columbian Tower, Memphis 3, and 80 Lombardy Rd., Memphis, Tenn.

WOODWARD, Kenneth C. (*M* 1944) Engr., Dvlpt. Dept., Chase Brass & Copper Co., Inc., Waterbury, and • 6 South Court, Meriden, Conn.

WOOLLARD, Mason S. (*M* 1934) Registered Engr., Harry H. Angus, 1221 Bay St., Toronto 1, and • 31 Hillcrest Park Ave., Toronto 5, Ont., Canada.

WOOLCOCK, Edwin (*A* 1938) Owner, • Woolcock Plumbing & Heating Co., 2217–15th St., and 440 Memorial Pkway., Niagara Falls, N. Y.

WOOLSTON, Robert H. (*A* 1944; *J* 1941) Partner, • Woolston Woods Co., 2132 Cherry St., and 12 English Village, Wynnewood, Pa.

WOOTEN, M. Frank, Jr. (*M* 1941; *A* 1940) Cons. Engr., • Wooten & Wooten, 215 Latta Arcade, and 400 Cherokee Rd., Charlotte, N. C.

WORKMAN, Albert E. (*A* 1941) Htg. & Air Cond. Sales Dir., • United Gas Corp., P. O. Box 2628, Houston 1, and 1658 W. Main, Apt. 7, Houston, Texas.

WORMLEY, Robert F. (*A* 1938) Mgr., Engineered Products Div., • Grinnell Company of Canada, Ltd., 2440 Dundas St. W., and 1 L'Estrange Place, Toronto 9, Ont., Canada.

WORSHAM, Herman (*M* 1925; *J* 1918) Mgr., War Products Training Service, • Frigidaire Div., General Motors Corp., 300 Taylor St., Dayton 1, and 524 Daytona Pkwy., Dayton 6, Ohio.

WORTHINGTON, James M. (*A* 1944) Application Engr., Fulton Sylphon Co., 410 N. Michigan Ave., Room 560, Chicago, and • 3334 Church St., Evanston, Ill.

WORTHINGTON, Thomas H. (*M* 1937) Asst. Mgr., Eastern Sales, • Standard Sanitary & Dominion Radiator, Ltd., 405 Beaubien St. W., Montreal, Que., Canada.

WORTON, William (*M* 1937) Mgr., • C. A. Dunham Co., Ltd., 504 Scott Block, and 292 Lansdowne Ave., Winnipeg, Man., Canada.

WRIGHT, C. P. (*A* 1944) Mgr., • C. P. Wright & Co., 4300 Carnegie Ave., Cleveland 3, and 2987 Torrington Rd., Shaker Heights 22, Ohio.

WRIGHT, Calvert C. (*M* 1944) Prof. of Fuel Technology, • The Pennsylvania State College, School of Mineral Industries, and 314 Arbor Way, State College, Pa.

WRIGHT, Clarence E. (*A* 1940; *J* 1935; *S* 1933) Mgr., Htg. Dept., Fairmont Wall Plaster Co., Tenth St., and • 303 Nuzum Pl., Fairmont, W. Va.

WRIGHT, Harris H. (*M* 1917) Owner, H. H. Wright Co., 1322 Walnut St., Kansas City 6, and • 808 Greenway Terrace, Kansas City 5, Mo.

WRIGHT, Jack (*A* 1944) Owner, • Canadian Plumbing & Heating Specialties, Ltd., 701 Craig W., Montreal, Que., and 24 Thurlow, Hampstead, Canada.

WRIGHT, John B. (*M* 1940) Sales Engr., • The Nash Engineering Co., South Norwalk, and Rowayton, Conn.

WRIGHT, John F. (*A* 1945) Vent. & Methods Engr., • Servel, Inc., 119 Morton Ave., and 414 Stanley Ave., Evansville, Ind.

WRIGHT, K. A. (*M* 1921) Branch Mgr., • Johnson Service Co., 1905 Dunlap St., Cincinnati 14, Ohio, and 113 Orchard Rd., Ft. Mitchell, Covington, Ky.

WRIGHT, Lawrence T., Jr.* (*J* 1943) Asst. Prof. of Heat-Power Engrg., • Cornell University, College of Engineering, Ithaca, N. Y.

WRIGHT, Norman S., Jr. (*A* 1942; *J* 1941) Mfrs. Repr., • 250 Perry St., San Francisco, and 1015 Mariposa Ave., Berkeley 11, Calif.

WRIGHTSON, Wilbor T. (*M* 1937) Eastern Mgr., • Garden City Fan Co., 55 West 42nd St., New York 18, and 22 Sagamore Rd., Bronxville 8, N. Y.

WUNDERLICH, Milton S.* (*M* 1925) Dir. of Research, Insulite Div., Minnesota & Ontario Paper Co., 500 Baker Arcade, Minneapolis, and • 545 Mt. Curve Blvd., St. Paul 5, Minn.

WYATT, DeWitt H. (*M* 1936) Cons. Mech. Engr., • 123 Acton Rd., Columbus 2, Ohio, and 2004 West End Ave., Nashville 4, Tenn.

WYLD, Reginald G. (*M* 1937) • 24 Swan St., Rochester, N. Y.

WYLIE, Wilson J. (*M* 1945) Asst. to Mgr., Power Dept., • Toronto Hydro Electric System, 14 Carlton St., Toronto 2, and 459 Briar Hill Ave., Toronto 12, Ont., Canada.

Y

YAGER, John J. (*M* 1921) Pres., Goergen-Mackwirth Co., Inc., 817 Sycamore St., Buffalo, and • 425 Woodbridge Ave., Buffalo 14, N. Y.

YAGLOU, C. P.* (*M* 1923) Assoc. Prof., Indus. Hygiene, • Harvard School of Public Health, 55 Shattuck St., Boston 15, and 10 Vernon Rd., Waverly 79, Mass.

YATES, Joseph E. (*M* 1939) Safety Supvsr., • Pacific Power & Light Co., 405 Public Service Bldg., Portland 4, Ore.

YATES, Robert A. (*A* 1945) Mgr., • Yates & Co., 231 Tenth St., and 431–16th St., Brandon, Man., Canada.

YATES, Walter (*Life Member*; *M* 1902) Governing Dir., • Matthews & Yates, Ltd., Cyclone Works, Swinton, and 4 Egerton Park, Worsley, Man., England.

YEAGER, George F. (*M* 1944) Lead Inspector Air Cond. Dvlpt., Pennsylvania Railroad Test Dept., • 1221 Fourth Ave., Altoona, Pa.

YEAZEL, Harold F. (*A* 1945) Asst. Sales Mgr., • The Lennox Furnace Co., Inc., 400 N. Midler Ave., Syracuse 1, and 1411 Oak St., Syracuse 8, N. Y.

YELLEN, Charles F. (*A* 1945) Htg., Vtg., Air Cond. Maintenance Engr., Pratt & Whitney Aircraft Corp., Div. of United Aircraft, 400 S. Main St., East Hartford 8, and • 61 Imlay St., Hartford 5, Conn.

YEOMANS, Paul H. (*M* 1944) • 56 Cedarbrook Rd., Ardmore, Pa.

YERKES, William L. (*M* 1941) Br. Mgr., Direct Sales Div., Carrier Corp., 1500 S. Santa Fe Ave., Los Angeles 21, Calif.

YOST, Claude (*M* 1944) Mech. Engr. Designer & Draftsman, • Zumwalt & Vinther, 1807 Mercantile Bldg., Dallas 1, and 5502 Goodwin Ave., Dallas 6, Texas.

YOUNG, Chester C. (*M* 1943) Utilization Engr., • Lone Star Gas Co.—Dallas Div., of Distribution, 301 S. Harwood St., Dallas 1, and 4336 Stanhope St., Dallas 5, Texas.

YOUNG, Emil O. (*A* 1935) Owner, • Young Regulator Co., 5209 Euclid Ave., Cleveland 3, and 3628 Cummings Rd., Cleveland Heights, Ohio.

YOUNG, George Harrison (*J* 1944) Chief Engr., V. E. Sprouse Co., and • 2225 California St., Columbus, Ind.

YOUNG, Harold J. (*M* 1937) Sales Engr., Young Radiator Co., Occidental Hotel Bldg., and • 1364 Lakeshore Dr., Muskegon, Mich.

YOUNG, J. T., Jr. (*A* 1936) Mgr., • Crane Co., Box 709, and 538 Ogden Canyon, Ogden, Utah.

YOUNGBLOOD, Clete W. (*A* 1944) Owner, • Youngblood Plumbing & Heating Supply Co., 1700 Broadway, and 707 North 23rd St., Paducah, Ky.

YOUNGER, John R. (*J* 1941) M. M. 1/C, U. S. N. R., • Boat Pool 15, Navy 3205, F. P. O., San Francisco, Calif.

YOVANOVICH, Emil (*A* 1945) Expediter, Carnegie Illinois Steel Corp., Gary Works, and • 4204 Conn St., Gary, Ind.

YULA, Ralph W. (*A* 1944) Yula Water Heaters, Inc., 166 West 225th St., New York 63, N. Y.

Z

ZACK, H. J. (*M* 1928) Prop., The Zack Co., 2311 Van Buren St., Chicago 12, Ill.

ZAHNER, Leo W., Sr. (*A* 1944) Owner, • A. Zahner & Co., 3041-43 Wyandotte, Kansas City 8, and 4434 Madison, Kansas City, Mo.

ZAKI, Hussein M. (*A* 1945; *J* 1941; *S* 1940) Chief Engr., Refrigeration Dept., The Tractor and Engineering Co., P. O. B. 366, and • 5 Ismail Anwar St., Dokki, Cairo, Egypt.

ZANONE, Henry A. (*A* 1943) Htg. & Vent. Engr., Raymond M. Meyer, 1701 Rowan St., Louisville, and • Rt. 1, Box 441, Shively, Ky.

ZAVORSKI, Leonard (*M* 1944) Vice-Pres. in Charge of Production, Tuttle & Bailey, Inc., and • 6 Lake Court, New Britain, Conn.

ZELLAK, Joseph (*A* 1944) Engr. & Estimator, Sauer Co., Inc., 368 N. Washington Ave., Box 4657, Fort Hays Station, and • 1928 Baldridge Rd., Columbus, Ohio.

ZIBOLD, Carl E. (*M* 1929) Mech. Engr., Htg. & Vtg., War Dept., Service Command Engrs., 270 Broadway, New York, and • 13 Chadwick Rd., White Plains, N. Y.

ZIEBER, W. E. (*M* 1935) Dir. of Research, • York Corp., and 22 S. Keesey St., York, Pa.

ZIEGLER, Donald D. (*J* 1944; *S* 1942) Lt., • Propeller Laboratory, Wright Field, Dayton, Ohio.

ZIEL, Herbert E.* (*M* 1941) Mech. Engr., • Albert Kahn, Assoc., 345 New Center Bldg., and 694 Glynn Court, Detroit 2, Mich.

ZIEMBA, Walter P. (*A* 1945) Draftsman, • N. B. Hubbard, 220 Bagley, Detroit 26, and 19732 Kenosha, Detroit 24, Mich.

ZIESSE, Karl L. (*A* 1931) Partner, • Phoenix Sprinkler & Heating Co., 115 Campau Ave. N.W., Grand Rapids, Mich.

ZIGHEIM, William C. (*M* 1943) Owner, • William C. Zingheim Co., 450 E. Ohio St., and 8541 Drexel Ave., Chicago, Ill.

ZINK, David D. (*M* 1931) • 33 Calle de Varsailles, Mexico City, D.F., Mexico.

ZINTEL, George V. (*A* 1941) Lt., U. S. N. R., Navy Dept., 141 W. Jackson Blvd., Rm. 1500, Chicago 4, and • 1428 Summerdale, Chicago 40, Ill.

ZUBER, Otto C. (*A* 1938) Chief Engr., • Amana Society, Amana, and South Amana, Iowa.

ZUMBEHL, Lambert E. (*A* 1945) Sales Engr., • Forslund Pump & Machinery Co., 1717-19 Main St., and 1304 East 32nd St. Terrace, Kansas City, Mo.

ZUMWALT, Ross (*A* 1941; *J* 1938) Partner, • Zumwalt & Vinther, 1807 Mercantile Bank Bldg., Dallas, Texas.

ZUROW, William A. (*A* 1946; *J* 1937) Lt., U. S. N. R., • 1517 S.W. 61 Dr., Portland 1, Ore.

ZWERLING, Seymour J. (*J* 1944) Engr. & Supt., • Triangle Sheet Metal Works, Inc., 17-19 Meadow St., Brooklyn 6, and 4146 Bedford Ave., Brooklyn 29, N. Y.

ZYNDA, John R. (*J* 1942) Ensign, U. S. N. R., U. S. S. Conquest (AMc-71), U. S. Naval Frontier Base, Charleston, S.C., and • 8876 Burt Rd., Detroit 23, Mich.

ENGINEERS OF HUMAN COMFORT

The Heating, Ventilating and Air Conditioning Engineer through his work and research brings to our homes, our offices and our factories, in both summer and winter, that climate best suited to our comfort and health, and he creates for our industries those atmospheric conditions which improve quality of product and speed production.

He is truly an *Engineer of Human Comfort* and a public benefactor.

It was in 1894 that a small group of nationally known engineers, educators, and manufacturers organized the A.S.H.V.E. for the express purpose of improving the Art through the interchange of ideas and stimulation of scientific research and invention.

In 1894, it was agreed that the great art of heating and ventilating deserved and required recognition as an essential, distinctive and highly specialized division of modern engineering. These men realized the basic importance of heating and ventilating as the primary element in the well-being of civilized mankind, living and working mostly indoors.

These keen, alert, progressive men knew that the methods and equipment of their day could be improved even beyond their own vision if all the personalities striving for such improvement could be welded into one organized cooperative group, imbued with the same ideals and set toward the same goal. So they formed themselves into the nucleus of such an organization and called it THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

That the Charter Members had great faith in their enterprise is evident, although little did they dream that progress would be so rapid in their profession.

During the intervening years since that little group of 75 pioneers unfurled the banner of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—4506 of the real leaders of thought and action in heating, ventilating, and air conditioning have gathered about that standard and carried it proudly before them far along the way of real accomplishment. They may be identified among engineering groups by the distinctive emblem which was adopted by the Charter Members.

The record of the Society's accomplishments is a long and brilliant one, yet it must embrace but a small part of the real achievement which the Society has inspired both among its members and the many non-members whose work it has influenced. Rewarding the efforts of its founders, who gave unstintingly of their time and means, the Society has been tremendously effective in the advancement of the Art and Industry through its manifold activities.

FOUNDING OF THE A. S. H. V. E.

The organization of a Society of heating and ventilating engineers was first talked of by Hugh J. Barron and L. H. Hart in the early summer of 1894.

On August 2, 1894, a group of 15 men met at the office of *Heating and Ventilation*, 146 World Bldg., New York, and a committee of 5 was chosen to effect an organization.

The first meeting was called at 3:00 P.M., September 10, 1894, at the Broadway Central Hotel and temporary officers were elected: F. P. Smith, *Chairman*; and L. H. Hart, *Clerk*. The roll call showed that 75 persons had become Charter Members. It was voted that the name of the organization should be THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and a Constitution and By-Laws were adopted. The following were elected to hold office until the 1st Annual Meeting, January 22-24, 1895: *President*, E. P. Bates, Syracuse, N. Y.; *First Vice-President*, W. M. Mackay, New York, N. Y.; *Second Vice-President*, W. F. Wolfe, Boston, Mass.; *Third Vice-President*, Chas. S. Onderdonk, Philadelphia, Pa.; *Treasurer*, Judson A. Goodrich, New York, N. Y.; *Secretary*, L. H. Hart, New York, N. Y.; *Board of Managers*, E. P. Smith, H. J. Barron, A. A. Cary, New York; James A. Harding, Vineyard Haven, Mass.; Henry Adams, Washington, D. C.; *Council on Membership*, Chas. W. Newton, Baltimore, Md.; R. C. Carpenter, Ithaca, N. Y.; Albert A. Cryer, New York; F. W. Foster, Boston; U. G. Scollay, Brooklyn, N. Y.

HOW TO APPLY FOR MEMBERSHIP

The real accomplishments of life are usually measured by the service one has rendered to his fellows and the true cultural refinement of mind, the finest sense of personal and professional ethics, factors transcending all material elements in what man calls "success," are developed through association with those of high ideals and cherished ambitions in the same field of activity. THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS offers to him whose work is definitely within its province an opportunity for such association and an opportunity for real service to his profession.

Every man in the heating, ventilating and air conditioning profession needs the Society—

1—Because of the contacts that it brings through national and local meetings.

2—Because of the information supplied by Society Publications.

3—Because of the opportunities that research reveals in new applications for engineering services and equipment.

4—Because of the satisfaction to be derived in contributing to human comfort and well being.

A Candidate must make application on the printed form "Membership Application" which is available at the headquarters office or from Chapter Officers and members. A statement of qualifications and engineering experience is required and four members must act as sponsors except under certain conditions noted in *Article B*-III of the By-Laws.

Initiation Fees are: Members and Associate Members $10.00; Junior Members $5.00. The Initiation Fee must accompany application.

The annual dues of the Society are: Members and Associate Members $18.00; Junior Members $10.00; and Student Members $3.00. Dues of new members are pro-rated on a quarterly basis.

ARTICLE C-II—MEMBERSHIP.

Section 1. Persons connected with the arts and sciences related to heating, ventilating or air conditioning are eligible for admission into the Society.

Section 4. A Member shall be over thirty (30) years of age and shall have more than eight (8) years' experience in the sciences relating to the arts of heating, ventilating or air conditioning. He shall have been in active practice of his profession and in responsible charge of important work for four (4) years, consisting of design, construction, research, development or teaching, and shall be qualified to design or direct such engineering work.

Section 5. A Junior Member shall be a person over twenty (20) years and under thirty (30) years of age, who has had three (3) years' experience in the sciences relating to the arts of heating, ventilating or air conditioning. Each successfully completed year in an engineering school may be considered equivalent to one (1) year of such work.

Section 6. An Associate Member shall be twenty-five (25) years of age or over. He need not be an engineer, but must have been so connected with some branch of engineering or the art of heating, ventilating, air conditioning or the industries relating thereto, that he may be considered as qualified to co-operate with heating and ventilating engineers in the advancement of professional knowledge.

Section 7. A Student Member shall be a person between the ages of 18 and 25 years, who is regularly attending courses in an engineering college or technical school at the time of applying for membership.

PRESIDENTS OF THE A. S. H. V. E.

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

CHARTER MEMBERS OF THE SOCIETY

Henry Adams
Washington, D. C.
Homer Addams
Washington, D. C.
Newell P. Andrus
Brooklyn, N. Y.
Hugh J. Barron
New York, N. Y.
Thomas Barwick
New York, N. Y.
Edward P. Bates
Syracuse, N. Y.
Geo. C. Blackmore
Pittsburgh, Pa.
J. J. Blackmore
New York, N. Y.
L. R. Blackmore
New York, N. Y.
Samuel Burns
New York, N. Y.
B. Harold Carpenter
Wilkes Barre, Pa.
R. C. Carpenter
Ithaca, N. Y.
Albert A. Cary
New York, N. Y.
Robert C. Clarkson
Philadelphia, Pa.
Geo. B. Cobb
New York, N. Y.
H. D. Crane
Cincinnati, Ohio
Albert A. Cryer
New York, N. Y.
T. B. Cryer
Newark, N. J.
Mark Dean
Boston, Mass.
John Demarest
Boston, Mass.
Thos. J. Douglass
Norwich, Conn.
A. C. Edgar
Philadelphia, Pa.
Hermann Eisert
Baltimore, Md.
John A. Fish
Boston, Mass.
Frank W. Foster
Boston, Mass.
C. F. Gessert
New York, N. Y.
Judson A. Goodrich
New York, N. Y.
John Gormly
Philadelphia, Pa.
James A. Harding
New York, N. Y.
L. H. Hart
New York, N. Y.
Charles F. Hauss
New York, N. Y.
Jno. D. Hibbard
Chicago, Ill.
William H. Hill
New York, N. Y.
Geo. D. Hoffman
Chicago, Ill.
Charles S. Hopkins
Rochester, N. Y.
Alfred A. Hunting
Boston, Mass.
Stewart A. Jellett
Philadelphia, Pa.
J. H. Kinealy
St. Louis, Mo.
Jos. A. Langdon
Pittsburgh, Pa.
Chas. W. Light
Saginaw, Mich.
H. E. Light
Saginaw, Mich.
Chas. C. Lincoln
New York, N. Y.
C. K. Longenecker
New York, N. Y.
James Mackay
Chicago, Ill.
Wm. M. Mackay
New York, N. Y.
A. S. Mappett
New York, N. Y.
Wm. McMannis
New York, N. Y.
George L. Mehring
Chicago, Ill.
Edward A. Munro
Brooklyn, N. Y.
Robert Munro
Pittsburgh, Pa.
Charles W. Newton
Baltimore, Md.
Theodore C. Northcott
Elmira, N. Y.
Charles S. Onderdonk
Philadelphia, Pa.
John A. Payne
Providence, R. I.
John H. Petherick
Chattanooga, Tenn.
Geo. W. Plastow
Jersey City, N. J.
Henry B. Prather
Buffalo, N. Y.
Leon H. Prentice
Chicago, Ill.
D. M. Quay
Chicago, Ill.
Wm. A. Russell
New York, N. Y.
U. G. Scollay
Brooklyn, N. Y.
Percival H. Seward
New York, N. Y.
Le Roy B. Sherman
New York, N. Y.
Fred P. Smith
New York, N. Y.
B. F. Stangland
New York, N. Y.
Geo. P. Steel
Philadelphia, Pa.
Joseph M. Stoughton
Yonkers, N. Y.
H. M. Swetland
New York, N. Y.
Geo. H. Underhill
Boston, Mass.
T. J. Waters
Chicago, Ill.
J. R. Wendover
New York, N. Y.
W. B. Wilkinson
New York, N. Y.
James R. Willett
Chicago, Ill.
J. J. Wilson
Troy, N. Y.
Wiltsie F. Wolfe
Boston, Mass.

A.S.H.V.E. MILESTONES

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.

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 A.S.H.V.E. Guide published in September.

1923 Comfort Zone Established by A.S.H.V.E. 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.

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 W. H. 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.

1943 Membership at all-time high, 3300—Chapter with largest Charter Membership organized in Indianapolis.

1944 50th Anniversary.

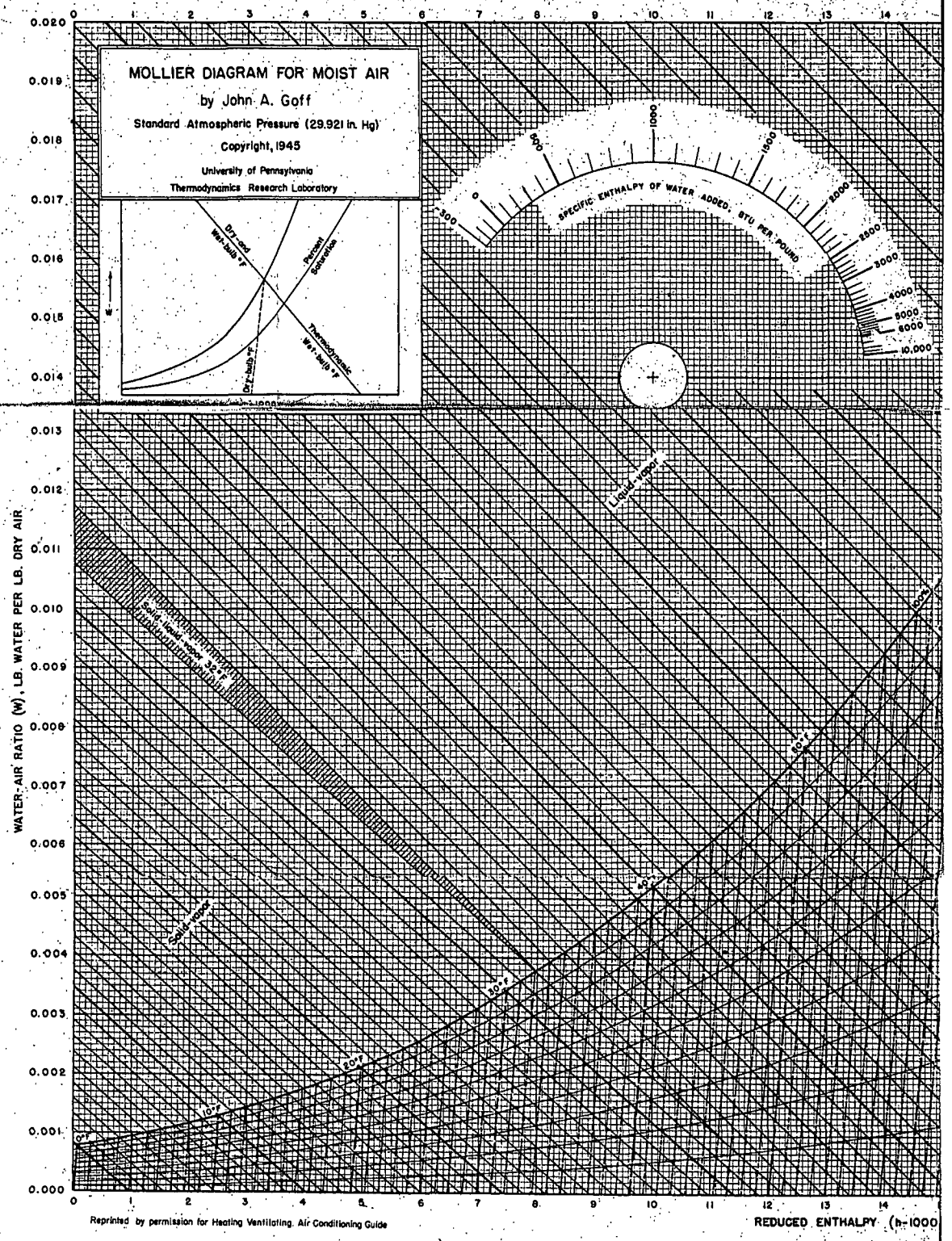
1945 Members serving in Armed Forces—416.

1946 New Charter approved.

Meetings of
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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.
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	Buffalo, N. Y.
				June 8-10	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
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	Mackinac Isl., Mich.
				*Oct. 30-31	Atlanta, Ga.
1940	46th	Jan. 23-26	Cleveland, Ohio	June 17-19	Washington, D. C.
				*Oct. 14-15	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.		
1946	52nd	Jan. 27-30	New York, N. Y.		

*Fall Meeting



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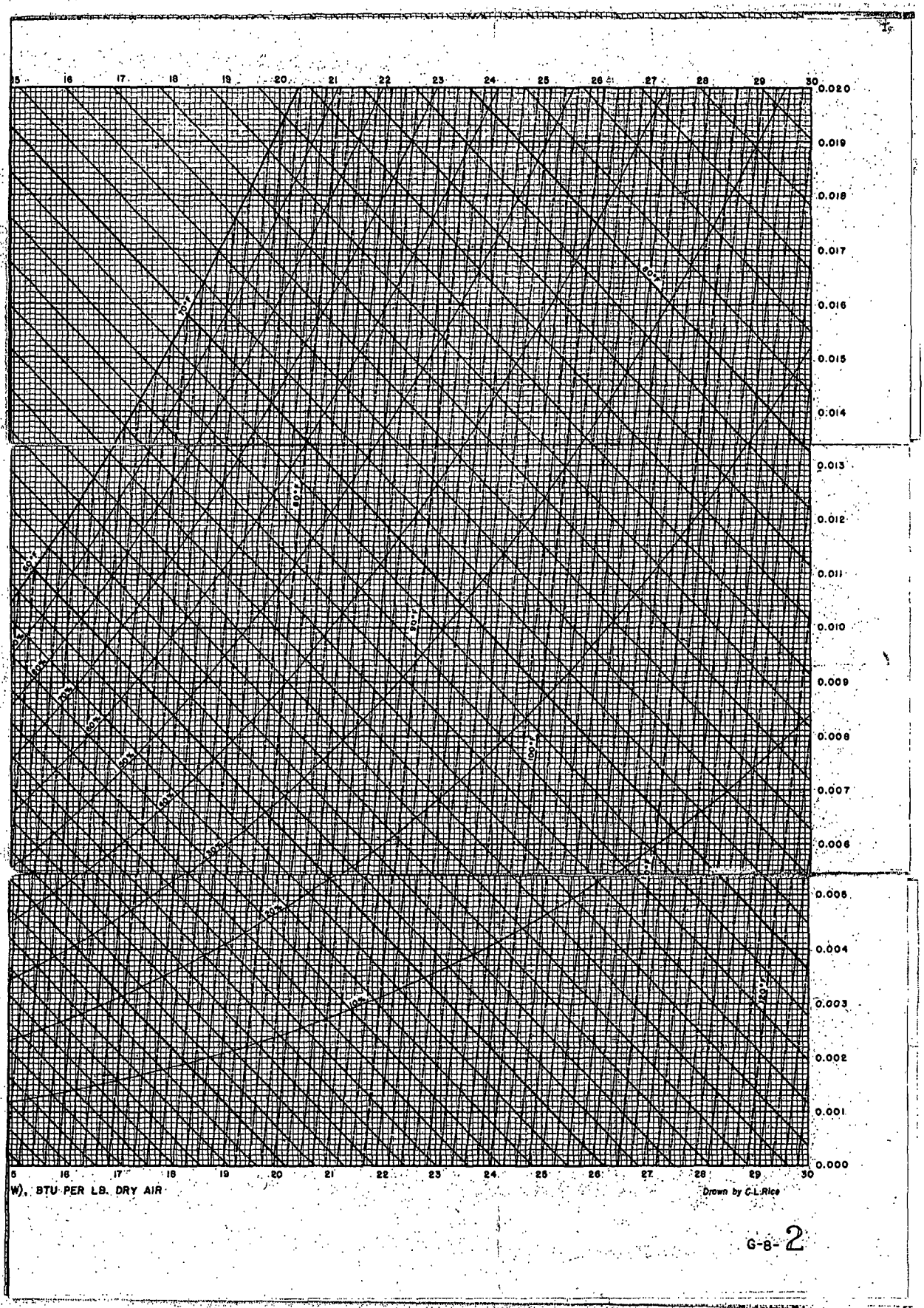
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Drawn by C.L.Rice

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