

THE
AMERICAN SOCIETY OF HEATING
and VENTILATING ENGINEERS
GUIDE
1936
HEATING VENTILATING AIR CONDITIONING

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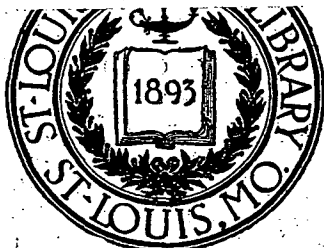
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**THE
AMERICAN SOCIETY of HEATING
and VENTILATING ENGINEERS
GUIDE 1936**

FOR
HEATING, VENTILATING, AIR CONDITIONING

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 TRANS-
ACTIONS—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. 14

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PREFACE TO THE 14th EDITION

*Amer Soc of Heating
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Vol. 3, 2nd Ed. 1936-1937*

THE Guide Publication Committee has incorporated in the fourteenth edition of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE many new and important additions to the Technical Data Section which extends its usefulness to all those individuals who are interested in the professional art of heating, ventilating and air conditioning.

The original conception of the founders of THE GUIDE has been carefully protected in this new volume so as to maintain its continued role as an authoritative and unbiased presentation of long established, as well as newly recognized scientific standards of the profession and allied industries.

Because of increasing demands from various users of THE GUIDE, and because of increased application of air conditioning and cooling to all types of industrial processing and comfort installations, four new chapters have been added to the Technical Data Section on the subjects of Refrigeration, Drying, Motors and Their Control and Railway Air Conditioning. All of the principle thermodynamic cycles of cooling adaptable to air conditioning are concisely described and illustrated in the new chapter on Refrigeration and in addition complete tables of the common refrigerants are included. Dryer design and methods of construction are given with proper control mechanism for adequate and desirable drying results. A new temperature and humidity chart which may be applied to all types of dryer calculations with a complete descriptive example is fully described. Electric motors and their proper adaptation to all types of applications for fans, filters, pumps, hoists, compressors, stokers, oil burners and other units used in heating, ventilating and air conditioning installations are discussed in the chapter on Motors with recommendations for control devices accomplishing convenient and effective regulation. All of the several complicated features involved in the design of passenger car railway air conditioning are amply discussed in a new chapter on this subject. A typical example is given involving the many factors which should be considered in the calculation of the cooling load for a passenger car. Curves and tables showing the costs per car mile for various types of cooling systems are given with cost equations which take into account the various mechanical efficiencies applicable to tractive power.

In addition to this new material all of the previous chapters have been carefully checked and reviewed and in some cases the text has been completely rewritten. Information on unit heaters, ventilators, coolers and air conditioners has been correlated in one chapter and completely revised. Two methods of estimating fuel consumption are discussed in the new chapter on Heat and Fuel Utilization. System characteristics of fans with curves to illustrate the effects of speed or system changes on fan operation have been added. Other chapters which have been amplified with much new material are: Cooling Methods, Industrial Exhaust Systems, Automatic Fuel Burning Equipment, Radiators and Gravity Convectors, Pipe, Fittings, Water Supply Piping, Water Heating, Electrical Heating, and Air Conditioning for Industrial Processes.

An additional feature of THE GUIDE 1936 is the new table of saturated water vapor extending over a range from minus 130 to plus 200 F with heat content values based on the varying values of specific heat for air. As a result of a comprehensive survey of cities in the United States and Canada, a table has been added giving the maximum city water main temperatures. The degree-day table has been amplified so as to segregate the monthly values for each city.

Basic and fundamental data which have been added to THE GUIDE each year since its inception in 1922 has been retained with additions from research sources made necessary by the progress of the heating and ventilating art. The Problems in Practice which were a major innovation of the last edition of THE GUIDE have been retained with new and practical examples to expand and amplify the text matter of each one of the 44 chapters. Over 60 new illustrations, curves and charts have been added which enhances the value of the descriptive text material.

The increasing demand for copies of this book by universities, engineering and technical schools for student instruction indicates its recognition as the standard authority in the field of heating, ventilating and air conditioning.

In connection with the manufacturers' Catalog Data Section the Committee has made a more determined effort to carry out the original conception of having the descriptive material as free as possible from comparative statements and superlative terms, and to present therein such condensed technical information concerning modern equipment as will be of practical value to the user of THE GUIDE. In general, the manufacturers have cooperated fully to make their data useful, informative and serviceable, so that the user can apply it effectively in the selection of materials or equipment to be used in general design. The Committee appreciates the aid of those far-seeing manufacturers who have assisted in this cooperative enterprise to advance the art of heating, ventilating and air conditioning which so vitally affects the comfort, happiness and health of everyone. In 1936 it will be possible to distribute THE GUIDE to more than 12,000 users and it is extremely gratifying that a greater number of manufacturers recognize the effective service rendered by this reference volume and have increased their advertising space in order to carry the story of their products in greater detail to the thousands who use this book.

It is felt that a careful study of both text and catalog data pages will amply reward the user who desires to obtain reliable data in convenient form and it is hoped that the brevity and simplicity of the presentation will impress the user who requires a volume for quick reference.

The Committee releases this fourteenth edition of over 12,000 copies with the sincere hope that it will receive the same generous reception that was accorded to its predecessors and that users will find it the most valuable reference volume in their library.

G. L. Larson, Chairman
GUIDE PUBLICATION COMMITTEE

EDITORIAL ACKNOWLEDGMENT

SINCE 1922 THE GUIDE has held an enviable record as an authoritative reference volume for the heating, ventilating and air conditioning profession and its allied industries. Each year the Guide Publication Committee invites and receives the helpful assistance of hundreds of engineers and technicians who have in the past given willingly of their time and knowledge to improve the advancement of the profession.

It is with a profound feeling of appreciation that this Guide Publication Committee gratefully acknowledges the cooperative efforts of the following individuals who have contributed generously to the improvement of this 1936 edition.

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Those individuals who constantly use THE GUIDE and members of the Society are especially indebted to those engineers who have aided in the preparation of this edition and the Guide Publication Committee hereby expresses its sincere appreciation for the loyal cooperation of the several contributors to this volume.

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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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Chapter 1

FUNDAMENTALS OF HEATING AND AIR CONDITIONING

Dalton's Law, Dry- and Wet-Bulb Temperatures, Properties of Air, Humidity, Relative Humidity, Specific Humidity, Relation of Dew Point to Relative Humidity, Adiabatic Saturation of Air, Total Heat and Heat Content, Enthalpy, Psychrometric Chart, Properties of Steam, Properties of Water, Rate of Evaporation

AIR conditioning has for its objective the supplying and maintaining, in a room or other enclosure, of an atmosphere having a composition, temperature, humidity, and motion which will produce desired effects upon the occupants of the room or upon materials stored or handled in it.

Dry air is a mechanical mixture of gases composed, in percentage of volume, as follows¹: nitrogen 78.03, oxygen 20.99, argon 0.94, carbon dioxide 0.03, and small amounts of hydrogen and other gases.

Atmospheric air at sea level is given in percentage by volume as: N₂ 77.08, O₂ 20.75, water vapor 1.2, A 0.93, CO₂ 0.03 and H₂ 0.01. The amount of water vapor varies greatly under different conditions and is frequently one of the most important constituents since it affects bodily comfort and greatly affects all kinds of hygroscopic materials.

LAW OF PARTIAL PRESSURES

A mixture of dry gases and water vapor, such as atmospheric air, obeys Dalton's Law of Partial Pressures: each gas or vapor in a mixture, at a given temperature, contributes to the observed pressure the same amount that it would have exerted by itself at the same temperature had no other gas or vapor been present. If p = the observed pressure of the mixture and p_1, p_2, p_3 , etc. = the pressure of the gases or vapors corresponding to the observed temperature, then

$$p = p_1 + p_2 + p_3 \text{, etc.} \quad (1)$$

DRY- AND WET-BULB TEMPERATURES

Air is said to be saturated at a given temperature when the water vapor mixed with the air is in the dry saturated condition or, what is the equivalent, when the space occupied by the mixture holds the maximum possible weight of water vapor at that temperature. If the water vapor mixed with the dry air is superheated, i.e., if its temperature is above the temperature of saturation for the actual water vapor partial pressure, the air is not saturated.

¹International Critical Tables.

The starting point of most applications of thermodynamic principles to air-conditioning problems is the experimental determination of the dry-bulb and wet-bulb temperatures, and sometimes the barometric pressure.

The *dry-bulb temperature* of the air is the temperature indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air. The *wet-bulb temperature* is determined by a thermometer with its bulb encased in a fine mesh fabric bag moistened with clean water and whirled through the air until the thermometer assumes a steady temperature. This steady temperature is the result of a dynamic equilibrium between the rate at which heat is transferred from the air to the water on the bulb and the rate at which this heat is utilized in evaporating moisture from the bulb. The rate at which heat is transferred from the air to the water is substantially proportional to the wet-bulb depression ($t - t'$), while the rate of heat utilization in evaporation is proportional to the difference between the saturation pressure of the water at the wet-bulb temperature and the actual partial pressure of the water vapor in the air ($e' - e$). Carrier's equation for this dynamic equilibrium is

$$\frac{e' - e}{t - t'} = \frac{B - e'}{2800 - 1.3t'} \quad (2a)$$

In the form commonly used,

$$e = e' - \frac{(B - e')(t - t')}{2800 - 1.3t'} \quad (2b)$$

where

- e = actual partial pressure of water vapor in the air, inches of mercury.
- e' = saturation pressure at wet-bulb temperature, inches of mercury.
- B = barometric pressure, inches of mercury.
- t = dry-bulb temperature, degrees Fahrenheit.
- t' = wet-bulb temperature, degrees Fahrenheit.

Formula 2b may be used to determine the actual partial pressure of the water vapor in a dry air-water vapor mixture. Then, from Dalton's Law of Partial Pressures, Equation 1, it follows that the partial pressure of the dry air is ($B - e$).

If a mixture of dry air and water vapor, initially unsaturated, be cooled at constant pressure, the temperature at which condensation of the water vapor begins is called the *dew-point temperature*. Clearly the dew-point is the saturation temperature corresponding to the actual partial pressure, e , of the water vapor in the mixture.

PROPERTIES OF AIR

Density is variously defined as the mass per unit of volume, the weight per unit of volume, or the ratio of the mass, or weight, of a given volume of a substance to the mass, or weight, of an equal volume of some other substance such as water or air under standard conditions of temperature and pressure. The term *specific gravity* is more commonly used to express the latter relation but, when the gram is taken as the unit of mass and the cubic centimeter as the unit of volume, density and specific gravity have

the same meaning. The term *specific density* is sometimes used to distinguish the weight in pounds per cubic foot; and as here used, *density* is the weight in pounds of one cubic foot of a substance.

The density of air decreases with increase in temperature when under constant pressure. The density of dry air at 70 F and under standard atmospheric pressure (29.92 in. of Hg) is approximately 0.075 lb (see Table 1), while that of a mixture of air and saturated water vapor at the same temperature and barometric pressure is only about 0.0743 lb. In the mixture the density of the dry air is 0.0731 and that of the vapor is 0.00115 lb (see Table 2).

In order to make comparisons of air volumes or velocities it is necessary to reduce the observations to a common pressure and temperature basis. The basic pressure is usually taken as 29.92 in. of Hg, but no basic temperature is universally recognized. Common temperatures for this purpose are 32 F, 60 F, 68 F, and 70 F. Since 70 F is the most commonly specified temperature to which rooms for human occupancy must be heated, it is usually understood, when no other temperature is specified, that 70 F is the basic temperature for measuring the volume or the velocity of air in heating and ventilating work.

The *specific volume* of air is the volume in cubic feet occupied by one pound of the air. Under constant pressure the specific volume varies inversely as the density and directly as the absolute temperature.

The *specific heat of air* is the number of Btu required to raise the temperature of 1 lb of air 1 F. The specific heat at constant pressure, C_p , and that at constant volume, C_v , are different. The specific heat at constant pressure is commonly used and it varies, under a pressure of one atmosphere, from a minimum at about 32 F from which it *increases* with either increase or decrease of temperature. The value 0.24 is sufficiently accurate for use at ordinary temperatures, but the values range¹ from 0.2399 at 32 F to 0.2404 at 212 F, 0.2413 at 392 F, 0.243 at -108 F, and 0.252 at -301 F.

The *mean specific heat of water vapor* at constant pressure is taken as 0.45 for all general engineering computations.

Table 3 is intended to aid in determining the density of moist air, taking into account its temperature, pressure, and moisture content.

Example 1. To show the use of Table 3: Given air at 83 F dry-bulb and 68 F wet-bulb (or a depression of 15 deg) with a barometric pressure of 29.40 in. of mercury. What will be the weight of this air in pounds per cubic foot?

Solution. From Table 3 the weight of saturated air at 80 F and 29.00 in. barometer is found to be 0.07034 lb per cubic foot. There is a decrease of 0.00015 lb per degree dry-bulb temperature above 80 F. There is an increase of 0.00025 lb for each 0.1 in. above 29.00 in. From the last column of Table 3 it is found that there is an increase of approximately 0.000035 lb per degree wet-bulb depression when the dry-bulb is 83 F. Tabulating the items:

- 0.07034 = weight of saturated air at 80 F and 29.00 bar.
- 0.00045 = decrement for 3 deg dry-bulb, 3×0.00015 .
- + 0.00100 = increment for 0.4 in. bar., 4×0.00025 .
- + 0.00053 = increment for 15 deg wet-bulb depression, 15×0.000035 .

0.07142 = weight in pounds per cubic foot of air at 83 F dry-bulb, 68 F wet-bulb, 29.40 in. bar.

TABLE 1. PROPERTIES OF DRY AIR^a
Barometric Pressure 29.921 In.

TEMPERATURE Deg F	WEIGHT PER CU FT POUNDS	PER CENT OF VOLUME AT 70 F	BTU ABSORBED BY ONE CU FT DRY AIR PER DEG F	CU FT DRY AIR WARMED ONE DEGREE PER BTU
0	0.08636	0.8680	0.02080	48.08
10	0.08453	0.8867	0.02039	49.05
20	0.08276	0.9057	0.01998	50.05
30	0.08107	0.9246	0.01957	51.10
40	0.07945	0.9434	0.01919	52.11
50	0.07788	0.9624	0.01881	53.17
60	0.07640	0.9811	0.01846	54.18
70	0.07495	1.0000	0.01812	55.19
80	0.07356	1.0190	0.01779	56.21
90	0.07222	1.0380	0.01747	57.25
100	0.07093	1.0570	0.01716	58.28
110	0.06968	1.0756	0.01687	59.28
120	0.06848	1.0945	0.01659	60.28
130	0.06732	1.1133	0.01631	61.32
140	0.06620	1.1320	0.01605	62.31
150	0.06510	1.1512	0.01578	63.37
160	0.06406	1.1700	0.01554	64.35
180	0.06205	1.2080	0.01506	66.40
200	0.06018	1.2455	0.01462	68.41
220	0.05840	1.2833	0.01419	70.48
240	0.05673	1.3212	0.01380	72.46
260	0.05516	1.3590	0.01343	74.46
280	0.05367	1.3967	0.01308	76.46
300	0.05225	1.4345	0.01274	78.50
350	0.04903	1.5288	0.01197	83.55
400	0.04618	1.6230	0.01130	88.50
450	0.04368	1.7177	0.01070	93.46
500	0.04138	1.8113	0.01018	98.24
550	0.03932	1.9060	0.00967	103.42
600	0.03746	2.0010	0.00923	108.35
700	0.03423	2.1900	0.00847	118.07
800	0.03151	2.3785	0.00782	127.88
900	0.02920	2.5670	0.00728	137.37
1000	0.02720	2.7560	0.00680	147.07

^aFrom Fan Engineering.

It is usual to assume that dry air, moist air, and the water vapor in the air follow the laws of perfect gases. This assumption while not absolutely true, especially with saturated vapor at temperatures much above 140 F,

TABLE 2. PROPERTIES OF SATURATED AIR^a

Weights of Air, Vapor of Water, and Saturated Mixture of Air and Vapor at 29.921 Inches of Mercury

TEMP. Deg F	WEIGHT IN A CUBIC FOOT OF MIXTURE			BTU ABSORBED BY ONE CUBIC FOOT SAT. AIR PER DEG F	CUBIC FEET SAT. AIR WARMED ONE DEGREE PER BTU	SPECIFIC HEAT BTU PER POUND OF MIXTURE
	WEIGHT OF DRY AIR POUNDS	WEIGHT OF VAPOR POUNDS	TOTAL WEIGHT OF THE MIXTURE POUNDS			
0	0.08625	0.000068	0.08632	0.02083	48.02	0.2413
10	0.08433	0.000110	0.08444	0.02039	49.05	0.2415
20	0.08246	0.000176	0.08264	0.01998	50.07	0.2418
30	0.08062	0.000277	0.08090	0.01958	51.07	0.2420
40	0.07878	0.000409	0.07919	0.01921	52.06	0.2426
50	0.07694	0.000587	0.07753	0.01885	53.05	0.2431
60	0.07506	0.000828	0.07589	0.01851	54.02	0.2439
70	0.07310	0.001151	0.07425	0.01819	54.97	0.2450
80	0.07103	0.001578	0.07261	0.01790	55.87	0.2465
90	0.06879	0.002134	0.07092	0.01762	56.76	0.2485
100	0.06635	0.002850	0.06920	0.01736	57.59	0.2509
110	0.06364	0.003762	0.06740	0.01714	58.35	0.2543
120	0.06060	0.004914	0.06551	0.01695	59.00	0.2587
130	0.05715	0.006351	0.06350	0.01679	59.56	0.2644
140	0.05319	0.008120	0.06131	0.01668	59.96	0.2721
150	0.04864	0.010295	0.05894	0.01662	60.17	0.2820
160	0.04340	0.012936	0.05634	0.01662	60.17	0.2950
170	0.03734	0.016108	0.05345	0.01668	59.96	0.3121
180	0.03035	0.019896	0.05025	0.01684	59.38	0.3351
190	0.02228	0.024400	0.04668	0.01710	58.49	0.3663
200	0.01300	0.029715	0.04272	0.01749	57.18	0.4094
210	0.00230	0.035938	0.03824	0.01802	55.50	0.4712
212	0.00000	0.037307	0.03731	0.01815	55.10	0.4865

^aFrom Fan Engineering.

is sufficiently accurate for practical purposes and it greatly simplifies computations.

Boyle's Law refers to the relation between the pressure and volume of a gas, and may be stated as follows: *With temperature constant, the volume of a given weight of gas varies inversely as its absolute pressure.* Hence, if P_1 and P_2 represent the initial and final absolute pressures, and V_1 and V_2 represent corresponding volumes of the same mass, say one pound of gas, then $\frac{V_1}{V_2} = \frac{P_2}{P_1}$, or $P_1 V_1 = P_2 V_2$, but since $P_1 V_1$ for any given case is a definite constant quantity, it follows that the product of the absolute

TABLE 3. WEIGHTS OF SATURATED AND PARTLY SATURATED AIR FOR VARIOUS BAROMETRIC AND HYGROMETRIC CONDITIONS.^{a, b}

Dry-Bulb Temperature, Degrees F	Barometric Pressure—Inches											
	26			27			28			29		
	Wt. per Cu. Ft. Sat. unad. Air	Deer's Wt. per Deg. Inc. Dry-Bulb	Incr's Wt. per 0.1° Rise in Bar.	Wt. per Cu. Ft. Sat. unad. Air	Deer's Wt. per Deg. Inc. Dry-Bulb	Incr's Wt. per 0.1° Rise in Bar.	Wt. per Cu. Ft. Sat. unad. Air	Deer's Wt. per Deg. Inc. Dry-Bulb	Incr's Wt. per 0.1° Rise in Bar.	Wt. per Cu. Ft. Sat. unad. Air	Deer's Wt. per Deg. Inc. Dry-Bulb	Incr's Wt. per 0.1° Rise in Bar.
0	0.07500	0.00016	0.00029	0.07788	0.00016	0.00029	0.08077	0.00017	0.00029	0.08365	0.00018	0.00029
10	0.07338	0.00016	0.00028	0.07450	0.00016	0.00028	0.07733	0.00017	0.00028	0.08185	0.00018	0.00028
20	0.07180	0.00016	0.00028	0.07143	0.00016	0.00027	0.07409	0.00016	0.00027	0.07839	0.00017	0.00027
30	0.07027	0.00015	0.00027	0.06992	0.00015	0.00026	0.07252	0.00016	0.00026	0.07675	0.00016	0.00026
40	0.06879	0.00015	0.00026	0.06843	0.00015	0.00025	0.07098	0.00015	0.00025	0.07512	0.00016	0.00025
50	0.06732	0.00015	0.00026	0.06692	0.00015	0.00025	0.06943	0.00015	0.00025	0.07353	0.00016	0.00025
60	0.06588	0.00015	0.00026	0.06542	0.00015	0.00025	0.06789	0.00015	0.00025	0.07193	0.00016	0.00025
70	0.06442	0.00015	0.00025	0.06398	0.00016	0.00024	0.06629	0.00016	0.00024	0.07034	0.00016	0.00025
80	0.06297	0.00015	0.00025	0.06258	0.00016	0.00024	0.06465	0.00016	0.00024	0.06870	0.00016	0.00024
90	0.06146	0.00015	0.00024	0.06111	0.00016	0.00023	0.06293	0.00016	0.00023	0.06703	0.00017	0.00024
100	0.05991	0.00016	0.00024	0.05957	0.00016	0.00023	0.06111	0.00018	0.00023	0.06526	0.00018	0.00023
110	0.05828	0.00016	0.00023	0.05782	0.00018	0.00023	0.05917	0.00019	0.00022	0.06339	0.00019	0.00023
120	0.05653	0.00018	0.00023	0.05611	0.00018	0.00022	0.05704	0.00021	0.00022	0.06142	0.00020	0.00022
130	0.05467	0.00019	0.00023	0.05483	0.00019	0.00022	0.05570	0.00021	0.00022	0.05925	0.00022	0.00022
140	0.05262	0.00021	0.00022	0.05253	0.00023	0.00022	0.05471	0.00023	0.00022	0.05689	0.00024	0.00022
150	0.05036	0.00023	0.00022	0.05001	0.00025	0.00022	0.05216	0.00026	0.00021	0.05430	0.00026	0.00021
160	0.04788	0.00025	0.00022	0.04720	0.00028	0.00021	0.04931	0.00029	0.00021	0.05141	0.00031	0.00021
170	0.04509	0.00028	0.00021	0.04404	0.00031	0.00021	0.04611	0.00032	0.00021	0.04818	0.00034	0.00021
180	0.04197	0.00031	0.00021	0.04049	0.00036	0.00021	0.04253	0.00036	0.00021	0.04457	0.00037	0.00021
190	0.03845	0.00035	0.00021	0.03650	0.00040	0.00020	0.03851	0.00040	0.00020	0.04052	0.00041	0.00020
200	0.03449	0.00040	0.00020									

^aFrom Fan Engineering.^bA convenient and accurate chart for quickly determining the weight of air under any condition of dry-bulb, wet-bulb, and pressure is A Chart for Determining the Weight of Moist Air in Pounds per Cubic Foot, by John E. Younger. Published in Mechanical Engineering, June, 1925.

pressure and volume of a gas is a constant, or $PV = C$, when T is kept constant. Any change in the pressure and volume of a gas at constant temperature is called an *isothermal change*.

Charles' Law refers to the relation among pressure, volume, and temperature of a gas and may be stated as follows: *The volume of a given weight of gas varies directly as the absolute temperature at constant pressure, and the pressure varies directly as the absolute temperature at constant volume.* Hence, when heat is added at constant volume, V_c , the resulting

equation is $\frac{P_2}{P_1} = \frac{T_2}{T_1}$, or, for the same temperature range at constant pressure, P_c , the relation is $\frac{V_2}{V_1} = \frac{T_2}{T_1}$.

In general, for any weight of gas, W , since volume is proportional to weight, the relation among P , V , and T is

$$PV = WRT \quad (3)$$

where

P = the absolute pressure of the gas, pounds per square foot.

V = the volume of the weight W , cubic feet.

W = the weight of the gas, pounds.

R = a constant depending on the nature of the gas. The average value of R for air is 53.34.

T = the absolute temperature, degrees Fahrenheit.

This is the characteristic equation for a perfect gas, and while no gases are perfect in this sense, they conform so nearly that Equation 3 will apply to most engineering computations.

HUMIDITY

Humidity is the water vapor mixed with dry air in the atmosphere. *Absolute humidity* has a multiplicity of meanings, but usually the term refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. With this meaning, absolute humidity is nothing but the actual density of the water vapor in the mixture and might better be so called. A study of Keenan's Steam Tables² indicates that water vapor, either saturated or super-heated, at partial pressures lower than 4 in. of mercury may be treated as a gas with a gas constant R of 1.21 in the characteristic equation of the gas $pV = wR(t + 460)$. Within such limits, the density (d) of water vapor is

$$d = \frac{w}{V} = \frac{e}{1.21(t + 460)} \text{ (pounds per cubic foot)} \quad (4a)$$

$$= \frac{5785 e}{t + 460} \text{ (grains per cubic foot)} \quad (4b)$$

where

e = actual partial pressure of vapor, inches of mercury.

t = dry-bulb temperature, degrees Fahrenheit.

²Published by American Society of Mechanical Engineers, see abstract in Table 6.

Specific Humidity

It simplifies many problems which deal with mixtures of dry air and water vapor to express the weight or the mass of the vapor in terms of the weight or the mass of dry air. If the weight of the water vapor in a mixture be divided by the weight of the dry air, and the weight of dry air be made unity, we have an expression of the weight of water vapor carried by a unit weight of dry air. This relation has no generally accepted name. It has been variously called: mixing ratio, proportionate humidity, mass or density ratio, absolute humidity, and specific humidity. Of all these terms *specific humidity* is the most suggestive of the meaning which it is desired to express and it has found considerable use in this sense even though it is defined in International Critical Tables as the ratio of the mass of vapor to the total mass. It will be understood here that *specific humidity* refers to the weight of water vapor in pounds carried by one pound of dry air.

The gas constant for dry air, when the partial pressure of the air is expressed in inches of Hg, is 0.753; so that the specific humidity, if represented by W , is

$$W = \frac{e}{1.21 (t + 460)} \div \frac{B - e}{0.753 (t + 460)} \\ = 0.622 \left(\frac{e}{B - e} \right) \text{ (pounds)} \quad (5a)$$

$$= 4354 \left(\frac{e}{B - e} \right) \text{ (grains)} \quad (5b)$$

where

e = actual partial pressure of vapor, inches of mercury.

B = total pressure of mixture (barometric pressure), inches of mercury.

Relative Humidity

Relative humidity (Φ) is either the ratio of the actual partial pressure, e , of the water vapor in the air to the saturation pressure, e_t , at the dry-bulb temperature, or the ratio of the actual density, d , of the vapor to the density of saturated vapor, d_t , at the dry-bulb temperature. That is:

$$\Phi = \frac{e}{e_t} = \frac{d}{d_t} \quad (6)$$

The relative humidity of a given mixture at a given temperature is not the same as the specific humidity, W , of the mixture divided by the specific humidity, W_t , of saturated vapor at the same temperature, for from Equations 5a and 6

$$\frac{W}{W_t} = 0.622 \left(\frac{\Phi e_t}{B - \Phi e_t} \right) \div 0.622 \left(\frac{e_t}{B - e_t} \right) = \frac{\Phi (B - e_t)}{B - \Phi e_t} \quad (7)$$

The specific humidity of an unsaturated air-vapor mixture cannot, therefore, be accurately found by multiplying the specific humidity of saturated vapor by its relative humidity; although the error is usually small especially when the relative humidity is high.

With a relative humidity of 100 per cent, the dry-bulb, wet-bulb, and

dew-point temperatures are equal. With a relative humidity less than 100 per cent, the dry-bulb exceeds the wet-bulb, and the wet-bulb exceeds the dew-point temperature.

RELATION OF DEW POINT TO RELATIVE HUMIDITY

A peculiar relationship exists between the dew point and the relative humidity and this is found most useful in air conditioning work. This relationship is, that for a fixed relative humidity there is substantially a constant difference between the dew point and the dry-bulb temperature over a considerable temperature range. Table 4, giving the dry-bulb and dew-point temperatures and the dew-point differentials for 50 per cent relative humidity, illustrates this relationship clearly.

TABLE 4. DRY-BULB AND DEW-POINT TEMPERATURES FOR 50 PER CENT RELATIVE HUMIDITY

Dry-bulb temperature.....	65.0	70.0	75.0	80.0	85.0	90.0
Dew-point temperature.....	45.8	50.5	55.25	59.75	64.25	68.75
Difference between dew-point and dry-bulb temperature.....	19.2	19.5	19.75	20.25	20.75	21.25

It will be seen from an inspection of this table that the difference between the dew-point temperature and the room temperature is approximately 20 deg throughout this range of dry-bulb temperatures or, to be more exact, the differential increases only 10 per cent for a range of practically 25 deg.

This principle holds true for other humidities and is due to the fact that the pressure of the water vapor practically doubles for every 20 deg through this range.

The approximate relative humidity for any difference between dew-point and dry-bulb temperature may be expressed in per cent as:

$$\frac{100}{2^{\frac{t-t_d}{20}}} \quad (8)$$

where

t_d = dew-point temperature.

This principle is very useful in determining the available cooling effect obtainable with saturated air when a desired relative humidity is to be maintained in a room, even though there may be a wide variation in room temperature. This problem is one which applies to certain industrial conditions, such as those in cotton mills and tobacco factories, where relatively high humidities are carried and where one of the principal problems is to remove the heat generated by the machinery. It also permits the use of a differential thermostat, responsive to both the room temperature and the dew-point temperature, to control the relative humidity in the room.

Table 5 gives, for different temperatures, the density of saturated vapor,

TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES^a

Temp., F	PRESSURE OF SATURATED VAPOR $\times 10^{-4}$		WEIGHT OF SATURATED VAPOR				VOLUME IN Cu Ft. BAROMETER, 29.92 In. Hg		HEAT CONTENT PER Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds $\times 10^{-4}$	Grains	Pounds $\times 10^{-4}$	Grains					
-130	0.376	0.1356	0.000693	0.000049	0.005738	0.00040	8.31	8.31	-31.71	1000.7	-31.71
-129	0.386	0.1503	0.000766	0.000054	0.006362	0.00045	8.33	8.33	-31.46	1001.2	-31.46
-128	0.398	0.1660	0.000843	0.000059	0.007027	0.00049	8.36	8.36	-31.21	1001.6	-31.21
-127	0.411	0.1832	0.000928	0.000065	0.007755	0.00054	8.39	8.39	-30.96	1002.1	-30.96
-126	0.425	0.2019	0.001019	0.000071	0.008545	0.00060	8.41	8.41	-30.71	1002.5	-30.71
-125	0.440	0.2235	0.001125	0.000079	0.009459	0.00066	8.43	8.43	-30.46	1003.0	-30.46
-124	0.456	0.2481	0.001230	0.000086	0.010377	0.00073	8.46	8.46	-30.21	1003.4	-30.21
-123	0.472	0.2762	0.001332	0.000093	0.011277	0.00079	8.48	8.48	-29.96	1003.9	-29.96
-122	0.489	0.3074	0.001440	0.000100	0.012166	0.00085	8.51	8.51	-29.72	1004.3	-29.72
-121	0.508	0.3420	0.001555	0.000112	0.013044	0.00097	8.53	8.53	-29.47	1004.8	-29.47
-120	0.528	0.3810	0.001679	0.000123	0.013911	0.00107	8.56	8.56	-29.22	1005.2	-29.22
-119	0.549	0.4244	0.001812	0.000134	0.014766	0.00117	8.58	8.58	-28.97	1005.7	-28.97
-118	0.571	0.4724	0.001954	0.000145	0.015611	0.00127	8.61	8.61	-28.72	1006.1	-28.72
-117	0.594	0.5250	0.002105	0.000157	0.016446	0.00137	8.63	8.63	-28.47	1006.6	-28.47
-116	0.618	0.5824	0.002264	0.000169	0.017271	0.00146	8.66	8.66	-28.23	1007.0	-28.23
-115	0.643	0.6456	0.002430	0.000181	0.018086	0.00155	8.68	8.68	-27.98	1007.5	-27.98
-114	0.669	0.7147	0.002603	0.000193	0.018891	0.00164	8.71	8.71	-27.73	1007.9	-27.73
-113	0.696	0.7898	0.002784	0.000205	0.019686	0.00173	8.73	8.73	-27.48	1008.4	-27.48
-112	0.724	0.8710	0.002972	0.000217	0.020471	0.00181	8.76	8.76	-27.23	1008.8	-27.23
-111	0.753	0.9584	0.003167	0.000229	0.021246	0.00189	8.78	8.78	-26.99	1009.3	-26.99
-110	0.783	1.0520	0.003369	0.000241	0.022011	0.00197	8.81	8.81	-26.74	1009.7	-26.74
-109	0.814	1.1519	0.003578	0.000253	0.022766	0.00205	8.83	8.83	-26.49	1010.2	-26.49
-108	0.846	1.2582	0.003793	0.000265	0.023511	0.00213	8.86	8.86	-26.24	1010.6	-26.24
-107	0.879	1.3719	0.004014	0.000277	0.024246	0.00221	8.89	8.89	-26.00	1011.1	-26.00
-106	0.913	1.4930	0.004241	0.000289	0.024971	0.00229	8.91	8.91	-25.75	1011.5	-25.75
-105	0.948	1.6214	0.004474	0.000301	0.025686	0.00237	8.94	8.94	-25.50	1012.0	-25.50
-104	0.984	1.7572	0.004713	0.000313	0.026391	0.00245	8.96	8.96	-25.26	1012.4	-25.26
-103	1.021	1.9005	0.004958	0.000325	0.027086	0.00253	8.99	8.99	-25.01	1012.8	-25.01
-102	1.059	2.0514	0.005208	0.000337	0.027771	0.00261	9.01	9.01	-24.76	1013.3	-24.76
-101	1.100	2.2199	0.005463	0.000349	0.028446	0.00269	9.04	9.04	-24.51	1013.8	-24.51
-100	1.142	2.4060	0.005723	0.000361	0.029111	0.00277	9.06	9.06	-24.27	1014.2	-24.27
-99	1.186	2.6097	0.005988	0.000373	0.029766	0.00285	9.09	9.09	-24.02	1014.7	-24.02
-98	1.232	2.8322	0.006258	0.000385	0.030411	0.00293	9.11	9.11	-23.78	1015.1	-23.78
-97	1.279	3.0747	0.006533	0.000397	0.031046	0.00301	9.14	9.14	-23.53	1015.6	-23.53
-96	1.328	3.3372	0.006813	0.000409	0.031671	0.00309	9.16	9.16	-23.28	1016.0	-23.28

^aCompiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES^a (CONTINUED)

Temp., F	PRESSURE OF SATURATED VAPOR $\times 10^{-4}$		WEIGHT OF SATURATED VAPOR				VOLUME IN Cu Ft. BAROMETER, 29.92 In. Hg		HEAT CONTENT PER Lb		
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds $\times 10^{-4}$	Grains	Pounds $\times 10^{-4}$	Grains					
-95	7.047	3.4004	0.015990	0.00112	0.1465	0.01026	9.19	9.19	-23.04	1016.5	-23.04
-94	6.987	3.3687	0.01584	0.00111	0.1450	0.01012	9.21	9.21	-22.79	1016.9	-22.79
-93	6.919	3.3277	0.01567	0.00110	0.1435	0.01000	9.23	9.23	-22.55	1017.3	-22.55
-92	6.843	3.2877	0.01549	0.00109	0.1420	0.00987	9.25	9.25	-22.30	1017.8	-22.30
-91	6.760	3.2486	0.01530	0.00108	0.1405	0.00975	9.27	9.27	-22.05	1018.3	-22.05
-90	6.671	3.2105	0.01511	0.00107	0.1390	0.00963	9.29	9.29	-21.81	1018.7	-21.81
-89	6.577	3.1734	0.01492	0.00106	0.1375	0.00951	9.31	9.31	-21.56	1019.2	-21.56
-88	6.479	3.1372	0.01473	0.00105	0.1360	0.00939	9.33	9.33	-21.32	1019.6	-21.32
-87	6.376	3.1019	0.01454	0.00104	0.1345	0.00927	9.35	9.35	-21.07	1020.1	-21.07
-86	6.269	3.0674	0.01435	0.00103	0.1330	0.00915	9.37	9.37	-20.83	1020.5	-20.83
-85	6.158	3.0337	0.01416	0.00102	0.1315	0.00903	9.39	9.39	-20.58	1021.0	-20.58
-84	6.043	2.9998	0.01397	0.00101	0.1300	0.00891	9.41	9.41	-20.34	1021.4	-20.34
-83	5.924	2.9658	0.01378	0.00100	0.1285	0.00879	9.43	9.43	-20.09	1021.9	-20.09
-82	5.801	2.9317	0.01359	0.00099	0.1270	0.00867	9.45	9.45	-19.84	1022.3	-19.84
-81	5.674	2.8974	0.01340	0.00098	0.1255	0.00855	9.47	9.47	-19.60	1022.8	-19.60
-80	5.543	2.8629	0.01321	0.00097	0.1240	0.00843	9.49	9.49	-19.36	1023.2	-19.36
-79	5.409	2.8282	0.01302	0.00096	0.1225	0.00831	9.51	9.51	-19.11	1023.7	-19.11
-78	5.272	2.7934	0.01283	0.00095	0.1210	0.00819	9.53	9.53	-18.87	1024.1	-18.87
-77	5.132	2.7584	0.01264	0.00094	0.1195	0.00807	9.55	9.55	-18.62	1024.6	-18.62
-76	4.989	2.7232	0.01245	0.00093	0.1180	0.00795	9.57	9.57	-18.38	1025.0	-18.37
-75	4.843	2.6878	0.01226	0.00092	0.1165	0.00783	9.59	9.59	-18.13	1025.5	-18.12
-74	4.694	2.6522	0.01207	0.00091	0.1150	0.00771	9.61	9.61	-17.89	1025.9	-17.88
-73	4.542	2.6164	0.01188	0.00090	0.1135	0.00759	9.63	9.63	-17.64	1026.4	-17.63
-72	4.388	2.5804	0.01169	0.00089	0.1120	0.00747	9.65	9.65	-17.40	1026.8	-17.39
-71	4.231	2.5442	0.01150	0.00088	0.1105	0.00735	9.67	9.67	-17.16	1027.3	-17.15
-70	4.071	2.5078	0.01131	0.00087	0.1090	0.00723	9.69	9.69	-16.91	1027.7	-16.90
-69	3.908	2.4712	0.01112	0.00086	0.1075	0.00711	9.71	9.71	-16.66	1028.2	-16.66
-68	3.742	2.4344	0.01093	0.00085	0.1060	0.00699	9.73	9.73	-16.41	1028.6	-16.41
-67	3.573	2.3974	0.01074	0.00084	0.1045	0.00687	9.75	9.75	-16.17	1029.1	-16.17
-66	3.401	2.3602	0.01055	0.00083	0.1030	0.00675	9.77	9.77	-15.93	1029.5	-15.93
-65	3.226	2.3228	0.01036	0.00082	0.1015	0.00663	9.79	9.79	-15.69	1030.0	-15.87
-64	3.049	2.2852	0.01017	0.00081	0.1000	0.00651	9.81	9.81	-15.45	1030.4	-15.43
-63	2.870	2.2474	0.00998	0.00080	0.0985	0.00639	9.83	9.83	-15.21	1030.9	-15.19
-62	2.689	2.2094	0.00979	0.00079	0.0970	0.00627	9.85	9.85	-14.96	1031.3	-14.94
-61	2.506	2.1712	0.00960	0.00078	0.0955	0.00615	9.87	9.87	-14.72	1031.8	-14.70

^aCompiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES^a (CONTINUED)

TEMP., F.	PRESSURE OF SATURATED VAPOR $\times 10^{-3}$		WEIGHT OF SATURATED VAPOR				VOLUME IN CU FT. BAROMETER, 29.92 IN. Hg		HEAT CONTENT PER LB.		
	In. of Hg	Lb per Sq In.	per Cu Ft.		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F. Datum	Vapor 32 F. Datum	Dry Air with Vapor to Saturate it
			Pounds $\times 10^{-3}$	Grains	Pounds $\times 10^{-3}$	Grains					
-60	101.4	49.808	0.20993	0.01470	2.108	0.14756	10.07	10.07	-14.48	1032.2	-14.46
-59	108.8	53.443	0.22469	0.01573	2.262	0.15834	10.09	10.09	-14.23	1032.7	-14.21
-58	116.3	57.127	0.23958	0.01677	2.418	0.16926	10.12	10.12	-13.99	1033.1	-13.97
-57	124.8	61.302	0.25645	0.01795	2.595	0.18165	10.14	10.14	-13.75	1033.6	-13.72
-56	133.4	65.526	0.27344	0.01914	2.773	0.19411	10.17	10.17	-13.50	1034.0	-13.47
-55	143.0	70.242	0.29239	0.02047	2.973	0.20811	10.19	10.19	-13.26	1034.5	-13.23
-54	153.0	75.154	0.31207	0.02184	3.181	0.22207	10.22	10.22	-13.02	1034.9	-12.99
-53	163.5	80.311	0.33267	0.02329	3.399	0.23793	10.24	10.24	-12.78	1035.4	-12.74
-52	174.9	86.911	0.35499	0.02485	3.636	0.25452	10.27	10.27	-12.53	1035.8	-12.49
-51	187.0	91.854	0.37862	0.02650	3.898	0.27216	10.29	10.29	-12.29	1036.3	-12.25
-50	199.9	98.191	0.40376	0.02826	4.156	0.29092	10.32	10.32	-12.05	1036.7	-12.01
-49	213.0	105.63	0.43017	0.03004	4.428	0.30998	10.34	10.34	-11.81	1037.2	-11.76
-48	227.0	113.19	0.45784	0.03187	4.705	0.32938	10.37	10.37	-11.57	1037.6	-11.52
-47	243.1	121.41	0.48744	0.03374	5.051	0.34913	10.40	10.40	-11.32	1038.1	-11.27
-46	259.5	127.47	0.51903	0.03563	5.395	0.37005	10.42	10.42	-11.08	1038.5	-11.02
-45	276.7	135.92	0.55213	0.03865	5.753	0.40271	10.45	10.45	-10.84	1039.0	-10.78
-44	295.0	144.90	0.58722	0.04111	6.133	0.43831	10.47	10.47	-10.60	1039.4	-10.54
-43	314.7	154.58	0.62493	0.04375	6.543	0.4801	10.50	10.50	-10.35	1039.9	-10.28
-42	335.3	164.70	0.66426	0.04650	6.971	0.48797	10.52	10.52	-10.11	1040.3	-10.04
-41	357.6	175.65	0.70672	0.04947	7.435	0.52045	10.55	10.55	-9.872	1040.8	-9.795
-40	380.3	186.80	0.74988	0.05249	7.907	0.55349	10.57	10.57	-9.629	1041.2	-9.547
-39	405.5	199.18	0.7976	0.05583	8.431	0.59017	10.60	10.60	-9.388	1041.7	-9.300
-38	431.2	211.81	0.8461	0.05922	8.965	0.62755	10.62	10.62	-9.146	1042.1	-9.053
-37	459.2	225.66	0.8959	0.06292	9.548	0.66838	10.65	10.65	-8.905	1042.6	-8.805
-36	489.4	239.90	0.9538	0.06677	10.16	0.71120	10.67	10.67	-8.663	1043.0	-8.557
-35	519.5	255.18	1.0122	0.07085	10.80	0.75900	10.69	10.69	-8.422	1043.5	-8.309
-34	552.4	271.34	1.0738	0.07517	11.49	0.80430	10.72	10.72	-8.180	1043.9	-8.062
-33	586.5	288.09	1.1374	0.07962	12.20	0.84900	10.75	10.75	-7.939	1044.4	-7.819
-32	623.7	306.36	1.2087	0.08447	12.97	0.90790	10.77	10.77	-7.698	1044.8	-7.582
-31	661.8	325.08	1.2774	0.08942	13.76	0.96320	10.80	10.80	-7.457	1045.3	-7.333
-30	701.2	344.33	1.3499	0.09449	14.58	1.0206	10.82	10.82	-7.216	1045.7	-7.094
-29	742.2	364.57	1.4260	0.09982	15.43	1.0801	10.85	10.85	-6.975	1046.2	-6.814
-28	791.2	388.64	1.5166	0.10516	16.45	1.1515	10.87	10.87	-6.734	1046.6	-6.562
-27	841.0	413.10	1.6083	0.11258	17.49	1.2243	10.90	10.90	-6.493	1047.1	-6.310
-26	892.1	438.20	1.7021	0.11914	18.55	1.2985	10.92	10.92	-6.251	1047.5	-6.057

^aCompiled by W. M. Sawdon, vapor pressures converted from International Critical Tables.TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR AT LOW TEMPERATURES^a (CONCLUDED)

TEMP., F.	PRESSURE OF SATURATED VAPOR $\times 10^{-3}$		WEIGHT OF SATURATED VAPOR				VOLUME IN CU FT. BAROMETER, 29.92 IN. Hg		HEAT CONTENT PER LB.		
	In. of Hg	Lb per Sq In.	per Cu Ft.		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F. Datum	Vapor 32 F. Datum	Dry Air with Vapor to Saturate it
			Pounds $\times 10^{-3}$	Grains	Pounds $\times 10^{-3}$	Grains					
-25	946.4	404.87	1.8016	0.12611	19.68	1.3776	10.95	10.95	-6.011	1048.0	-5.805
-24	1003.	422.67	1.9049	0.13334	20.86	1.4602	10.97	10.97	-5.770	1048.4	-5.551
-23	1064.	522.64	2.0162	0.14113	22.13	1.5491	11.00	11.00	-5.529	1048.9	-5.297
-22	1126.	553.09	2.1287	0.14901	23.42	1.6394	11.02	11.02	-5.288	1049.3	-5.052
-21	1192.	585.51	2.2484	0.15739	24.79	1.7353	11.05	11.05	-5.047	1049.8	-4.787
-20	1262.0	619.89	2.3750	0.16625	26.25	1.8375	11.07	11.07	-4.807	1050.2	-4.531
-19	1337.	656.73	2.5105	0.17574	27.81	1.9467	11.10	11.10	-4.566	1050.7	-4.274
-18	1416.	695.54	2.6527	0.18569	29.12	2.0615	11.13	11.13	-4.325	1051.1	-4.015
-17	1496.	734.84	2.7963	0.19574	30.54	2.1834	11.15	11.15	-4.085	1051.6	-3.758
-16	1584.	778.06	2.9542	0.20679	32.95	2.3068	11.18	11.18	-3.844	1052.0	-3.497
-15	1675.0	822.76	3.1168	0.21818	34.84	2.4388	11.20	11.21	-3.604	1052.5	-3.237
-14	1772.	870.41	3.2899	0.23029	36.86	2.5802	11.23	11.24	-3.363	1052.9	-2.975
-13	1870.	920.51	3.4714	0.24300	38.98	2.7286	11.25	11.26	-3.123	1053.4	-2.712
-12	1980.	972.58	3.6569	0.25617	41.19	2.8833	11.28	11.29	-2.883	1053.8	-2.449
-11	2093.	1028.1	3.8569	0.27019	43.54	3.0478	11.30	11.31	-2.642	1054.3	-2.183
-10	2210.0	1085.6	4.0666	0.28466	45.98	3.2186	11.33	11.34	-2.402	1054.7	-1.917
-9	2335.	1147.0	4.2871	0.30009	48.58	3.4006	11.35	11.36	-2.162	1055.2	-1.649
-8	2463.	1209.8	4.5120	0.31584	51.25	3.5875	11.38	11.39	-1.921	1055.6	-1.380
-7	2592.	1273.0	4.7534	0.33204	52.06	3.6442	11.40	11.41	-1.681	1056.1	-1.109
-6	2745.	1348.3	5.0066	0.35046	57.12	3.9984	11.43	11.44	-1.441	1056.5	-0.8375
-5	2898.0	1423.5	5.2738	0.36917	60.30	4.2210	11.45	11.46	-1.201	1057.0	-0.5636
-4	3055.	1500.6	5.5473	0.38831	63.57	4.4439	11.48	11.49	-0.9604	1057.4	-0.2882
-3	3222.	1582.6	5.8379	0.40865	67.03	4.6883	11.50	11.51	-0.7203	1057.9	-0.01098
-2	3397.	1668.6	6.1414	0.42960	70.60	4.9483	11.53	11.54	-0.4802	1058.3	+0.2679
-1	3580.	1758.5	6.4583	0.45208	74.50	5.2150	11.55	11.57	-0.2401	1058.8	+0.5487
0	3773.0	1853.3	6.7914	0.47500	78.52	5.5000	11.58	11.59	0	1059.2	+0.8317

^aCompiled by W. M. Sawdon, vapor pressures converted from International Critical Tables.

TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F_a

TEMP. F	PRESSURE OF SATURATED VAPOR		WEIGHT OF SATURATED VAPOR						VOLUME IN CU FT BAROMETER, 29.92 IN. Hg		HEAT CONTENT PER LB		
			per Cu Ft.		per lb of Dry Air								
	In. of Hg	Lb per Sq In.	Pounds	Grains	Pounds	Grains	of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it		
0	0.03773	0.01853	0.000007914	0.475	0.0007852	5.50	11.58	11.59	0.0000	1059.2	0.8317		
1	0.03975	0.01963	0.000071395	0.500	0.008275	5.79	11.60	11.62	0.2401	1059.7	1.117		
2	0.04186	0.02056	0.000075021	0.525	0.008714	6.10	11.63	11.64	0.4801	1060.1	1.404		
3	0.04409	0.02166	0.000078851	0.552	0.009179	6.43	11.65	11.67	0.7201	1060.6	1.694		
4	0.04645	0.02282	0.000082890	0.580	0.009671	6.77	11.68	11.70	0.9601	1061.0	1.986		
5	0.04896	0.02400	0.000087005	0.609	0.010171	7.12	11.70	11.72	1.200	1061.5	2.280		
6	0.05144	0.02527	0.000091399	0.640	0.01071	7.50	11.73	11.75	1.440	1061.9	2.577		
7	0.05412	0.02658	0.000095955	0.672	0.01127	7.89	11.75	11.77	1.680	1062.4	2.877		
8	0.05692	0.02796	0.00010070	0.705	0.01186	8.30	11.78	11.80	1.920	1062.8	3.180		
9	0.05988	0.02941	0.00010572	0.740	0.01247	8.73	11.80	11.83	2.160	1063.3	3.486		
10	0.06295	0.03092	0.00011090	0.776	0.01311	9.18	11.83	11.85	2.400	1063.7	3.795		
11	0.06618	0.03251	0.00011634	0.814	0.01379	9.65	11.86	11.88	2.640	1064.2	4.108		
12	0.06958	0.03418	0.00012206	0.854	0.01450	10.15	11.88	11.91	2.880	1064.6	4.424		
13	0.07309	0.03590	0.00012794	0.896	0.01523	10.66	11.91	11.93	3.120	1065.1	4.742		
14	0.07677	0.03771	0.00013410	0.940	0.01600	11.20	11.93	11.96	3.359	1065.5	5.064		
15	0.08067	0.03963	0.00014092	0.984	0.01682	11.77	11.96	11.99	3.600	1066.0	5.392		
16	0.08469	0.04160	0.00014732	1.031	0.01766	12.36	11.98	12.01	3.849	1066.4	5.722		
17	0.08895	0.04369	0.00015440	1.081	0.01855	12.99	12.00	12.04	4.079	1066.9	6.058		
18	0.09337	0.04586	0.00016174	1.132	0.01947	13.63	12.03	12.07	4.319	1067.3	6.397		
19	0.09797	0.04812	0.00016935	1.185	0.02043	14.30	12.06	12.09	4.569	1067.8	6.741		
20	0.1028	0.05050	0.00017747	1.242	0.02144	15.01	12.08	12.12	4.798	1068.2	7.088		
21	0.1078	0.05295	0.00018564	1.299	0.02250	15.75	12.11	12.15	5.038	1068.7	7.443		
22	0.1132	0.05560	0.00019439	1.361	0.02361	16.53	12.13	12.18	5.278	1069.1	7.802		
23	0.1186	0.05826	0.00020335	1.423	0.02476	17.33	12.16	12.20	5.518	1069.6	8.166		
24	0.1244	0.06111	0.00021276	1.489	0.02596	18.17	12.18	12.23	5.768	1070.0	8.536		
25	0.1304	0.06405	0.00022255	1.558	0.02722	19.05	12.21	12.26	5.998	1070.5	8.912		
26	0.1366	0.06710	0.00023248	1.629	0.02853	19.97	12.23	12.29	6.237	1071.0	9.292		
27	0.1432	0.07034	0.00024372	1.704	0.02991	20.94	12.26	12.32	6.477	1071.4	9.682		
28	0.1500	0.07368	0.00025545	1.781	0.03133	21.93	12.28	12.34	6.717	1071.8	10.075		
29	0.1571	0.07717	0.00026597	1.862	0.03283	22.99	12.31	12.37	6.957	1072.3	10.477		
30	0.1645	0.08080	0.00027797	1.946	0.03439	24.07	12.33	12.40	7.197	1072.7	10.886		
31	0.1722	0.08468	0.00029043	2.033	0.03601	25.19	12.36	12.43	7.437	1073.2	11.292		
32	0.1802	0.08886	0.00030337	2.123	0.03771	26.41	12.38	12.46	7.677	1073.6	11.703		
33	0.1879	0.09330	0.00031673	2.203	0.03933	27.52	12.41	12.49	7.917	1074.1	12.139		
34	0.1957	0.09610	0.00032660	2.288	0.04094	28.66	12.43	12.51	8.157	1074.5	12.556		

^aCompiled by W. M. Sawdon, vapor pressures converted from International Critical Tables.TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F_a (CONTINUED)

Temp., F	PRESSURE OF SATURATED VAPOR		WEIGHT OF SATURATED VAPOR				VOLUME IN CU FT BAROMETER, 29.92 IN. Hg		HEAT CONTENT PER LB		
			per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it	
	In. of Hg	Lb per Sq In.	Pounds	Grains	Pounds	Grains					
35	0.20360	0.1000	0.0003394	2.376	0.004262	29.83	12.46	12.54	8.397	1075.0	12.979
36	0.21195	0.1041	0.0003527	2.469	0.004438	31.07	12.48	12.57	8.636	1075.4	13.409
37	0.22050	0.1083	0.0003662	2.563	0.004618	32.33	12.51	12.60	8.876	1075.9	13.835
38	0.22925	0.1126	0.0003799	2.660	0.004803	33.62	12.53	12.63	9.116	1076.3	14.262
39	0.23842	0.1171	0.0003943	2.760	0.004996	34.97	12.56	12.66	9.356	1076.8	14.688
40	0.24778	0.1217	0.0004090	2.863	0.005194	36.36	12.59	12.69	9.596	1077.2	15.114
41	0.25755	0.1265	0.0004243	2.970	0.005401	37.80	12.61	12.72	9.838	1077.7	15.540
42	0.26773	0.1315	0.0004401	3.081	0.005616	39.31	12.64	12.75	10.08	1078.1	15.966
43	0.27832	0.1367	0.0004566	3.196	0.005840	40.88	12.66	12.78	10.32	1078.6	16.392
44	0.28911	0.1420	0.0004735	3.315	0.006069	42.48	12.69	12.81	10.56	1079.0	16.818
45	0.30031	0.1475	0.0004909	3.436	0.006306	44.14	12.71	12.84	10.80	1079.5	17.244
46	0.31191	0.1532	0.0005088	3.562	0.006553	45.87	12.74	12.87	11.04	1080.0	17.670
47	0.32393	0.1591	0.0005274	3.692	0.006808	47.66	12.76	12.90	11.28	1080.4	18.096
48	0.33635	0.1652	0.0005465	3.826	0.007072	49.50	12.79	12.93	11.52	1080.8	18.522
49	0.34917	0.1715	0.0005663	3.964	0.007345	51.42	12.81	12.96	11.76	1081.3	18.948
50	0.36241	0.1780	0.0005866	4.106	0.007620	53.38	12.84	12.99	12.00	1081.7	19.374
51	0.37625	0.1848	0.0006078	4.255	0.007921	55.45	12.86	13.02	12.23	1082.2	19.800
52	0.39061	0.1918	0.0006296	4.407	0.008226	57.58	12.89	13.06	12.47	1082.6	20.226
53	0.40546	0.1989	0.0006516	4.561	0.008534	59.74	12.91	13.09	12.71	1083.1	20.652
54	0.42093	0.2063	0.0006746	4.722	0.008856	61.99	12.94	13.12	12.95	1083.5	21.078
55	0.43670	0.2140	0.0006984	4.889	0.009192	64.34	12.96	13.15	13.19	1084.0	21.504
56	0.45288	0.2219	0.0007228	5.060	0.009536	66.75	12.99	13.19	13.43	1084.4	21.930
57	0.46945	0.2300	0.0007477	5.234	0.009890	69.23	13.01	13.22	13.67	1084.9	22.356
58	0.48645	0.2384	0.0007735	5.411	0.01026	71.82	13.04	13.25	13.91	1085.3	22.782
59	0.50310	0.2471	0.0008003	5.602	0.01064	74.48	13.06	13.29	14.15	1085.8	23.208
60	0.52142	0.2561	0.0008278	5.795	0.01103	77.21	13.09	13.32	14.39	1086.2	23.634
61	0.54035	0.2654	0.0008562	5.993	0.01144	80.08	13.11	13.35	14.63	1086.7	24.060
62	0.55970	0.2749	0.0008852	6.196	0.01186	83.02	13.14	13.39	14.87	1087.1	24.486
63	0.57885	0.2848	0.0009153	6.407	0.01229	86.03	13.16	13.42	15.11	1087.6	24.912
64	0.60042	0.2949	0.0009460	6.622	0.01274	89.18	13.19	13.46	15.35	1088.0	25.338
65	0.62179	0.3054	0.0009778	6.845	0.01320	92.40	13.21	13.49	15.59	1088.5	25.764
66	0.64378	0.3162	0.010105	7.074	0.01368	95.76	13.24	13.53	15.83	1089.0	26.190
67	0.66638	0.3273	0.010440	7.308	0.01417	99.19	13.26	13.57	16.07	1089.4	26.616
68	0.68968	0.3388	0.010816	7.571	0.01468	102.8	13.29	13.60	16.31	1089.9	27.042
69	0.71382	0.3506	0.011140	7.798	0.01520	106.4	13.31	13.64	16.55	1090.3	27.468

^aCompiled by W. M. Sawdon, vapor pressures converted from International Critical Tables.

TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F^a (CONTINUED)

TEMP., F	PRESSURE OF SATURATED VAPOR		WEIGHT OF SATURATED VAPOR				VOLUME IN Cu Ft BAROMETER, 29.92 IN. Hg		HEAT CONTENT PER Lb		
	In. of Hg	Lb per Sq. In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds	Grains	Pounds	Grains					
70	0.73866	0.3928	0.0011507	8.055	0.01574	110.2	13.34	13.68	16.79	1080.7	33.96
71	0.74331	0.3954	0.0011584	8.139	0.01603	111.2	13.37	13.71	17.07	1081.2	34.53
72	0.74808	0.3983	0.0012269	8.588	0.01688	118.2	13.40	13.74	17.37	1081.6	35.10
73	0.75288	0.4016	0.0012667	8.867	0.01748	122.6	13.42	13.79	17.51	1082.1	35.67
74	0.75765	0.4053	0.0013075	9.153	0.01809	126.6	13.44	13.83	17.75	1082.5	36.24
75	0.76244	0.4095	0.0013497	9.448	0.01873	131.1	13.47	13.87	17.99	1083.0	36.81
76	0.76728	0.4140	0.0013927	9.749	0.01938	135.4	13.49	13.91	18.23	1083.4	37.38
77	0.77216	0.4188	0.0014371	10.06	0.02005	140.7	13.52	13.95	18.47	1083.9	37.95
78	0.77708	0.4238	0.0014825	10.38	0.02075	145.3	13.54	13.99	18.71	1084.3	38.52
79	0.78205	0.4290	0.0015295	10.71	0.02147	150.3	13.57	14.03	18.95	1084.8	39.09
80	0.78708	0.4344	0.0015777	11.04	0.02221	155.5	13.59	14.08	19.19	1085.2	39.66
81	0.79216	0.4399	0.0016273	11.38	0.02295	160.9	13.62	14.12	19.43	1085.7	40.23
82	0.79728	0.4456	0.0016784	11.73	0.02370	166.4	13.64	14.16	19.67	1086.1	40.80
83	0.80244	0.4513	0.0017304	12.08	0.02446	172.1	13.67	14.21	19.91	1086.6	41.37
84	0.80765	0.4572	0.0017841	12.44	0.02523	178.0	13.69	14.26	20.15	1087.0	41.94
85	0.81291	0.4632	0.0018389	12.81	0.02601	184.0	13.72	14.30	20.39	1087.5	42.51
86	0.81822	0.4693	0.0018950	13.19	0.02680	190.3	13.74	14.34	20.63	1088.0	43.08
87	0.82358	0.4755	0.0019531	13.58	0.02760	196.7	13.77	14.39	20.87	1088.4	43.65
88	0.82898	0.4818	0.0020116	13.98	0.02841	203.3	13.79	14.44	21.11	1088.8	44.22
89	0.83442	0.4882	0.0020725	14.39	0.02923	210.1	13.82	14.48	21.35	1089.3	44.79
90	0.83991	0.4947	0.0021344	14.81	0.03006	217.1	13.84	14.53	21.59	1089.7	45.36
91	0.84544	0.5013	0.0021982	15.24	0.03090	224.4	13.87	14.58	21.83	1090.2	45.93
92	0.85101	0.5080	0.0022634	15.68	0.03175	231.8	13.89	14.63	22.07	1090.6	46.50
93	0.85662	0.5148	0.0023304	16.13	0.03261	239.5	13.92	14.69	22.32	1091.1	47.07
94	0.86228	0.5217	0.0023992	16.59	0.03348	247.5	13.94	14.73	22.56	1091.5	47.64
95	0.86798	0.5287	0.0024697	17.06	0.03436	255.6	13.97	14.79	22.80	1092.0	48.21
96	0.87372	0.5358	0.0025416	17.54	0.03524	263.9	13.99	14.84	23.04	1092.4	48.78
97	0.87951	0.5430	0.0026145	18.03	0.03613	272.7	14.02	14.90	23.28	1092.9	49.35
98	0.88534	0.5503	0.0026884	18.53	0.03702	281.9	14.04	14.95	23.52	1093.3	49.92
99	0.89121	0.5577	0.0027700	19.04	0.03791	291.5	14.07	15.01	23.76	1093.8	50.49
100	0.89712	0.5652	0.0028506	19.56	0.03880	300.5	14.10	15.07	24.00	1094.2	51.06
101	0.90307	0.5728	0.0029316	20.09	0.03970	310.3	14.12	15.12	24.24	1094.7	51.63
102	0.90906	0.5804	0.0030156	21.11	0.04060	320.4	14.15	15.18	24.48	1095.1	52.20
103	0.91509	0.5881	0.0031017	21.71	0.04150	330.8	14.17	15.23	24.72	1095.6	52.77
104	0.92116	0.5959	0.0031887	22.32	0.04240	341.5	14.20	15.29	24.96	1096.0	53.34

^aCompiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 F^a (CONTINUED)

TEMP., F	PRESSURE OF SATURATED VAPOR		WEIGHT OF SATURATED VAPOR				VOLUME IN Cu Ft BAROMETER, 29.92 IN. Hg		HEAT CONTENT PER Lb		
	In. of Hg	Lb per Sq. In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it
			Pounds	Grains	Pounds	Grains					
105	2.2414	1.1009	0.0032786	22.95	0.05037	352.6	14.22	15.37	25.20	1106.5	80.83
106	2.3084	1.1338	0.0033715	23.60	0.05200	364.0	14.25	15.44	25.44	1106.9	81.40
107	2.3770	1.1675	0.0034650	24.26	0.05368	375.8	14.27	15.50	25.68	1107.4	81.97
108	2.4473	1.2020	0.0035612	24.93	0.05541	387.9	14.30	15.57	25.92	1107.8	82.54
109	2.5196	1.2375	0.0036603	25.62	0.05719	400.3	14.32	15.64	26.16	1108.3	83.11
110	2.5939	1.274	0.0037622	26.34	0.05904	413.3	14.35	15.71	26.40	1108.7	83.68
111	2.6692	1.311	0.0038669	27.07	0.06092	426.4	14.37	15.78	26.64	1109.2	84.25
112	2.7466	1.350	0.0039748	27.81	0.06282	439.7	14.39	15.85	26.88	1109.6	84.82
113	2.8260	1.389	0.0040858	28.57	0.06473	453.3	14.41	15.93	27.12	1110.1	85.39
114	2.9094	1.429	0.0041991	29.34	0.06670	468.0	14.43	16.00	27.36	1110.5	85.96
115	2.9929	1.470	0.0043047	30.13	0.06873	483.9	14.47	16.08	27.60	1111.0	86.53
116	3.0784	1.512	0.0044208	30.95	0.07083	499.4	14.50	16.16	27.84	1111.4	87.10
117	3.1660	1.555	0.0045372	31.76	0.07291	515.3	14.52	16.24	28.08	1111.9	87.67
118	3.2578	1.600	0.0046620	32.63	0.07500	532.0	14.55	16.32	28.32	1112.3	88.24
119	3.3492	1.645	0.0047846	33.49	0.07710	548.8	14.57	16.41	28.56	1112.8	88.81
120	3.4449	1.692	0.0049115	34.38	0.07923	566.5	14.60	16.50	28.80	1113.2	89.38
121	3.5406	1.739	0.0050400	35.28	0.08134	584.4	14.62	16.58	29.04	1113.7	89.95
122	3.6404	1.788	0.0051733	36.21	0.08348	603.1	14.65	16.68	29.28	1114.1	90.52
123	3.7422	1.838	0.0053111	37.18	0.08562	622.4	14.67	16.77	29.52	1114.6	91.09
124	3.8460	1.889	0.0054540	38.15	0.08775	642.3	14.70	16.87	29.76	1115.0	91.66
125	3.9519	1.941	0.0055990	39.13	0.08988	662.8	14.72	16.96	30.00	1115.5	92.23
126	4.0618	1.995	0.0057534	40.14	0.09200	683.9	14.75	17.06	30.24	1116.0	92.80
127	4.1718	2.049	0.0059082	41.17	0.09413	705.6	14.77	17.15	30.48	1116.5	93.37
128	4.2858	2.105	0.0060631	42.22	0.09628	728.0	14.80	17.25	30.72	1117.0	93.94
129	4.4039	2.163	0.0062186	43.32	0.09843	751.8	14.83	17.35	30.96	1117.5	94.51
130	4.5200	2.221	0.0063744	44.41	0.10057	776.9	14.85	17.40	31.20	1118.0	95.08
131	4.6441	2.281	0.0065304	45.53	0.10271	800.1	14.88	17.51	31.44	1118.5	95.65
132	4.7703	2.343	0.0066871	46.70	0.10486	824.0	14.90	17.61	31.68	1119.0	96.22
133	4.8989	2.406	0.0068439	47.87	0.10700	848.9	14.93	17.71	31.92	1119.5	96.79
134	5.0289	2.470	0.0070101	49.07	0.10915	874.9	14.95	17.81	32.16	1120.0	97.36
135	5.1633	2.536	0.0071855	50.30	0.11130	901.9	14.98	17.91	32.41	1120.5	97.93
136	5.2997	2.603	0.0073647	51.55	0.11345	929.3	15.00	18.01	32.65	1121.0	98.50
137	5.4402	2.672	0.0075477	52.83	0.11560	957.4	15.03	18.11	32.89	1121.5	99.07
138	5.5827	2.742	0.0077322	54.12	0.11775	986.9	15.05	18.21	33.13	1122.0	99.64
139	5.7293	2.814	0.0079233	55.46	0.11990	1017.1	15.08	18.31	33.37	1122.5	100.21

^aCompiled by W. M. Sawdon, vapor pressures converted from *International Critical Tables*.

TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 Fa (CONTINUED)

Temp., F	PRESSURE OF SATURATED VAPOR		WEIGHT OF SATURATED VAPOR				VOLUME IN Cu Ft BAROMETER, 29.92 IN. Hg		HEAT CONTENT PER Lb			
	In. of Hg	Lb per Sq In.	per Cu Ft.		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it	
			Pounds	Grains	Pounds	Grains						
140	5.8779	2.887	0.008116	56.81	0.1521	1064.7	15.10	18.79	33.61	1122.2	204.30	
141	6.0030	2.962	0.008133	58.19	0.1570	1089.0	15.13	18.94	33.85	1122.7	210.11	
142	6.1273	3.037	0.008150	59.56	0.1618	1113.4	15.16	19.10	34.09	1123.1	216.26	
143	6.2516	3.112	0.008167	60.93	0.1667	1137.8	15.19	19.26	34.33	1123.6	222.53	
144	6.3759	3.187	0.008184	62.30	0.1715	1162.2	15.22	19.43	34.57	1124.0	229.02	
145	6.4999	3.262	0.008201	63.67	0.1764	1186.6	15.25	19.60	34.81	1124.5	235.71	
146	6.6240	3.337	0.008218	65.04	0.1812	1211.0	15.28	19.78	35.05	1124.9	242.71	
147	6.7480	3.412	0.008235	66.41	0.1861	1235.4	15.31	19.96	35.29	1125.4	250.02	
148	6.8721	3.487	0.008252	67.78	0.1909	1259.8	15.34	20.15	35.53	1125.8	257.43	
149	6.9961	3.562	0.008269	69.15	0.1958	1284.2	15.37	20.35	35.77	1126.3	265.20	
150	7.1202	3.637	0.008286	70.52	0.2007	1308.6	15.40	20.55	36.02	1126.7	273.19	
151	7.2442	3.712	0.008303	71.89	0.2056	1333.0	15.43	20.76	36.26	1127.2	281.54	
152	7.3683	3.787	0.008320	73.26	0.2105	1357.4	15.46	20.97	36.50	1127.6	290.21	
153	7.4923	3.862	0.008337	74.63	0.2154	1381.8	15.49	21.20	36.74	1128.1	299.25	
154	7.6164	3.937	0.008354	76.00	0.2203	1406.2	15.52	21.43	36.98	1128.5	308.61	
155	7.7404	4.012	0.008371	77.37	0.2252	1430.6	15.55	21.67	37.22	1129.0	318.34	
156	7.8645	4.087	0.008388	78.74	0.2301	1455.0	15.58	21.93	37.46	1129.4	328.51	
157	7.9885	4.162	0.008405	80.11	0.2350	1479.4	15.61	22.19	37.70	1129.9	339.04	
158	8.1126	4.237	0.008422	81.48	0.2399	1503.8	15.64	22.46	37.94	1130.3	350.02	
159	8.2366	4.312	0.008439	82.85	0.2448	1528.2	15.67	22.74	38.18	1130.8	361.36	
160	8.3607	4.387	0.008456	84.22	0.2497	1552.6	15.70	23.03	38.43	1131.2	373.38	
161	8.4847	4.462	0.008473	85.59	0.2546	1577.0	15.73	23.33	38.67	1131.7	386.76	
162	8.6088	4.537	0.008490	86.96	0.2595	1601.4	15.76	23.65	38.91	1132.1	399.89	
163	8.7328	4.612	0.008507	88.33	0.2644	1625.8	15.79	23.98	39.15	1132.5	413.84	
164	8.8569	4.687	0.008524	89.70	0.2693	1650.2	15.82	24.33	39.39	1133.0	428.42	
165	8.9809	4.762	0.008541	91.07	0.2742	1674.6	15.85	24.69	39.63	1133.5	443.34	
166	9.1050	4.837	0.008558	92.44	0.2791	1699.0	15.88	25.07	39.87	1133.9	458.81	
167	9.2290	4.912	0.008575	93.81	0.2840	1723.4	15.91	25.46	40.11	1134.4	473.11	
168	9.3531	4.987	0.008592	95.18	0.2889	1747.8	15.94	25.88	40.35	1134.8	489.18	
169	9.4771	5.062	0.008609	96.55	0.2938	1772.2	15.97	26.31	40.59	1135.3	505.11	
170	9.6012	5.137	0.008626	97.92	0.2987	1796.6	16.00	26.77	40.83	1135.7	521.91	
171	9.7252	5.212	0.008643	99.29	0.3036	1821.0	16.03	27.24	41.07	1136.2	539.79	
172	9.8493	5.287	0.008660	100.66	0.3085	1845.4	16.06	27.74	41.32	1136.6	558.68	
173	9.9733	5.362	0.008677	102.03	0.3134	1869.8	16.09	28.28	41.56	1137.1	578.76	
174	10.0974	5.437	0.008694	103.40	0.3183	1894.2	16.12	28.84	41.80	1137.5	613.05	

*Compiled by W. M. Sawdon; vapor pressures converted from International Critical Tables.

TABLE 5. PROPERTIES OF SATURATED WATER VAPOR WITH AIR, 0 F TO 200 Fa (CONCLUDED)

Temp., F	PRESSURE OF SATURATED VAPOR		WEIGHT OF SATURATED VAPOR				VOLUME IN Cu Ft BAROMETER, 29.92 IN. Hg		HEAT CONTENT PER Lb			
	In. of Hg	Lb per Sq In.	per Cu Ft.		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it	Dry Air 0 F Datum	Vapor 32 F Datum	Dry Air with Vapor to Saturate it	
			Pounds	Grains	Pounds	Grains						
175	13.674	6.716	0.017914	125.4	0.5235	3664.5	15.98	29.43	42.04	1138.0	637.78	
176	13.985	6.869	0.018294	128.1	0.5459	3827.3	16.01	30.03	42.28	1138.4	653.73	
177	14.303	7.025	0.018684	130.8	0.5697	3987.9	16.03	30.71	42.52	1138.9	669.35	
178	14.627	7.184	0.019080	133.6	0.5949	4146.3	16.06	31.41	42.76	1139.3	685.53	
179	14.954	7.345	0.019477	136.3	0.6215	4305.5	16.08	32.15	43.00	1139.8	701.39	
180	15.280	7.510	0.019888	139.2	0.6501	4465.7	16.11	32.94	43.24	1140.2	718.48	
181	15.632	7.678	0.020304	142.1	0.6805	4626.5	16.13	33.78	43.49	1140.7	735.74	
182	15.981	7.849	0.020729	145.1	0.7131	4788.1	16.16	34.68	43.73	1141.1	753.45	
183	16.337	8.024	0.021159	148.1	0.7481	4950.7	16.18	35.65	43.97	1141.6	771.80	
184	16.697	8.201	0.021598	151.2	0.7854	5115.8	16.21	36.67	44.21	1142.0	790.81	
185	17.066	8.382	0.022045	154.3	0.8258	5283.6	16.23	37.78	44.45	1142.5	810.51	
186	17.440	8.566	0.022497	157.5	0.8693	5454.1	16.26	38.98	44.69	1143.0	830.81	
187	17.821	8.753	0.022956	160.7	0.9162	5627.4	16.28	40.27	44.93	1143.4	851.81	
188	18.210	8.944	0.023424	164.0	0.9673	5803.6	16.31	41.67	45.18	1143.8	873.51	
189	18.605	9.138	0.023900	167.3	1.0227	5982.8	16.34	43.04	45.42	1144.3	895.81	
190	19.008	9.336	0.024384	170.7	1.0833	6165.1	16.36	44.55	45.66	1144.7	918.81	
191	19.419	9.538	0.024881	174.2	1.1490	6350.6	16.39	46.11	45.90	1145.2	942.51	
192	19.839	9.744	0.025380	177.7	1.2204	6539.4	16.41	47.74	46.14	1145.6	966.81	
193	20.266	9.954	0.025893	181.3	1.3006	6731.6	16.44	49.43	46.38	1146.1	991.81	
194	20.702	10.168	0.026413	184.9	1.3897	6927.4	16.46	51.18	46.62	1146.5	1017.51	
195	21.144	10.385	0.026939	188.6	1.4899	7126.8	16.49	53.00	46.86	1147.0	1043.81	
196	21.592	10.605	0.027472	192.3	1.6133	7329.9	16.51	54.89	47.10	1147.4	1070.81	
197	22.048	10.829	0.028019	196.1	1.7421	7535.8	16.54	56.85	47.34	1147.9	1098.51	
198	22.512	11.057	0.028571	200.0	1.8800	7744.6	16.56	58.88	47.59	1148.3	1126.81	
199	22.984	11.289	0.029129	203.9	2.061	7956.4	16.59	61.00	47.83	1148.8	1155.81	
200	23.465	11.525	0.029700	207.9	2.261	8171.2	16.61	63.20	48.07	1149.2	1185.51	

*Compiled by W. M. Sawdon; vapor pressures converted from International Critical Tables.

d_t , the weight of saturated vapor mixed with 1 lb of dry air, W_t , (at a relative humidity of 100 per cent and a barometric pressure, B , of 29.92 in. of mercury), the specific volume of dry air, and the volume of an air-vapor mixture containing 1 lb of dry air (at a relative humidity of 100 per cent and a pressure of 29.92 in. of mercury). The preceding equations or the data from Table 5 may be conveniently used in solving the following typical problems:

Example 2. Humidifying and Heating. Air is to be maintained at 70 F with a relative humidity of 40 per cent ($\Phi = 0.4$) when the outside air is at 0 F and 70 per cent relative humidity ($\Phi = 0.7$) and a barometric pressure, B , of 29.92 in. of mercury. Find the weight of water vapor added to each pound of dry air and the dew-point temperature of the humidified air.

Solution. From Equation 5a and Table 5,

$$W_1 = 0.622 \left(\frac{0.7 \times 0.03773}{29.92 - 0.0264} \right) = 0.000548 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622 \left(\frac{0.4 \times 0.7386}{29.92 - 0.295} \right) = 0.00618 \text{ lb per pound of dry air.}$$

The water vapor added per pound of dry air must be ($W_2 - W_1$) or 0.005632 lb. By inspection of Table 5, $W_2 = 0.00618$ at 44.5 F, so this is the dew-point temperature of the humidified air.

An approximation of the same result from Table 5 is

$$W_1 = 0.7 \times 0.0007852 = 0.00054964 \text{ lb per pound of dry air.}$$

$$W_2 = 0.4 \times 0.01574 = 0.006296 \text{ lb per pound of dry air.}$$

The water vapor added per pound of dry air is approximately 0.00574636 lb and the dew-point temperature is approximately 45 F. The degree of approximation is evident.

Example 3. Dehumidifying and Cooling. Air with a dry-bulb temperature of 84 F, a wet-bulb of 70 F, or a relative humidity of 50 per cent ($\Phi = 0.5$), and a barometric pressure, B , of 29.92 in. of mercury is to be cooled to 54 F. Find the dew-point temperature of the entering air and the weight of vapor condensed per pound of dry air.

Solution. From Equation 5a and Table 5,

$$W_1 = 0.622 \left(\frac{0.5 \times 1.1752}{29.92 - 0.5876} \right) = 0.01248 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622 \left(\frac{0.42003}{29.92 - 0.42003} \right) = 0.00887 \text{ lb per pound of dry air.}$$

Since $W_1 = W_t$ when $t = 63.4$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is ($W_1 - W_2$) or 0.00361 lb per pound of dry air.

An approximate result is

$$W_1 = 0.5 \times 0.02543 = 0.012715 \text{ lb per pound of dry air.}$$

$$W_2 = 1 \times 0.008856 = 0.008856 \text{ lb per pound of dry air, since the exit air is saturated.}$$

Since $W_1 = W_t$ at $t = 64$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is 0.003859 lb per pound of dry air. The degree of approximation is again evident.

ADIABATIC SATURATION OF AIR

The process of adiabatic saturation of air is of considerable importance in air conditioning. Suppose that 1 lb of dry air, initially unsaturated but carrying W lb of water vapor with a dry-bulb temperature, t , and a wet-

bulb temperature, t' , be made to pass through a tunnel containing an exposed water surface. Further assume the tunnel to be completely insulated, thermally, so that the only heat transfer possible is that between the air and water. As the air passes over the water surface, it will gradually pick up water vapor and will approach saturation at the initial wet-bulb temperature of the air, if the water be supplied at this wet-bulb temperature. During the process of adiabatic saturation, then, the dry-bulb temperature of the air drops to the wet-bulb temperature as a limit, the wet-bulb temperature remains substantially constant, and the weight of water vapor associated with each pound of dry air increases to W_t , as a limit, where W_t is the weight of saturated vapor per pound of dry air for saturation at the wet-bulb temperature.

Example 4. If air with a dry-bulb of 85 F and a wet-bulb of 70 F be saturated adiabatically by spraying with recirculated water, what will be the final temperature and the vapor content of the air?

Solution. The final temperature will be equal to the initial wet-bulb temperature or 70 F, and since the air is saturated at this temperature, from Table 5, $W = 0.01574$ lb per pound of dry air.

In the adiabatic saturation process, since the heat given up by the dry air and associated vapor in cooling to the wet-bulb temperature is utilized in evaporation of water at the wet-bulb temperature, W. H. Carrier has pointed out³ that the equation for the process of *adiabatic saturation*, and hence for a process of *constant wet-bulb temperature*, is:

$$h'_{fg}(W_t - W) = c_{pa}(t - t') + c_{ps}W(t - t') \quad (9a)$$

and using $c_{pa} = 0.24$ and $c_{ps} = 0.45$

$$h'_{fg}(W_t - W) = (0.24 + 0.45W)(t - t') \quad (9b)$$

where

h'_{fg} = latent heat of vaporization at t' , Btu per pound.

$(W_t - W)$ = increase in vapor associated with 1 lb of dry air when it is saturated adiabatically from an initial dry-bulb temperature, t , and an initial vapor content, W , pounds.

Knowing any two of the three primary variables, t , t' , or W , the third may be found from this equation for any process of adiabatic saturation.

TOTAL HEAT AND HEAT CONTENT

The total heat of a mixture of dry air and water vapor was originally defined by W. H. Carrier as

$$\Sigma = c_{pa}(t - 0) + W[h'_{fg} + c_{ps}(t - t')] \quad (10)$$

where

Σ = total heat of the mixture, Btu per pound of dry air.

c_{pa} = mean specific heat at constant pressure of dry air.

c_{ps} = mean specific heat at constant pressure of water vapor.

t = dry-bulb temperature, degrees Fahrenheit.

t' = wet-bulb temperature, degrees Fahrenheit.

W = weight of water vapor mixed with each pound of dry air, pounds.
 h'_{fg} = latent heat of vaporization at t' , Btu per pound.

Since this definition holds for any mixture of dry air and water vapor, the total heat of a mixture with a relative humidity of 100 per cent and at a temperature equal to the wet-bulb temperature (t') is

$$\Sigma' = c_{pa} (t' - 0) + W t' h'_{fg} \quad (11)$$

By equating Equation 10 to Equation 11, the equation for the adiabatic saturation process, Equation 9a, follows. This demonstrates that the adiabatic saturation process at constant wet-bulb temperature is also a process of constant total heat. In short, the total heat of a mixture of dry air and water vapor is the same for any two states of the mixture at the same wet-bulb temperature. This fact furnishes a convenient means of finding the total heat of an air-vapor mixture in any state.

Example 5. Find the total heat of an air-vapor mixture having a dry-bulb temperature of 85 F and a wet-bulb temperature of 70 F.

Solution. From Table 5, for saturation at the wet-bulb temperature $W_t = 0.01574$, and from Equation 11,

$$\Sigma' = c_{pa} (70 - 0) + 0.01574 h'_{fg} = 16.8 + 17.16 = 33.96$$

By considering the temperatures in Table 5 to be wet-bulb readings, the total heat of any air-vapor mixture may be obtained from the last column in the table.

Enthalpy

This total heat of an air-vapor mixture is not exactly equal to the true heat content or enthalpy of the mixture since the heat content of the liquid is not included in Equation 10. With the meaning of heat content in agreement with present practise in other branches of thermodynamics, the true heat content of a mixture of dry air and water vapor (with 0 F as the datum for dry air, and the saturated liquid at 32 F as the datum for the water vapor) is

$$h = c_{pa} (t - 0) + W h_s = 0.24 (t - 0) + W h_s \quad (12)$$

where

h = the heat content of the mixture, Btu per pound of dry air.
 t = the dry-bulb temperature, degrees Fahrenheit.
 W = the weight of vapor per pound of dry air, pounds.
 h_s = the heat content of the vapor in the mixture, Btu per pound.

The heat content of the water vapor in the mixture may be found in steam charts or tables when the dry-bulb temperature and the partial pressure of the vapor are known. Or, since the heat content of steam at low partial pressures, whether super-heated or saturated, depends only upon temperature, the following empirical equation, derived from Keenan's Steam Tables, may be used:

$$h_s = 1059.2 + 0.45 t \quad (13)$$

Substituting this value of h_s in Equation 12, the heat content of the mixture is

$$h = 0.24 (t - 0) + W (1059.2 + 0.45 t) \quad (14)$$

An energy equation can be written that applies, in general, to various air-conditioning processes, and this equation can be used to determine the quantity of heat transferred during such processes. In the most general form, this equation may be explained with the aid of Fig. 1 as follows:

The rectangle may represent any apparatus, e.g., a drier, humidifier, dehumidifier, cooling tower, or the like, by proper choice of the direction of the arrows.

In general, a mixture of air and water vapor, such as atmospheric air, enters the apparatus at 1 and leaves at 3. Water is supplied at some temperature, t_2 . For the flow of 1 lb of dry air (with accompanying vapor) through the apparatus, provided there is no appreciable change in the elevation or velocity of the fluids and no mechanical energy delivered to or by the apparatus,

$$h_1 + E_h + (W_s - W_1) h_2 = h_3 + R_c$$

or

$$E_h - R_c = h_3 - h_1 - (W_s - W_1) h_2 \quad (15)$$

where

E_h = the quantity of heat supplied per pound of dry air, Btu.
 R_c = the quantity of heat lost externally by heat transfer from the apparatus, Btu per pound of dry air.
 W_1 = the weight of water vapor entering, per pound of dry air.
 W_s = the weight of water vapor leaving, per pound of dry air.
 h_2 = the heat content of the water supplied at t_2 , Btu per pound.
 $h_3 - h_1$ = the increase in the heat content of the air-water vapor mixture in passing through the apparatus, Btu per pound of dry air.
 $= 0.24 (t_3 - t_1) + W_3 (1059.2 + 0.45 t_3) - W_1 (1059.2 + 0.45 t_1)$

The net quantity of heat added to or removed from air-water vapor mixtures in air conditioning work is frequently *approximated* by taking the differences in total heat at exit and entrance.

For example, in Fig. 1, an *approximate* result is

$$E_h - R_c = \Sigma_3 - \Sigma_1 \quad (16)$$

where

Σ_3 = the total heat of the air-vapor mixture at exit, Btu per pound of dry air.
 Σ_1 = the total heat of the air-vapor mixture at entrance, Btu per pound of dry air.

From the definitions of *total heat* and *heat content*, it may be demonstrated that Equation 16 is exactly equivalent to Equation 15, when, and only when, $t'_3 = t'_1 = t_2$; i.e., when the initial and final wet-bulb temperatures and the temperature of the water supplied are equal. The one process that meets these conditions is adiabatic saturation, and either equation will give a result of zero; for other conditions, Equation 16 is approximate but satisfactory for many calculations.

The following problems illustrate the application of these principles:

Example 6. Heating (data from Example 2). Assuming the water to be supplied at 50 F, the net quantity of heat supplied is, from Equation 15,

$$E_h - R_c = 0.24 (70 - 0) + 0.00548 \times 0.45 (70 - 0) + 0.005632 [1059.2 + 0.45 \times 70 - (60 - 32)] = 22.90 \text{ Btu per pound of dry air.}$$

Example 7. Cooling (data from Example 3). If the condensate is removed at 54 F the quantity of heat removed is found from Equation 15, by proper regard to the arrow direction in Fig. 1,

$$E_h + R_c = 0.24 (84 - 54) + 0.00887 \times 0.45 (84 - 54) + 0.00361 [1059.2 + 0.45 \times 84 - (54 - 32)] = 11.24 \text{ Btu per pound of dry air.}$$

Using Table 5, the initial total heat of the air-vapor mixture, since the wet-bulb temperature is 70 F, is 33.96 Btu per pound of dry air.

The final total heat is, from Table 5, since the exit air is saturated, 22.55 Btu per pound. Hence, using Equation 16, the quantity of heat removed is, approximately, (33.96 - 22.55) or 11.41 Btu per pound of dry air. The degree of approximation to the correct result is evident in this example.

PSYCHROMETRIC CHART⁴

The revised Bulkeley Psychrometric Chart⁵, will be found attached to the inside back cover. It shows graphically the relationships expressed in Equations 9a and 9b. It also gives the grains of moisture per pound of

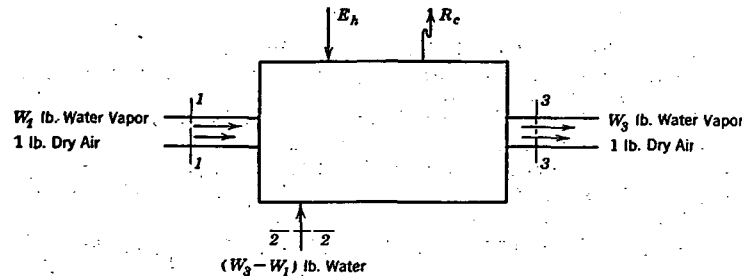


FIG. 1. DIAGRAM ILLUSTRATING ENERGY EQUATION 15

dry air for saturation, the grains of moisture per cubic foot of saturated air, the total heat in Btu per pound of dry air saturated with moisture, and the weight of the dry air in pounds per cubic foot. Fig. 2 shows the procedure to follow in using the Bulkeley Chart. The directrix curves above the saturation line are as follows:

A is the total heat in Btu contained in the mixture above 0 F, and is to be referred to the column of figures at the left side of the chart. Heat of the liquid is not included.

B is the grains of moisture of water vapor contained in each pound of the saturated mixture and is to be referred to the figures at the left side of the chart.

C is the grains of moisture of water vapor per cubic foot of saturated mixture, and is to be referred to the figures at the left side of the chart which are to be divided by 10.

D is the weight in decimal fractions of a pound, of one cubic foot of the saturated mixture, and is referred to the first column of figures to the right of the saturation line between the vertical dry-bulb temperature lines 170 and 180 F. The relative density of the mixture is read in a similar manner from the same curve by the column of figures between the vertical dry-bulb temperature lines 180 and 190 F.

E is similar to *D* but is for dry air, devoid of all moisture or water vapor. For con-

⁴See A Review of Psychrometric Charts, C. O. Mackey (Heating and Ventilating, June, July, 1931).

⁵The Bulkeley Psychrometric Chart was presented to the Society in 1926. (See A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926.) Single copy of the chart can be furnished at a cost of \$.50.

venience, the approximate absolute temperature of 500 F is given at 40 F on the saturation line for the purpose of calculating volume, weight per cubic foot, and relative density at partial saturation.

METHOD OF USING THE CHART

Example 8. Relative Humidity: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, the relative humidity is read directly on the straight diagonal lines as 46 per cent.

Example 9. Dew Point: At the intersection of the 78 F wet-bulb line, the dew-point temperature is read directly on the horizontal temperature lines as 70.9 F.

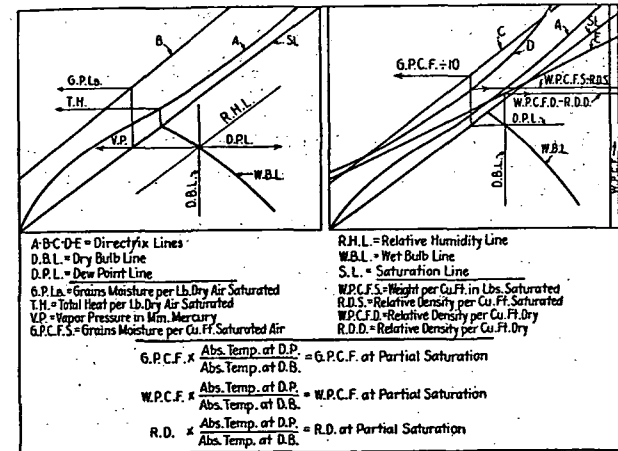


FIG. 2. DIAGRAMS SHOWING PROCEDURE TO FOLLOW IN USING BULKELEY CHART

Example 10. Vapor Pressure: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, pass in a horizontal direction to the left of the chart and on the logarithmic scale read the vapor pressure as 19.4 millimeters of mercury. (Divide by 25.4 for inches.)

Example 11. Total Heat Above 0 F in Mixture per Pound of Dry Air Saturated with Moisture: From where the wet-bulb line joins the saturation line, pass in a vertical direction on the 78 F dry-bulb line to its intersection with curve *A* and on the logarithmic scale at the left of the chart read 40.6 Btu per pound of mixture. The use of this curve to obtain the total heat in the mixture at any wet-bulb temperature is a great convenience, as the number of Btu required to heat the mixture and humidify it, as well as the refrigeration required to cool and dehumidify the mixture, can be obtained by taking the difference in total heat before and after treatment of the mixture.

Example 12. Grains of Moisture per Pound of Mixture: From 70.9 F dew-point temperature on the saturation line, pass vertically to the intersection with curve *B* and on the logarithmic scale at the left read 114 grains of moisture per pound.

Example 13. Grains of Moisture per Cubic Foot of Mixture, Partially Saturated: From 70.9 F dew-point temperature on the saturation line proceed in a vertical direction to curve *C*, and on the logarithmic scale to the left read 83.3 which, divided by 10, gives 8.33 grains. A temperature of 70.9 F is equal to an absolute temperature of 530.9, and 95 F equals 555, absolute temperature. Therefore, $\frac{530.9}{555} \times 8.33 = 7.97$ grains per cubic foot of partially saturated mixture.

Example 14. Grains of Moisture per Cubic Foot of Dry Air, Saturated: Starting at the saturation line at the desired temperature, pass in a vertical direction to curve *C* and on

the logarithmic scale at the left, read a number which, divided by 10, will give the answer.

Example 15. Weight per Cubic Foot of Dry Air and Relative Density: From the point where, for example, the 70 F vertical dry-bulb line intersects curve *E*, pass to right side and read 0.075 lb; if cubic feet per pound are desired, divide 1 by this amount. The relative density is read immediately to the right as 1.00.

Example 16. Weight per Cubic Foot of Saturated Air and Relative Density: From the point where, for example, the 70 F vertical line intersects the curve *D*, pass to the right and read weight per cubic foot as 0.07316 with a relative density of 0.9755 for saturated air at 70 F.

Example 17. Weight per Cubic Foot and Relative Density of Partially Saturated Air: Air at 50 F and a wet-bulb temperature of 46 F is to be heated to 130 F. The wet- and dry-bulb lines intersect at a dew-point temperature of 42 F. Pass to the left where this dew-point line intersects the saturation line and then pass in a vertical direction to where the 42 F dry-bulb line intersects with curve *D*. Then pass directly to the right and read the weight per cubic foot of saturated air at 42 F as 0.07844 and the relative density as 1.046. The absolute temperature at 42 F is 502, and at 130 F is 590. Therefore, $\frac{502}{590} = 0.851$. The weight of 1 cu ft of air at 50 F dry-bulb and 46 F wet-bulb when heated to 130 F is $0.07844 \times 0.851 = 0.06675$, and the relative density is $1.046 \times 0.851 = 0.89$.

PROPERTIES OF STEAM

Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. This change in state takes place at a definite and constant temperature which is determined solely by the pressure of the steam. The volume of a pound of steam is the *specific volume* which decreases as the pressure increases. The reciprocal of this, or the weight of steam per cubic foot, is the *density*. (See Table 6.)

Steam which is in contact with the water from which it was generated is known as *saturated steam*. If it contains no actual water in the form of mist or priming, it is called *dry saturated steam*. If this be heated and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated*, that is, its temperature will be higher than that of saturated steam at the same pressure.

PROPERTIES OF WATER

Composition of Water. Water is a chemical compound (H_2O) formed by the union of two volumes of hydrogen and one volume of oxygen, or two parts by weight of hydrogen and 16 parts by weight of oxygen.

Density of Water. Water has its greatest density at 39.2 F, and it expands when heated or cooled from this temperature. At 62 F a U. S. gallon of 231 cu in. of water weighs approximately $8\frac{1}{3}$ lb, and a cubic foot of water is equal to 7.48 gal. The specific volume of water depends on the temperature and it is always the reciprocal of its density. (See Table 7.)

Water Pressures. Pressures are often stated in feet or inches of water column. At 62 F, with *h* equal to the head in feet, the pressure of a column of water is $62.383h$ lb per square foot, or $0.433h$ lb per square inch. A column of water 2.309 ft (27.71 in.) high exerts a pressure of one pound per square inch at 62 F.

Boiling Point of Water. The boiling point of water varies with the pressure; it is lower at higher altitudes. A change in pressure will always

TABLE 6. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE^a

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid V _l	Evap. V _g	Sat. Vapor V _g	Sat. Liquid h _l	Evap. h _{fg}	Sat. Vapor h _g	Sat. Liquid s _l	Evap. s _{fg}	Sat. Vapor s _g	
1/4" Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7	0.0533	2.0422	2.0955	1/4" Hg
1/2" Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0	0.0754	1.9856	2.0609	1/2" Hg
3/4" Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9	0.0914	1.9451	2.0365	3/4" Hg
1" Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6	0.1147	1.8877	2.0024	1" Hg
1 1/2" Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8	0.1316	1.8468	1.9784	1 1/2" Hg
2" Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1	0.1450	1.8148	1.9598	2" Hg
3" Hg	115.08	0.01618	231.8	231.8	82.96	1027.9	1110.8	0.1561	1.7885	1.9446	3" Hg
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0	0.1326	1.8442	1.9769	1.0
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6	0.1750	1.7442	1.9192	2.0
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0	0.2009	1.6847	1.8856	3.0
4.0	152.99	0.01636	90.72	90.74	120.83	1005.9	1126.8	0.2198	1.6420	1.8618	4.0
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6	0.2348	1.6088	1.8435	5.0
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7	0.2473	1.5814	1.8287	6.0
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4	0.2580	1.5582	1.8162	7.0
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9	0.2674	1.5379	1.8053	8.0
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0	0.2758	1.5200	1.7958	9.0
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0	0.2834	1.5040	1.7874	10.0
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8	0.2903	1.4894	1.7797	11.0
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4	0.2968	1.4760	1.7727	12.0
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9	0.3027	1.4636	1.7663	13.0
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3	0.3082	1.4521	1.7604	14.0
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	14.696
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8	0.3184	1.4312	1.7496	16.0
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0	0.3274	1.4127	1.7402	18.0
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0	0.3356	1.3960	1.7317	20.0
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8	0.3431	1.3809	1.7240	22.0
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5	0.3500	1.3670	1.7170	24.0
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0	0.3564	1.3542	1.7106	26.0
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4	0.3624	1.3422	1.7046	28.0
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7	0.3680	1.3310	1.6990	30.0
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1165.0	0.3732	1.3206	1.6938	32.0
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1166.1	0.3783	1.3107	1.6890	34.0
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1167.2	0.3830	1.3014	1.6844	36.0
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1168.3	0.3876	1.2925	1.6800	38.0
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2	0.3919	1.2840	1.6759	40.0
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1170.2	0.3961	1.2759	1.6720	42.0
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1171.1	0.4000	1.2682	1.6683	44.0
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1171.9	0.4039	1.2608	1.6647	46.0
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1172.7	0.4076	1.2537	1.6613	48.0
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5	0.4111	1.2469	1.6580	50.0
60.0	283.49	0.01726	8.189	8.206	252.52	921.7	1174.3	0.4145	1.2404	1.6549	60.0
64.0	285.90	0.01728	7.902	7.919	254.99	920.0	1175.0	0.4178	1.2340	1.6518	64.0
68.0	288.23	0.01730	7.636	7.653	257.38	918.3	1175.7	0.4210	1.2279	1.6489	68.0
72.0	290.50	0.01732	7.388	7.405	259.71	916.6	1176.4	0.4241	1.2220	1.6461	72.0
80.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0	0.4271	1.2162	1.6434	80.0
82.0	294.85	0.01737	6.937	6.955	264.18	913.4	1177.6	0.4300	1.2107	1.6407	82.0
84.0	296.94	0.01739	6.732	6.749	266.33	911.9	1178.2	0.4329	1.2053	1.6382	84.0
86.0	298.98	0.01741	6.539	6.556	268.43	910.4	1178.8	0.4356	1.2001	1.6357	86.0
88.0	300.98	0.01743	6.357	6.375	270.49	908.9	1179.4	0.4384	1.1950	1.6333	88.0
90.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9	0.4410	1.1900	1.6310	90.0
92.0	304.82	0.01746	6.024	6.041	274.45	906.0	1180.5	0.4435	1.1852	1.6287	92.0
94.0	306.68	0.01748	5.870	5.887	276.37	904.6	1181.0	0.4460	1.1805	1.6265	94.0
96.0	308.50	0.01750	5.723	5.741	278.25	903.2	1181.5	0.4485	1.1759	1.6244	96.0
98.0	310.28	0.01752	5.584	5.602	280.09	901.9	1181.7	0.4509	1.1714	1.6223	98.0
100.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4	0.4532	1.1670	1.6202	100.0
102.0	313.74	0.01756	5.325	5.343	283.67	899.2	1182.9	0.4555	1.1627	1.6182	102.0
104.0	315.42	0.01757	5.204	5.222	285.42	897.9	1183.4	0.4578	1.1586	1.6163	104.0
106.0	317.06	0.01759	5.089	5.107	287.13	896.7	1183.8	0.4599	1.1545	1.6144	106.0
108.0	318.68	0.01761	4.979	4.997	288.80	895.4	1184.2	0.4621	1.1505	1.6126	108.0
110.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6	0.4642	1.1465	1.6107	110.0
112.0	321.83	0.01764	4.773	4.791	292.07	893.0	1185.0	0.4663	1.1427	1.6090	112.0
114.0	323.37	0.01766	4.676	4.694	293.67	891.8	1185.4	0.4683	1.1389	1.6072	114.0
116.0	324.88	0.01768	4.584	4.602	295.25	890.6	1185.8	0.4703	1.1352	1.6055	116.0
118.0	326.37	0.01769	4.494	4.512	296.80	889.4	1186.2	0.4723	1.1316	1.6038	118.0

^aAbstracted from *Steam Tables and Mollier Diagram*, by Prof. J. H. Keenan, 1930 edition, by permission of the publisher, *The American Society of Mechanical Engineers*.

TABLE 6. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Continued)

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid Vi	Evap. Vig	Sat. Vapor Vg	Sat. Liquid ht	Evap. hfg	Sat. Vapor hg	Sat. Liquid st	Evap. Sig	Sat. Vapor Sg	
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6	0.4742	1.1280	1.6022	100.0
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1186.9	0.4761	1.1245	1.6006	102.0
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1187.3	0.4779	1.1211	1.5990	104.0
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1187.6	0.4798	1.1177	1.5974	106.0
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1188.0	0.4816	1.1144	1.5959	108.0
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3	0.4834	1.1111	1.5944	110.0
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1188.6	0.4851	1.1079	1.5930	112.0
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1188.9	0.4868	1.1048	1.5915	114.0
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1189.2	0.4885	1.1017	1.5901	116.0
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	118.0
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8	0.4918	1.0956	1.5874	120.0
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1190.1	0.4934	1.0926	1.5860	122.0
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1190.4	0.4950	1.0897	1.5847	124.0
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1190.6	0.4965	1.0868	1.5834	126.0
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1190.9	0.4981	1.0840	1.5821	128.0
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2	0.4996	1.0812	1.5808	130.0
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1191.4	0.5011	1.0784	1.5796	132.0
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1191.7	0.5026	1.0757	1.5783	134.0
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1191.9	0.5041	1.0730	1.5771	136.0
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1192.2	0.5056	1.0703	1.5759	138.0
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4	0.5070	1.0677	1.5747	140.0
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1192.6	0.5084	1.0651	1.5735	142.0
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1192.9	0.5098	1.0625	1.5724	144.0
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1193.1	0.5112	1.0600	1.5712	146.0
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1193.3	0.5126	1.0575	1.5701	148.0
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5	0.5140	1.0550	1.5690	150.0
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1193.7	0.5153	1.0526	1.5679	152.0
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1193.9	0.5166	1.0502	1.5668	154.0
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1194.1	0.5180	1.0478	1.5658	156.0
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1194.3	0.5193	1.0454	1.5647	158.0
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5	0.5205	1.0431	1.5636	160.0
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1194.7	0.5218	1.0408	1.5626	162.0
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1194.9	0.5230	1.0385	1.5616	164.0
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1195.1	0.5243	1.0363	1.5606	166.0
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1195.3	0.5255	1.0340	1.5596	168.0
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4	0.5268	1.0318	1.5586	170.0
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1195.6	0.5280	1.0296	1.5576	172.0
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1195.8	0.5292	1.0275	1.5566	174.0
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1196.0	0.5304	1.0253	1.5557	176.0
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1196.1	0.5315	1.0232	1.5548	178.0
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3	0.5327	1.0211	1.5538	180.0
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1196.4	0.5339	1.0190	1.5529	182.0
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1196.6	0.5350	1.0169	1.5520	184.0
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1196.8	0.5362	1.0149	1.5511	186.0
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1196.9	0.5373	1.0129	1.5502	188.0
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0	0.5384	1.0109	1.5493	190.0
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1197.2	0.5395	1.0089	1.5484	192.0
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1197.3	0.5406	1.0070	1.5475	194.0
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1197.5	0.5417	1.0050	1.5467	196.0
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1197.6	0.5427	1.0031	1.5458	198.0
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8	0.5438	1.0012	1.5450	200.0
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1198.1	0.5465	0.9964	1.5429	205.0
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4	0.5491	0.9918	1.5409	210.0
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1198.7	0.5516	0.9873	1.5389	215.0
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0	0.5540	0.9829	1.5369	220.0
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1199.3	0.5565	0.9786	1.5350	225.0
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6	0.5588	0.9743	1.5332	230.0
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1199.8	0.5612	0.9702	1.5313	235.0
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1	0.5635	0.9661	1.5295	240.0
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1200.3	0.5658	0.9620	1.5278	245.0

TABLE 6. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Continued)

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid V _l	Evap. V _{lg}	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	
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1200.5	0.5680	0.9581	1.5261	250.0
260.0	404.43	0.01872	1.7536	1.7723	379.78	821.2	1201.0	0.5723	0.9504	1.5227	260.0
270.0	407.79	0.01877	1.6895	1.7083	383.44	818.0	1201.4	0.5765	0.9430	1.5194	270.0
280.0	411.06	0.01882	1.6302	1.6490	387.02	814.7	1201.8	0.5805	0.9357	1.5163	280.0
290.0	414.24	0.01887	1.5745	1.5934	390.50	811.6	1202.1	0.5845	0.9287	1.5132	290.0
300.0	417.33	0.01892	1.5225	1.5414	393.90	808.5	1202.4	0.5883	0.9220	1.5102	300.0
320.0	423.29	0.01901	1.4279	1.4469	400.47	802.5	1203.0	0.5957	0.9089	1.5046	320.0
340.0	428.96	0.01910	1.3439	1.3630	406.75	796.6	1203.4	0.6027	0.8965	1.4992	340.0
360.0	434.39	0.01918	1.2689	1.2881	412.80	790.9	1203.7	0.6094	0.8846	1.4940	360.0
380.0	439.59	0.01927	1.2015	1.2208	418.61	785.3	1203.9	0.6157	0.8733	1.4891	380.0
400.0	444.58	0.0194	1.1407	1.1601	424.2	779.8	1204.1	0.6218	0.8625	1.4843	400.0
420.0	449.38	0.0194	1.0853	1.1047	429.6	774.5	1204.1	0.6277	0.8520	1.4798	420.0
440.0	454.01	0.0195	1.0345	1.0540	434.8	769.3	1204.1	0.6334	0.8420	1.4753	440.0
460.0	458.48	0.0196	0.9881	1.0077	439.9	764.1	1204.0	0.6388	0.8322	1.4711	460.0
480.0	462.80	0.0197	0.9456	0.9633	444.9	759.0	1203.9	0.6441	0.8228	1.4670	480.0
500.0	466.99	0.0198	0.9063	0.9261	449.7	754.0	1203.7	0.6493	0.8137	1.4630	500.0
520.0	471.05	0.0198	0.8701	0.8899	454.4	749.0	1203.5	0.6543	0.8048	1.4591	520.0
540.0	474.99	0.0199	0.8363	0.8562	459.0	744.1	1203.2	0.6592	0.7962	1.4554	540.0
560.0	478.82	0.0200	0.8047	0.8247	463.6	739.3	1202.9	0.6639	0.7878	1.4517	560.0
580.0	482.55	0.0201	0.7751	0.7952	468.0	734.5	1202.5	0.6686	0.7796	1.4482	580.0
600.0	486.17	0.0202	0.7475	0.7677	472.3	729.8	1202.1	0.6731	0.7716	1.4447	600.0
620.0	489.71	0.0202	0.7217	0.7419	476.6	725.1	1201.7	0.6775	0.7638	1.4413	620.0
640.0	493.16	0.0203	0.6972	0.7175	480.8	720.5	1201.2	0.6818	0.7562	1.4380	640.0
660.0	496.53	0.0204	0.6744	0.6948	484.9	715.9	1200.8	0.6861	0.7487	1.4348	660.0
680.0	499.82	0.0205	0.6527	0.6732	488.9	711.3	1200.2	0.6902	0.7414	1.4316	680.0
700.0	503.04	0.0206	0.6321	0.6527	492.9	706.8	1199.7	0.6943	0.7342	1.4285	700.0
720.0	506.19	0.0206	0.6128	0.6334	496.8	702.4	1199.2	0.6983	0.7272	1.4255	720.0
740.0	509.28	0.0207	0.5944	0.6151	500.6	697.9	1198.6	0.7022	0.7203	1.4225	740.0
760.0	512.30	0.0208	0.5769	0.5977	504.4	693.5	1198.0	0.7060	0.7136	1.4196	760.0
780.0	515.27	0.0209	0.5602	0.5811	508.2	689.2	1197.4	0.7098	0.7069	1.4167	780.0
800.0	518.18	0.0209	0.5444	0.5653	511.8	684.9	1196.7	0.7135	0.7004	1.4139	800.0
820.0	521.03	0.0210	0.5293	0.5503	515.5	680.6	1196.0	0.7171	0.6940	1.4111	820.0
840.0	523.83	0.0211	0.5149	0.5360	519.0	676.4	1195.4	0.7207	0.6877	1.4084	840.0
860.0	526.58	0.0212	0.5013	0.5225	522.6	672.1	1194.7	0.7242	0.6815	1.4057	860.0
880.0	529.29	0.0213	0.4881	0.5094	526.0	667.9	1194.0	0.7277	0.6754	1.4031	880.0
900.0	531.95	0.0213	0.4756	0.4969	529.5	663.8	1193.3	0.7311	0.6694	1.4005	900.0
920.0	534.56	0.0214	0.4630	0.4849	532.9	659.7	1192.6	0.7344	0.6635	1.3980	920.0
940.0	537.13	0.0215	0.4509	0.4735	536.2	655.6	1191.8	0.7377	0.6577	1.3954	940.0
960.0	539.66	0.0216	0.4409	0.4623	539.6	651.5	1191.1	0.7410	0.6520	1.3930	960.0
980.0	542.14	0.0217	0.4303	0.4520	542.8	647.5	1190.3	0.7442	0.6464	1.3905	980.0
1000.0	544.58	0.0217	0.4202	0.4419	546.0	643.5	1189.6	0.7473	0.6408	1.3881	1000.0
1050.0	550.53	0.0219	0.3960	0.4179	554.0	633.6	1187.6	0.7550	0.6273	1.3822	1050.0
1100.0	556.28	0.0222	0.3738	0.3960	561.7	623.9	1185.6	0.7624	0.6141	1.3765	1100.0
1150.0	561.81	0.0224	0.3540	0.3764	569.2	614.3	1183.5	0.7695	0.6014	1.3709	1150.0
1200.0	567.14	0.0226	0.3356	0.3582	576.5	604.9	1181.4	0.7764	0.5891	1.3656	1200.0
1250.0	572.30	0.0228	0.3187	0.3415	583.6	595.6	1179.2	0.7831	0.5772	1.3603	1250.0
1300.0	577.32	0.0230	0.3029	0.3259	590.6	586.3	1177.0	0.7897	0.5654	1.3552	1300.0
1350.0	582.21	0.0232	0.2884	0.3116	597.5	577.2	1174.7	0.7962	0.5540	1.3501	1350.0
1400.0	586.96	0.0235	0.2748	0.2983	604.3	568.1	1172.4	0.8024	0.5428	1.3452	1400.0
1450.0	591.58	0.0237	0.2621	0.2858	611.0	559.1	1170.0	0.8086	0.5318	1.3404	1450.0
1500.0	596.08	0.0239	0.2502	0.2741	617.5	550.2	1167.6	0.8146	0.5212	1.3357	1500.0
1600.0	604.74	0.0244	0.2284	0.2528	630.2	532.6	1162.7	0.8262	0.5003	1.3265	1600.0
1700.0	612.98	0.0249	0.2089	0.2338	642.5	515.0	1157.5	0.8373	0.4801	1.3174	1700.0
1800.0	620.86	0.0254	0.1913	0.2167	654.7	497.2	1151.8	0.8482	0.4601	1.3083	1800.0
1900.0	628.39	0.0260	0.1754	0.2014	666.8	478.9	1145.7	0.8589	0.4402	1.2990	1900.0
2000.0	635.6	0.0265	0.1610	0.1875	679.0	460.0	1139.0	0.8696	0.4200	1.2896	2000.0
2200.0	649.2	0.0277	0.1346	0.1623	703.7	420.0	1123.8	0.8912	0.3788	1.2700	2200.0
2400.0	661.9	0.0292	0.1112	0.1404	729.4	376.4	1105.8	0.9133	0.3356	1.2488	2400.0
2600.0	673.8	0.0310	0.0895	0.1205	756.7	327.8	1084.5	0.9364	0.2892	1.2257	2600.0
2800.0	684.9	0.0333	0.0688	0.1021	786.7	272.3	1058.9	0.9618	0.2379	1.1996	2800.0
3000.0	695.2	0.0367	0.0477	0.0844	823.1	202.5	1025.6	0.9922	0.1754	1.1676	3000.0
3200.0	704.9	0.0459	0.0142	0.0601	887.0	75.9	962.9	1.0461	0.0651	1.1112	3200.0
3226.0	706.1	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	3226.0

be accompanied by a change in the boiling point, and there will be a corresponding change in the latent heat of evaporation. These values are given in Table 6.

Specific Heat. The specific heat of water, or the amount of heat (Btu) required to raise the temperature of one pound of water one degree Fahrenheit, varies with the temperature, but it is commonly assumed to be unity at all temperatures. Steam tables are based on exact values, however. The specific heat of ice at 32 F is 0.492 Btu per pound. The amount of heat required to raise one pound of water at 32 F through a known temperature interval depends on the average specific heat for the temperature range.

Sensible and Latent Heat. The heat necessary to raise the temperature of one pound of water from 32 F to the boiling point is known as the *heat of the liquid or sensible heat*. When more heat is added, the water begins to evaporate and expand at constant temperature until the water is entirely changed into steam. The heat thus added is known as the *latent heat of evaporation*.

TABLE 7. THERMAL PROPERTIES OF WATER

TEMPERATURE DEG F	SAT. PRESS. LB PER SQ IN.	VOLUME CU FT PER LB	WEIGHT LB PER CU FT	SPECIFIC HEAT
32	0.0887	0.01602	62.42	1.0093
40	0.1217	0.01602	62.42	1.0048
50	0.1780	0.01602	62.42	1.0015
60	0.2561	0.01603	62.38	0.9995
70	0.3628	0.01605	62.31	0.9982
80	0.5067	0.01607	62.23	0.9975
90	0.6980	0.01610	62.11	0.9971
100	0.9487	0.01613	62.00	0.9970
110	1.274	0.01616	61.88	0.9971
120	1.692	0.01620	61.73	0.9974
130	2.221	0.01625	61.54	0.9978
140	2.887	0.01629	61.39	0.9984
150	3.716	0.01634	61.20	0.9990
160	4.739	0.01639	61.01	0.9998
170	5.990	0.01645	60.79	1.0007
180	7.510	0.01650	60.61	1.0017
190	9.336	0.01656	60.39	1.0028
200	11.525	0.01663	60.13	1.0039
210	14.123	0.01669	59.92	1.0052
212	14.696	0.01670	59.88	1.0055
220	17.188	0.01676	59.66	1.0068
240	24.97	0.01690	59.17	1.0104
260	35.43	0.01706	58.62	1.0148
280	49.20	0.01723	58.04	1.020
300	67.01	0.01742	57.41	1.026
350	134.62	0.01797	55.65	1.044
400	247.25	0.01865	53.62	1.067
450	422.61	0.0195	51.3	1.095
500	681.09	0.0205	48.8	1.130
550	1045.4	0.0219	45.7	1.200
600	1544.6	0.0241	41.5	1.362
700	3096.4	0.0394	25.4	

RATE OF EVAPORATION

In problems of air conditioning and drying, as well as in other industrial applications of evaporation, such as cooling towers, it is desirable to determine the rate of evaporation. There are two distinct cases of evaporation. The *first* case is that in which the source of heat is primarily from the water itself and in which the air temperature may even be raised. The *second* is that in which the heat for evaporation is obtained entirely from the air itself, in which case the air is cooled and the temperature of the water remains substantially constant at the wet-bulb temperature. Both cases, however, may be reduced to a common basis of calculation. It has been found that the increase in the rate of evaporation is nearly in direct proportion to the increase in the air velocity, and that it is in direct proportion to the difference in vapor pressure between the vapor pressure of the water and the pressure of the vapor in the air.

The general formula covering the experimental data may be expressed as follows:

$$\frac{dw}{dt} = (a + bv)(e' - e) \quad (17)$$

where

$\frac{dw}{dt}$ = rate of evaporation.

a = the rate of evaporation in still air.

b = the rate of increase with velocity.

e' = the vapor pressure of the liquid.

e = the vapor pressure in the atmosphere.

v = velocity.

The only difference between case *one* and case *two* is that in case *one* the vapor pressure of the liquid is one of the known or assumed factors, being dependent upon the known temperature of the liquid, while in case *two*, e' is the vapor pressure corresponding to the wet-bulb temperature of the air.

This wet-bulb or evaporation temperature is dependent upon the dry-bulb temperature and the moisture content, or upon the total heat of the air as indicated in the previous paragraph.

The effect of air velocity depends upon whether the flow of air is parallel to the surface or perpendicular to the surface elements. For a flow of air *parallel* to a horizontal surface

$$w = 0.093 \left(1 + \frac{v}{230} \right) (e' - e) \quad (\text{approximately}) \quad (18)$$

where

w = pounds evaporated per square foot per hour.

v = velocity of atmosphere over surfaces, feet per minute.

e' = vapor pressure of the water corresponding to its temperature.

e = vapor pressure in the surrounding atmosphere.

For *transverse* flow, as across a tubular surface, the rate of evaporation is nearly doubled.

These relationships are indicated graphically on the chart, Fig. 3.

Since the difference in vapor pressures is substantially proportional to the difference between the wet- and dry-bulb temperatures (i.e., the wet-bulb depression) the rate of evaporation is also, for case *two*, substantially proportionate to the wet-bulb depression.

In case *two*, the rate of sensible heat transfer from the air to the liquid to produce evaporation is substantially the same as the rate of heat transfer with the same type of surface, without moisture being present, but with the same temperature differences. In other words, the rate of heat transfer depends upon the temperature difference only, whether the surface is wet or not. For example, it has been shown that the rate of

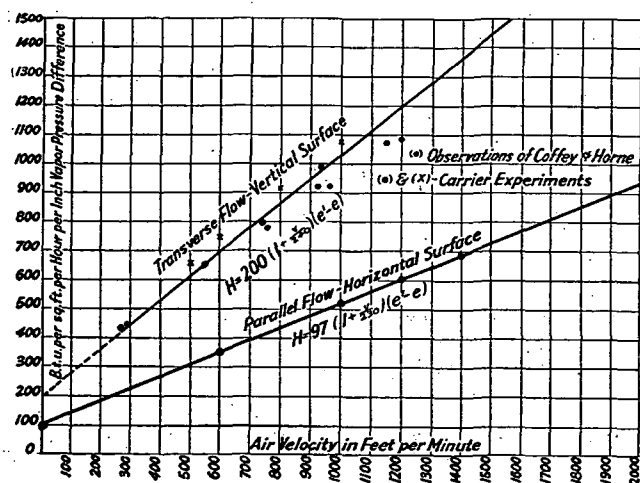


FIG. 3. HEAT TRANSMITTED BY EVAPORATION

heat transfer with air flowing across staggered coils (transverse flow) may be represented by the formula:

$$U_t = \frac{1}{0.0447 + \frac{50.66}{v}} \quad (19)$$

where

U_t = heat transfer, expressed in Btu per hour per square foot per degree difference in temperature between steam and air, for transverse flow.

At a velocity of 400 fpm, $U_t = 5.8$; at a velocity of 800 fpm, $U_t = 9.3$.

Referring to Fig. 3, showing the rate of heat transmission by evaporation for different air velocities, it will be noted that for transverse flow there are 560 Btu per hour per square foot transferred per inch difference of vapor pressure at a velocity of 400 fpm, and 910 Btu per hour per square foot per inch difference in vapor pressure at a velocity of 800 fpm. One inch of vapor pressure difference corresponds approximately to 95 deg. difference between the wet- and dry-bulb temperature. Dividing by 95,

the value of 5.9 Btu per square foot per degree difference in temperature is obtained for a velocity of 400 fpm, and 9.55 Btu per square foot for a velocity of 800 fpm.

It will be noted that for these two cases the heat transfer by evaporation per degree difference in temperature corresponds almost exactly with the heat transfer by convection coils. The similarity may be noted by comparing the formula for heat transfer in parallel flow,

$$U_p = \frac{1}{0.026 + \frac{161}{v}} \quad (20)$$

with the heat transfer by evaporation with parallel flow. The relationship will be seen to be very close in both cases and would indicate that the heat transfer by evaporation is actually brought about by a process of convection.

The difference in form of the two formulae may be due in part to errors in observation at the higher and lower velocities.

In cooling air and condensing out the moisture therefrom the heat transfer is considerably more rapid than when the air is dry and no moisture is condensed. In general the rate of heat transmission on the air side is increased an amount which is proportionate to the latent heat removed as compared with the sensible heat removed. That is, if the latent heat removed was 50 per cent of the sensible heat removed, then the conductivity of the surface in contact with the air would be increased approximately 50 per cent.

REFERENCES

- A Review of Psychrometric Charts, by C. O. Mackey (*Heating and Ventilating*, June, July, 1931).
- A New Psychrometric Chart, by C. A. Bulkeley (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926).
- Air Conditioning Applied to Cold Storage and a New Psychrometric Chart, by C. A. Bulkeley (*Refrigerating Engineering*, February, 1932).
- Air Conditioning Theory, by John A. Goff (*Refrigerating Engineering*, January, 1933).
- Rational Psychrometric Formulae, by W. H. Carrier (A.S.M.E. Transactions, Vol. 33, 1911).
- Temperature of Evaporation, by W. H. Carrier (A.S.H.V.E. TRANSACTIONS, Vol. 24, 1918).
- Principles of Engineering Thermodynamics, by Kiefer and Stuart.
- Basic Theory of Air Conditioning, by Lawrence Washington (Western Conference on Air Conditioning, San Francisco, Calif., February 9-10, 1933).
- Mixtures of Air and Water Vapor, by C. A. Bulkeley (*Refrigerating Engineering*, January, 1933).
- Temperature of Evaporation of Water into Air, by W. H. Carrier and D. C. Lindsay (A.S.M.E. Transactions, 1924).
- Chemical Engineering, by Lewis, Walker and McAdams.
- Fan Engineering, Buffalo Forge Co.
- The Psychrometric Chart, by E. V. Hill (*Aerologist*, April, May, June, 1932).

PROBLEMS IN PRACTICE

1 ● Given air at 70 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.00 in. Hg, find the weight of vapor per pound of dry air.

Pressure of saturated vapor = $e_s = 0.7387$ in. Hg (Table 5).

From Equation 5a,

$$W = 0.622 \left(\frac{0.5 \times 0.7387}{29.00 - (0.5)(0.7387)} \right)$$

$W = 0.008024$ lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

Approximate Method:

Weight of saturated vapor per pound of dry air = $W_s = 0.01574$ lb (Table 5). $0.01574 \times 0.5 = 0.00787$ lb of vapor per pound of dry air at 70 F dry-bulb and 50 per cent relative humidity.

2 ● Given air with a dry-bulb temperature of 80 F, relative humidity of 55 per cent, and a barometric pressure of 28.85 in. Hg, calculate the weight of a cubic foot of mixture.

Pressure of saturated vapor at 80 F = $e_s = 1.0316$ in. Hg (Table 5).

Pressure of the vapor in the mixture = $1.0316 \times 0.55 = 0.5676$ in. Hg.

Pressure of the dry air in the mixture = $28.85 - 0.5676 = 28.282$ in. Hg.

$pV = wR(t + 460)$ ($R = 0.753$ when partial pressure of air is expressed in in. Hg). $28.282 \times 1 = d_a \times 0.753 \times (80 + 460)$

$$d_a = \frac{28.282}{0.753 \times 540} = 0.06955 \text{ lb.} = \text{weight of dry air in 1 cu ft of the mixture.}$$

Likewise from Equation 4a,

$$d_v = \frac{0.5676}{1.21 \times 540} = 0.000868 \text{ lb} = \text{weight of vapor per cubic feet at 55 per cent relative humidity.}$$

Weight of 1 cu ft of the mixture = $0.06955 + 0.000868 = 0.070418$ lb.

3 ● Given air with a dry-bulb temperature of 75 F, a relative humidity of 60 per cent, and a barometric pressure of 28.80 in. Hg, calculate the volume of 1 lb of the mixture.

Pressure of saturated vapor at 75 F = $e_s = 0.8745$ in. Hg.

Pressure of vapor in the mixture = $0.8745 \times 0.6 = 0.525$ in. Hg.

Pressure of dry air in the mixture = $28.80 - 0.525 = 28.275$ in. Hg.

$$d_a = \frac{28.275}{0.753 \times 535} = 0.07018 \text{ lb} = \text{weight of dry air in 1 cu ft of the mixture.}$$

From Equation 4a,

$$d_v = \frac{0.525}{1.21 \times 535} = 0.000811 \text{ lb} = \text{weight of vapor per cubic feet at 55 per cent relative humidity.}$$

Weight of 1 cu ft of the mixture = $0.07018 + 0.000811 = 0.070991$ lb.

Volume of 1 lb of the mixture = $\frac{1}{0.070991} = 14.08$ cu ft.

4 ● Given saturated air at a temperature of 75 F and a barometric pressure of 29.92 in. Hg, determine the total heat of the mixture per pound of dry air.

From Equation 11 and Tables 5 and 6,

C_{pa} = mean specific heat at constant pressure of dry air = 0.24.

h_{fg} = latent heat of vaporization at the wet-bulb temperature = 1050.1 Btu per lb.

W_t = weight of water vapor mixed with each pound of dry air = 0.01873 lb.

$$\Sigma = 0.24(75 - 0) + (0.01873)(1050.1).$$

$$\Sigma = 37.67 \text{ Btu per lb of dry air.}$$

5 ● Given air at 85 F dry-bulb temperature, 75 F wet-bulb temperature, and a barometric pressure of 29.92 in. Hg; determine the total heat of the mixture per pound of dry air.

From Equation 10 and Tables 5 and 6,

$$C_{pa} = 0.24. \quad h'_{fg} = 1050.1 \text{ Btu.}$$

Relative humidity = 62.3 per cent (from psychrometric chart).

$$W = 0.622 \left(\frac{0.623 \times 1.2135}{29.92 - 0.756} \right) = 0.01612 \text{ lb of moisture per lb of dry air.}$$

$$\Sigma = 0.24(85 - 0) + 0.01612[1050.1 + 0.45(85 - 75)].$$

$$\Sigma = 37.40 \text{ Btu per pound of dry air.}$$

It will be seen from Questions 4 and 5 that the total heat is a function of the wet-bulb temperature.

6 ● It is desired to maintain a temperature of 80 F and a relative humidity of 50 per cent in a factory where the equipment gives off 6,000 Btu per hour. If the entering air is at 70 F with an average barometric pressure of 29.92 in. Hg; determine the relative humidity, and the pounds of air required per hour if there is no heat interchange between the walls, windows, or floors of the building.

Pressure of saturated vapor at 80 F = 1.0316 in. Hg (Table 5).

Pressure of vapor in the mixture = $1.0316 \times 0.5 = 0.5158$ in. Hg.

$$W = 0.622 \left(\frac{0.5158}{29.92 - 0.5158} \right) = 0.01091 \text{ lb.}$$

Pressure of saturated vapor at 70 F = 0.7387 in. Hg.

With the same specific humidity

$$0.01091 = 0.622 \left(\frac{0.7387 \times \Phi}{29.92 - (0.7387 \times \Phi)} \right)$$

$$\Phi = 69.8 \text{ per cent relative humidity at 70 F.}$$

$h = 0.24 \times 80 + 0.01091(1059.2 + 0.45 \times 80) = 31.15$ Btu per pound, the heat content of the mixture at 80 F and 50 per cent relative humidity.

$h = 0.24 \times 70 + 0.01091(1059.2 + 0.45 \times 70) = 28.70$ Btu per pound, the heat content of the mixture at 70 F and the same specific humidity.

$31.15 - 28.70 = 2.45$ Btu to be removed per pound of air.

6000 Btu = heat given off by equipment per hour.

$$\frac{6000}{2.45} = 2449 \text{ lb of air required per hour.}$$

7 ● Given 1 lb of dry air at 78 F and a barometric pressure of 29.92 in. Hg; calculate the volume. If the temperature is raised to 96 F and the volume remains constant, what will be the new pressure, P_2 , in in. Hg?

$$pV = wR(t + 460)$$

$$R \text{ (for air)} = 53.34.$$

$$W = 1 \text{ lb.}$$

P = absolute pressure, pounds per square foot.

$$V = \frac{1 \times 53.34 \times (78 + 460)}{29.92 \times 0.491 \times 144}$$

$$V = 13.57 \text{ cu ft} = \text{volume of 1 lb.}$$

$$\frac{P_2}{P_1} = \frac{T_2}{T_1}; \quad P_2 = \frac{T_2 P_1}{T_1}$$

$$P_2 = \frac{(96 + 460)(29.92 \times 0.491 \times 144)}{(78 + 460)(0.491 \times 144)} = 30.90 \text{ in. Hg.}$$

8 • Given saturated air at a temperature of 75 F and a barometric pressure of 29.92 in. Hg; determine the heat content of the mixture per pound of dry air, including the heat content of the liquid above 32 F.

From Equation 12,

$$h = 0.24(t - 0) + W(1059.2 + 0.45t).$$

where

$$h_s = 1059.2 + 0.45t \text{ (Empirical equation derived from Keenan's Steam Tables.)}$$

$$t = 75 \text{ F.}$$

$$W = 0.01873 \text{ lb of water vapor (Table 5).}$$

$$h = 0.24(75 - 0) + 0.01873(1059.2 + 0.45 \times 75).$$

$$h = 38.47 \text{ Btu per pound of dry air.}$$

9 • A building requires 50,000 cu ft of air per hour to be raised from -10 F dry-bulb and 75 per cent relative humidity to 72 F dry-bulb and 30 per cent relative humidity. Determine the amount of heat and the weight of water which it is necessary to supply per hour if the temperature of the supply water is 50 F and the barometric pressure is 28.75 in. Hg.

Assume air volume to be dry air at 70 F.

$$\text{Weight of air} = 0.075 \times 50,000 = 3750 \text{ lb per hour.}$$

From Table 5,

$$\text{Pressure of vapor in the mixture, outside air} = 0.75 \times 0.0221 = 0.0166 \text{ in. Hg.}$$

$$\text{Specific humidity, outside air} = 0.622 \left(\frac{0.0166}{28.75 - 0.0166} \right) = 0.0003589 \text{ lb.}$$

From Table 5,

$$\text{Pressure of vapor in the mixture, inside air} = 0.30 \times 0.7906 = 0.2372 \text{ in. Hg.}$$

$$\text{Specific humidity, inside air} = 0.622 \left(\frac{0.2372}{28.75 - 0.2372} \right) = 0.005174 \text{ lb.}$$

$$\text{Water to be added} = 3750(0.005174 - 0.0003589) = 18.06 \text{ lb per hour. Heat content, inside air} = 0.24 \times 72 + 0.005174(1059.2 + 0.45 \times 72) = 22.925 \text{ Btu per pound.}$$

$$\text{Heat content, outside air} = 0.24 \times (-10) + 0.0003589(1059.2 + (0.45 \times -10)) = -2.021 \text{ Btu per pound.}$$

Btu added incident to the water per pound of dry air.

$$(0.005174 - 0.0003589)(50 - 32) = 0.0867 \text{ Btu per pound.}$$

$$\text{Heat requirement per hour} = (22.925 - (-2.021 + 0.0867)) \times 3750 = 93,221 \text{ Btu.}$$

10 • Determine the amount of heat and water that must be extracted to cool 3750 lb of air (weighed dry) from 95 F and 60 per cent relative humidity to 50 F and 100 per cent relative humidity with a barometric pressure of 28.75 in. Hg.

$$\text{Pressure of vapor in the mixture, outside air} = 0.6 \times 1.659 = 0.995 \text{ in. Hg.}$$

$$\text{Specific humidity, outside air} = 0.622 \left(\frac{0.995}{28.75 - 0.995} \right) = 0.0223 \text{ lb.}$$

$$\text{Specific humidity, inside air} = 0.007626 \text{ lb.}$$

$$\text{Weight of water to be extracted per hour} = (0.0223 - 0.007626) \times 3750 = 55.03 \text{ lb.}$$

$$\text{Heat content, outside air} = 0.24 \times 95 + 0.0223(1059.2 + 0.45 \times 95) = 47.37 \text{ Btu per pound.}$$

$$\text{Heat content, inside air} = 0.24 \times 50 + 0.00764(1059.2 + 0.45 \times 50) = 20.26 \text{ Btu per pound.}$$

$$\text{Heat to be extracted} = (47.37 - 20.26) \times 3750 = 101,662 \text{ Btu.}$$

Chapter 2

REFRIGERATION

Classification of Systems, Refrigerants, Mechanical Compression Systems, Theoretical Cycle, Refrigerating Effect per Pound, Coefficient of Performance, Steam Ejector System, System Characteristic, Closed Absorption System, Open Adsorption Systems, Reverse Cycle

THE various types of refrigeration systems most commonly used for air conditioning purposes may be classified fundamentally as follows:

1. Compression Systems.
 - a. Mechanical—Reciprocating, Rotary, and Centrifugal.
 - b. Ejector.
2. Absorption Systems.
 - a. Closed.
 - b. Open.

Of these, the mechanical systems are the most extensively used at the present time and will be given complete consideration in the following discussion.

REFRIGERANTS

The common refrigerants are volatile liquids which produce refrigeration by their evaporation under reduced pressure. Factors usually influencing the choice of refrigerant are safety, chemical stability, operating pressures and adaptability for the type of system to be used.

Of the six refrigerants whose properties are listed in Tables 1 to 6, ammonia, carbon dioxide, dichlorodifluoromethane (F_{12}) and methyl chloride are used in reciprocating and rotary mechanical compression systems. Monofluorotrichloromethane (F_{11}) and water are used in centrifugal compression systems. Water is used almost to the exclusion of other refrigerants in ejector systems. Closed absorption systems may use ammonia, methyl chloride, water, or others as the refrigerant.

MECHANICAL REFRIGERATION SYSTEMS

While the mechanical refrigeration systems differ in the methods used for compression of the refrigerant vapor, they are all fundamentally similar. Refrigerant vapor, usually saturated or slightly superheated, is drawn into a compressor (see Fig. 1). It is then compressed and discharged at a higher pressure to a condenser. The vapor is condensed as it contacts a heat transfer surface over which is flowing a cooling medium such as water, air or a combination of the two. The liquid refrigerant

flows to the evaporator through an expansion valve which reduces its pressure and regulates its flow. The evaporator absorbs heat from a medium which is to be cooled. When this medium is water or brine, the evaporator is known as a water or brine cooler and the refrigeration system, if used for air cooling, is known as an indirect system. When the medium cooled is air, the evaporator is known as a direct expansion cooler and the system is known as a direct expansion system.

Fundamentally, the function of the system is to absorb heat at one temperature and *pump* it to a higher temperature, where it may be removed by an available cooling medium. In order to conserve refrigerant,

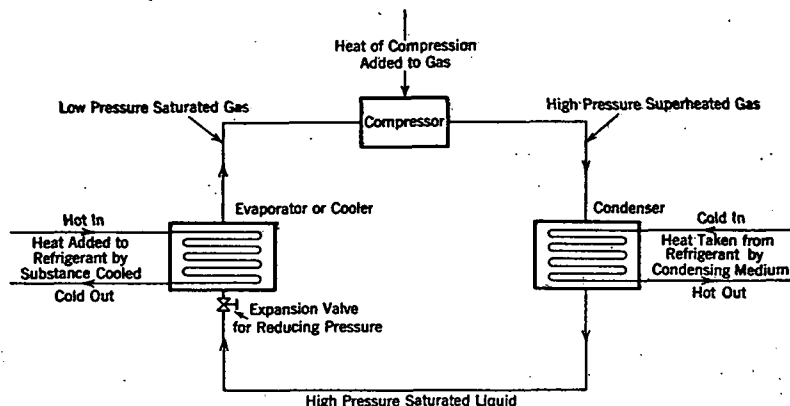


FIG. 1. MECHANICAL REFRIGERATION SYSTEM

virtually all refrigeration systems are completely closed and the same refrigerant is recirculated.

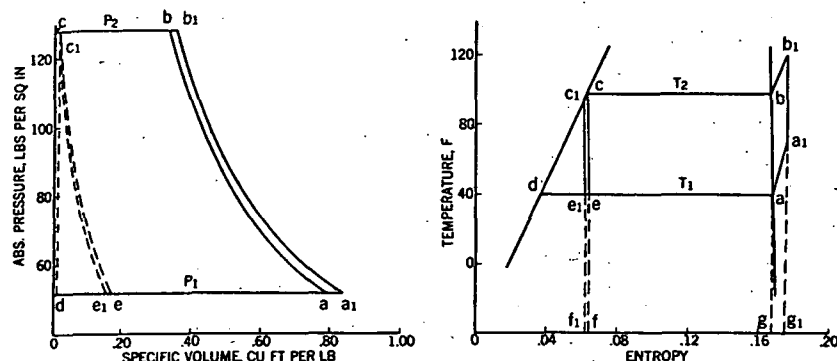
Theoretical Mechanical Refrigeration Cycle

The complete mechanical refrigeration cycle may be illustrated on the temperature-entropy diagram, and also on the pressure-volume diagram both of which are shown in Fig. 2.

Considering the theoretical cycle, saturated vapor is drawn into the compressor at *a* and compressed at constant entropy (adiabatically) and then delivered to the condenser at *b*. Condensation occurs at constant temperature T_2 from *b* to *c* with a contraction from the vapor to the liquid volume. The line *cd* represents cooling from the temperature of the condenser to that of the evaporator by an external cooling means. At the same time, the pressure is lowered to P_1 . Evaporation then occurs from *d* to *a* at temperature T_1 , completing the work cycle *abcd*. Since no external means of cooling the refrigerant liquid is normally available, the cooling is generally accomplished by evaporation of a portion of the refrigerant. Since the work of expansion is usually used up as friction in the expansion valve, this process is carried on at constant total heat, as represented by the line *ce*, on the temperature-entropy diagram. Thus the refrigerating effect is represented by an area *eagfe*. While the normal

theoretical cycle starts with saturated vapor, operation is common at a condition of superheated vapor (as at *a*₁). Moreover, expansion may start either with a mixture of liquid and vapor or with a sub-cooled liquid, as at *c*₁, with expansion to *e*₁. It is obvious that this latter is desirable as it increases the refrigerating effect. Area *a*₁*b*₁*cd**a*₁ represents the work of such a superheated cycle, while the area *e*₁*a*₁*g*₁*f*₁*e*₁ represents the refrigerating effect of the cycle with superheated vapor and sub-cooled refrigerant liquid.

It will be noted on the pressure-volume diagram the volume of the liquid is indicated by a dotted line close to and parallel to the ordinate.


 FIG. 2. THEORETICAL DICHLORODIFLUOROMETHANE (F₁₂) CYCLES

It is obvious that it is necessary to include the work of pumping the liquid as well as pumping the vapor as part of the refrigeration work.

Theoretical Work per Pound

The temperature-entropy and pressure-volume diagrams are based on one pound of the refrigerant. Likewise, the theoretical work and the refrigerating effects are conveniently based on a pound of refrigerant. The compression work per pound may be found by several methods.

The temperature-entropy method starts with state point *a*. Since the quality of *a* is known, the heat content of the vapor H_a is known, and also the entropy S_a . Since $S_a = S_b$ and with T_2 given, H_b can be determined. If W = work in foot-pounds per pound of refrigerant, then

$$W = (H_b - H_a) \times 778 \quad (1)$$

The pressure-volume method starts with state point *a*, whose pressure and specific volume are known. The work of compression is the adiabatic work of compression from P_1 to P_2 , plus the work of expelling the vapor at constant pressure P_2 minus the external work of evaporation of the vapor to volume V_1 at pressure P_1 .

$$W = \frac{n}{n-1} \times P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad (2)$$

TABLE 1. PROPERTIES OF AMMONIA

SAT. TEMP. F	ABS. PRESS. LB PER SQ IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM -40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	30.42	0.02419	9.116	42.9	611.8	0.0975	1.3352	666.8	1.4439	720.3	1.5317
5	34.27	0.02432	8.150	48.3	613.3	0.1092	1.3253	668.9	1.4339	722.7	1.5215
10	38.51	0.02446	7.304	53.8	614.9	0.1208	1.3157	670.9	1.4242	725.0	1.5115
15	43.14	0.02460	6.562	59.2	616.3	0.1323	1.3062	673.0	1.4148	727.3	1.5018
20	48.21	0.02474	5.910	64.7	617.8	0.1437	1.2969	675.0	1.4056	729.6	1.4925
25	53.73	0.02488	5.334	70.2	619.1	0.1551	1.2879	677.0	1.3965	731.9	1.4833
30	59.74	0.02503	4.825	75.7	620.5	0.1663	1.2790	678.9	1.3879	734.2	1.4744
35	66.26	0.02518	4.373	81.2	621.7	0.1775	1.2704	680.8	1.3794	736.5	1.4658
40	73.32	0.02533	3.971	86.8	623.0	0.1885	1.2618	682.7	1.3712	738.6	1.4575
45	80.96	0.02548	3.614	92.3	624.1	0.1996	1.2535	684.6	1.3630	740.9	1.4493
50	89.19	0.02564	3.294	97.9	625.2	0.2105	1.2453	686.4	1.3552	743.1	1.4412
55	98.06	0.02581	3.008	103.5	626.3	0.2214	1.2373	688.1	1.3474	745.3	1.4335
60	107.6	0.02597	2.751	109.2	627.3	0.2322	1.2294	689.9	1.3399	747.4	1.4260
65	117.8	0.02614	2.520	114.8	628.2	0.2430	1.2216	691.7	1.3326	749.5	1.4186
70	128.8	0.02632	2.312	120.5	629.1	0.2537	1.2140	693.5	1.3254	751.6	1.4114
75	140.5	0.02650	2.125	126.2	629.9	0.2643	1.2065	695.3	1.3184	753.7	1.4044
80	153.0	0.02668	1.955	132.0	630.7	0.2749	1.1991	696.6	1.3116	755.8	1.3976
85	166.4	0.02687	1.801	137.8	631.4	0.2854	1.1918	698.2	1.3048	757.9	1.3909
90	180.6	0.02707	1.661	143.5	632.0	0.2958	1.1846	699.7	1.2983	759.9	1.3843
95	195.8	0.02727	1.534	149.4	632.6	0.3062	1.1775	701.2	1.2919	761.9	1.3783
100	211.9	0.02747	1.419	155.2	633.0	0.3166	1.1705	702.7	1.2855	763.8	1.3718
105	228.9	0.02769	1.313	161.1	633.4	0.3269	1.1635	704.2	1.2793	765.7	1.3655

$$n = \frac{C_p}{C_v} = 1.3172$$

TABLE 2. PROPERTIES OF CARBON DIOXIDE

SAT. TEMP. F	ABS. PRESS. LB PER SQ IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM -40 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	305.5	0.01570	0.2904	18.8	138.9	0.0418	0.3024	153.7	0.3342	167.5	0.3612
5	332.0	0.01592	0.2661	20.3	138.8	0.0472	0.3000	153.7	0.3312	167.8	0.3582
10	360.2	0.01614	0.2437	24.0	138.7	0.0526	0.2970	153.7	0.3281	168.0	0.3550
15	390.0	0.01637	0.2236	26.7	138.5	0.0581	0.2939	153.7	0.3257	168.3	0.3517
20	421.8	0.01663	0.2049	29.4	138.3	0.0638	0.2909	153.7	0.3227	168.6	0.3489
25	455.3	0.01690	0.1879	32.3	138.1	0.0697	0.2879	153.7	0.3196	168.8	0.3464
30	490.8	0.01719	0.1722	35.4	137.8	0.0758	0.2849	153.7	0.3164	169.1	0.3441
35	529.3	0.01752	0.1577	38.5	137.3	0.0817	0.2813	153.7	0.3147	169.3	0.3415
40	567.8	0.01787	0.1444	41.7	136.7	0.0874	0.2776	153.7	0.3132	169.6	0.3391
45	609.6	0.01826	0.1321	45.0	135.9	0.0935	0.2739	153.7	0.3112	169.9	0.3365
50	653.6	0.01868	0.1205	48.4	135.0	0.1000	0.2699	153.7	0.3081	170.1	0.3342
55	700.0	0.01917	0.1096	51.9	133.7	0.1062	0.2656	153.7	0.3051	170.4	0.3320
60	748.6	0.01970	0.0994	55.5	132.1	0.1135	0.2608	153.7	0.3022	170.7	0.3297
65	799.8	0.02034	0.0899	59.4	130.2	0.1206	0.2554	153.7	0.2995	170.9	0.3277
70	853.4	0.02112	0.08040	63.7	127.5	0.1282	0.2487	153.7	0.2971	171.2	0.3257
75	909.7	0.02217	0.07072	68.4	123.7	0.1370	0.2404	153.7	0.2947	171.4	0.3237
80	968.7	0.02370	0.06064	73.9	118.7	0.1476	0.2304	153.7	0.2927	171.7	0.3220
85	1030.3	0.02620	0.05006	81.4	112.2	0.1668	0.2169	153.7	0.2909	180.0	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

$$n = \frac{C_p}{C_v} = 1.28$$

TABLE 3. PROPERTIES OF DICHLORODIFLUOROMETHANE (F₁₂)

SAT. TEMP. F	ABS. PRESS. LB PER SQ IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM -40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	23.87	0.0110	1.637	8.25	78.21	0.01869	0.17091	81.71	0.17829	85.26	0.18547
5	26.51	0.0111	1.485	9.32	78.79	0.02097	0.17052	82.29	0.17786	85.89	0.18502
10	29.35	0.0112	1.351	10.39	79.36	0.02328	0.17015	82.90	0.17747	86.51	0.18460
15	32.44	0.0112	1.230	11.48	79.94	0.02556	0.16981	83.49	0.17710	87.13	0.18420
20	35.75	0.0113	1.121	12.55	80.49	0.02783	0.16949	84.09	0.17679	87.76	0.18382
25	39.33	0.0114	1.025	13.66	81.06	0.03008	0.16920	84.67	0.17643	88.37	0.18349
30	43.16	0.0115	0.939	14.76	81.61	0.03233	0.16887	85.25	0.17612	88.97	0.18315
35	47.28	0.0116	0.863	15.88	82.16	0.03458	0.16860	85.83	0.17582	89.56	0.18285
40	51.68	0.0116	0.792	17.00	82.71	0.03680	0.16833	86.41	0.17554	90.16	0.18256
45	56.38	0.0117	0.730	18.14	83.26	0.03904	0.16808	86.96	0.17528	90.76	0.18227
50	61.39	0.0118	0.673	19.27	83.78	0.04126	0.16785	87.54	0.17505	91.38	0.18203
55	66.74	0.0119	0.622	20.41	84.31	0.04348	0.16763	88.09	0.17482	91.93	0.18181
60	72.41	0.0119	0.575	21.57	84.82	0.04568	0.16741	88.64	0.17458	92.51	0.18155
65	78.44	0.0120	0.532	22.72	85.32	0.04789	0.16721	89.18	0.17436	93.11	0.18132
70	84.82	0.0121	0.493	23.90	85.82	0.05009	0.16701	89.72	0.17417	93.66	0.18114
75	91.60	0.0122	0.458	25.08	86.32	0.05229	0.16681	90.25	0.17397	94.23	0.18092
80	98.76	0.0123	0.425	26.28	86.80	0.05446	0.16662	90.78	0.17379	94.80	0.18075
85	106.4	0.0124	0.395	27.48	87.28	0.05665	0.16644	91.27	0.17361	95.33	0.18056
90	114.3	0.0125	0.368	28.70	87.74	0.05882	0.16624	91.77	0.17344	95.86	0.18040
95	122.8	0.0126	0.343	29.93	88.19	0.06100	0.16604	92.27	0.17323	96.39	0.18020
100	131.6	0.0127	0.319	31.16	88.62	0.06316	0.16584	92.75	0.17308	96.92	0.18004
105	140.9	0.0128	0.298	32.40	89.03	0.06534	0.16564	93.24	0.17291	97.46	0.17991
110	150.7	0.0129	0.277	33.65	89.43	0.06749	0.16542	93.66	0.17274	97.93	0.17976
115	161.0	0.0130	0.258	34.90	89.80	0.06965	0.16520	94.07	0.17253	98.40	0.17955
120	171.8	0.0132	0.240	36.16	90.15	0.07180	0.16495	94.47	0.17233	98.84	0.17939

$$n = \frac{C_p}{C_v} = 1.133$$

TABLE 4. PROPERTIES OF WATER

SAT. TEMP. F	ABS PRESS. LB PER SQ IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM -32 F							
				Heat Content		Entropy		50 F Superheat		100 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	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

TABLE 5. PROPERTIES OF METHYL CHLORIDE

SAT. TEMP. F	ABS. PRESS. LB. PER SQ. IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM -40 F							
				Heat Content		Entropy		100 F Superheat		200 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	18.5	0.01628	5.00	14.5	194.0	0.032	0.424	215.6	0.467	237.2	0.507
5	20.89	0.01635	4.52	16.3	195.3	0.036	0.4225	217.0	0.464	238.5	0.503
10	23.3	0.01645	4.09	18.0	196.5	0.040	0.421	218.5	0.463	240.0	0.500
15	25.8	0.01652	3.73	19.9	198.0	0.044	0.420	219.8	0.461	241.0	0.498
20	28.3	0.01660	3.37	21.7	199.0	0.048	0.418	221.0	0.460	242.5	0.496
25	32.0	0.01669	3.07	23.5	200.0	0.052	0.417	222.1	0.459	244.0	0.494
30	35.0	0.01678	2.78	25.2	201.2	0.056	0.415	223.4	0.457	245.2	0.493
35	38.8	0.01689	2.55	27.0	202.4	0.060	0.414	224.5	0.454	246.5	0.492
40	42.8	0.01700	2.32	28.9	203.7	0.064	0.413	225.7	0.453	247.7	0.490
45	46.5	0.01710	2.13	30.7	204.5	0.068	0.412	226.8	0.451	249.1	0.488
50	51.0	0.01721	1.94	32.5	205.3	0.071	0.410	227.9	0.449	250.5	0.487
55	55.7	0.01727	1.79	34.5	206.1	0.075	0.408	228.8	0.448	251.7	0.486
60	61.0	0.01734	1.64	36.5	207.0	0.079	0.406	229.6	0.446	253.0	0.484
65	66.8	0.01746	1.51	38.5	207.5	0.083	0.404	230.5	0.443	254.3	0.483
70	72.6	0.01758	1.39	40.5	208.0	0.086	0.402	231.3	0.441	255.5	0.481
75	79.5	0.01768	1.28	42.5	208.5	0.090	0.400	232.2	0.439	256.8	0.479
80	87.0	0.01779	1.18	44.5	209.0	0.093	0.398	233.0	0.437	257.9	0.478
85	93.9	0.01790	1.09	46.3	209.5	0.097	0.396	233.8	0.435	259.2	0.477
90	102.0	0.01801	1.01	48.0	210.0	0.100	0.394	234.5	0.433	260.4	0.476
95	108.7	0.01813	0.94	50.0	210.3	0.104	0.392	235.2	0.432	261.5	0.475
100	117.0	0.01825	0.87	52.0	210.5	0.107	0.390	236.0	0.431	262.4	0.474
105	124.3	0.01836	-----	53.8	210.6	0.111	0.388	237.0	0.430	263.3	0.473
110	133.0	0.01847	-----	55.5	210.7	0.114	0.386	237.9	0.428	264.3	0.472
115	143.0	-----	-----	57.3	210.8	0.118	0.384	238.5	0.427	265.0	0.470
120	154.0	-----	-----	59.0	211.0	0.121	0.382	239.0	0.425	265.6	0.468

$$n = \frac{C_p}{C_v} = 1.20$$

It is frequently helpful to think of the compression of the vapor in terms of head. The head may be likened to a vertical column of vapor in which is located the vapor to be compressed. The compression occurs when the vapor is moved down from a level corresponding to P_1 to a new level corresponding to P_2 , in equilibrium with the surrounding vapor. If this process is carried on isentropically, the result will be the same as indicated previously. Then if h is the head in feet,

$$W = h \quad (3)$$

This relationship may easily be seen from the fact that a small difference of head dh divided by the specific volume of the vapor V is equal to the increment of pressure difference dP .

Thus, $dh = VdP$; also it can be seen by a comparison of the diagram that $VdP = SdT$, whence by integration all three of the previous expressions may be derived.

Head is very useful in considering the performance of centrifugal compressors, which merely substitute a centrifugal force for the gravity head. It is also useful in considering problems of fluid flow. In these problems, the head per degree can be obtained either by direct calculation or

TABLE 6. PROPERTIES OF MONOFLUOROTRICHOROMETHANE (F11)

SAT. TEMP. F	ABS. PRESS. LB. PER SQ. IN.	VOLUME		HEAT CONTENT AND ENTROPY TAKEN FROM -40 F							
				Heat Content		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Ht. Ct.	Entropy	Ht. Ct.	Entropy
0	2.59	0.01020	13.7	7.81	90.4	0.0178	0.1975	93.9	0.2049	97.4	0.2120
5	2.96	0.01024	12.1	8.81	91.2	0.0200	0.1974	94.7	0.2047	98.2	0.2117
10	3.38	0.01028	10.7	9.82	92.0	0.0222	0.1973	95.5	0.2045	99.0	0.2114
15	3.85	0.01032	9.53	10.8	92.8	0.0243	0.1971	96.3	0.2043	99.8	0.2111
20	4.36	0.01036	8.49	11.9	93.7	0.0264	0.1970	97.2	0.2041	100.7	0.2109
25	4.94	0.01040	7.58	12.9	94.5	0.0286	0.1969	98.0	0.2039	101.5	0.2107
30	5.57	0.01045	6.77	13.9	95.3	0.0307	0.1969	98.8	0.2038	102.3	0.2105
35	6.27	0.01049	6.08	14.9	96.1	0.0328	0.1968	99.6	0.2037	103.1	0.2103
40	7.03	0.01053	5.46	16.0	96.8	0.0349	0.1968	100.3	0.2036	103.8	0.2101
45	7.88	0.01057	4.92	17.0	97.6	0.0370	0.1967	101.1	0.2035	104.6	0.2099
50	8.79	0.01062	4.44	18.1	98.4	0.0391	0.1967	101.9	0.2034	105.4	0.2098
55	9.80	0.01066	4.02	19.1	99.2	0.0412	0.1967	102.7	0.2033	106.2	0.2097
60	10.9	0.01071	3.64	20.2	100.0	0.0432	0.1967	103.5	0.2033	107.0	0.2096
65	12.1	0.01076	3.30	21.3	100.8	0.0453	0.1967	104.3	0.2032	107.8	0.2094
70	13.4	0.01081	3.00	22.4	101.5	0.0473	0.1967	105.0	0.2032	108.5	0.2093
75	14.8	0.01086	2.74	23.5	102.2	0.0493	0.1967	105.7	0.2031	109.2	0.2092
80	16.3	0.01091	2.50	24.5	102.9	0.0513	0.1966	106.4	0.2030	109.9	0.2090
85	17.9	0.01096	2.28	25.6	103.6	0.0533	0.1966	107.1	0.2029	110.6	0.2089
90	19.7	0.01101	2.09	26.7	104.4	0.0553	0.1966	107.9	0.2028	111.4	0.2088
95	21.6	0.01106	1.918	27.8	105.1	0.0573	0.1966	108.6	0.2028	112.1	0.2087
100	23.6	0.01111	1.761	28.9	105.7	0.0593	0.1965	109.2	0.2027	112.7	0.2085
105	25.9	0.01116	1.620	30.1	106.4	0.0613	0.1965	109.9	0.2026	113.4	0.2084

$$n = \frac{C_p}{C_v} = 1.13$$

approximately by dividing the total head by the temperature difference $T_2 - T_1$. The velocity head loss can then be calculated in degrees, using the customary formula $V^2 = 2gh$.

Refrigerating Effect per Pound

The refrigerating effect per pound is computed by the same method, regardless of the type of refrigeration system. The solution is indicated on the temperature-entropy diagram of Fig. 2. Assuming that the vapor leaving the evaporator is saturated, the refrigerating effect in Btu per pound is obtained by subtracting from the heat content of the vapor at temperature T_1 , the heat content of the liquid at T_2 , or if the liquid is sub-cooled, the liquid temperature.

Thus, the refrigerating effect in Btu per pound is equal to

$$H_a - H_c = H_a - H_e \quad (4)$$

If the vapor entering the compressor is superheated or supersaturated, a correction in the heat of the vapor is made accordingly.

The unit of refrigeration is the *ton*, based on the latent heat of fusion of one ton of ice in 24 hours.

Thus one ton = 200 Btu per minute = 12,000 Btu per hour.

Coefficient of Performance

The coefficient of performance of a refrigeration system is the ratio of the refrigerating effect to the work of compression, both expressed in the same units.

The ideal or Carnot coefficient of performance depends upon the temperatures T_1 and T_2 in much the same way as the ideal efficiency of a steam engine depends upon its working temperature, with an inverse relationship.

$$\text{Ideal C. of P.} = \frac{T_1}{T_2 - T_1} \quad (5)$$

Evidently the smaller the compression range, the less power will be required to produce a given refrigerating effect.

The theoretical coefficient of performance of actual refrigerants is always less than the ideal due to the tendency of most refrigerants to superheat when compressed, and due to the heat of the liquid which must

TABLE 7. THEORETICAL COMPARISON OF VARIOUS REFRIGERANTS¹

REFRIGERANT	WORK PER LB FT LB	EQUIVA- LENT HEAD FT	GAS TEMP. LEAVING COMP. F	REFRIG. EFFECT PER LB	LB PER MIN PER TON	COEFF. OF PERFOR.	CYCLE EFF. PER CENT
Ammonia.....	53,900	53,900	209	489.8	0.408	7.06	84.8
Carbon Dioxide ^a	9,940	9,940	146	64.0	3.125	5.00	46.9
Dichlorodifluoro- methane (F ₁₂).....	6,280	6,280	134	55.74	3.638	6.90	82.9
Methyl Chloride.....	16,500	16,500	183	156.9	1.275	7.38	88.8
Monofluorotrichloro- methane (F ₁₁).....	7,050	7,050	101	74.0	2.605	7.53	90.4
Water ^b	114,300	114,300	360	1010.8	0.181	6.85	82.3
Ideal (Carnot).....						8.33	100.0

¹Evap. Refrig. Temp. = 40 F.

Suction Vapor Temp. = 65 F.

Cond. Refrig. Temp. = 100 F.

Liquid Temp. = 98 F.

^aBased on 1057 lb per sq in. (87 F) Condenser Pressure, and 85 F Liquid Temp.

^bBased on 40 F Temp.

be removed. The cycle efficiency is the theoretical C. of P. divided by the ideal for the same temperatures. The cycle efficiency usually changes as the compression temperatures change.

Comparative results of modified theoretical cycles of the refrigerants, are given in Table 7.

Practical Cycle

Fig. 3 illustrates the pressure-volume and temperature-entropy diagrams for an actual cycle. These diagrams are based upon the compressor receiving vapor superheated and upon sub-cooling of the liquid going to the evaporator. The theoretical cycle is $a_1b_1cc_1e_1a_1$. However, the vapor during compression actually follows line a_1b_2 due to superheating as a result of the inefficient work of compression. The theoretical work of compression is $a_1b_1cda_1$. Added to this is the area $b_2b_1g_1h_1b_2$ on the temperature-entropy diagram which represents the inefficient work of compression (assuming no compressor heat losses). The sum of these areas represents the total work of the compressor per pound of refrigerant, and

the ratio of theoretical cycle work to the actual work represents the overall efficiency. It should be noted that area $a_1b_2b_1a_1$ is considered as part of the inefficient work and is commonly termed the superheat loss. The refrigerating effect per pound is the same for the practical as for the theoretical cycle, working with the same sub-cooling of liquid and superheating of vapor, that is, area $e_1a_1g_1f_1e_1$.

Sources of loss which are usually recognized as reflected by the overall efficiency referring particularly to reciprocating and rotary systems, are as follows:

1. The superheat loss.

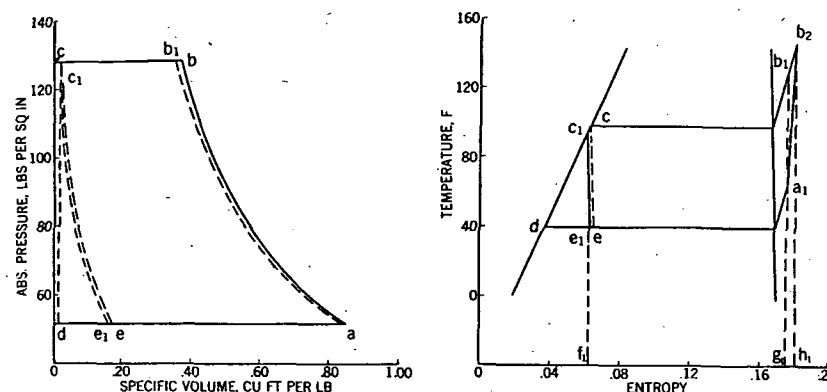
2. A pressure loss to and from the cylinder of the compressor.

(The line pressure drop between the compressor and the evaporator and condenser, respectively, is usually taken into account separately in the design of the refrigeration system.)

3. Leakage loss through valves and past pistons is quite small in most compressors.

4. With an oil soluble refrigerant, there may be an absorption loss due to absorption and re-evaporation of refrigerant in the oil of the cylinder.

5. Mechanical losses are always present and are usually a large part of the total.

FIG. 3. PRACTICAL DICHLORODIFLUOROMETHANE (F₁₂) CYCLES

Reciprocating and rotary compressors always take in less vapor than that which corresponds to the displacement. The overall volumetric efficiency is the ratio of the suction vapor volume to the piston displacement. Part of this is the re-expansion volumetric efficiency which is the volume, at suction pressure, of the usefully re-expanded vapor which was in the clearance volume. This is expressed by the following equations:

$$\text{Volumetric Efficiency} = 1 - \frac{v_c}{v_d} \times \left[\left(\frac{P_2}{P_1} \right)^{\frac{1}{n}} - 1 \right] \quad (6)$$

where

v_c = clearance volume.

v_d = cylinder displacement volume.

The balance of the overall volumetric efficiency is known as the superheat volumetric efficiency even though it includes some other sources of capacity loss.

The mechanical efficiency of a reciprocating and rotary compressor must be multiplied by the superheat volumetric efficiency to give the overall efficiency of the compressor.

$$\text{Eff. overall} = \frac{\text{Vol Eff. overall}}{\text{Vol. Eff. reexp.}} \times \text{Mech. Eff.} = \text{Super. Vol. Eff.} \times \text{Mech. Eff.} \quad (7)$$

Normally, the volumetric efficiency of a compressor varies with the

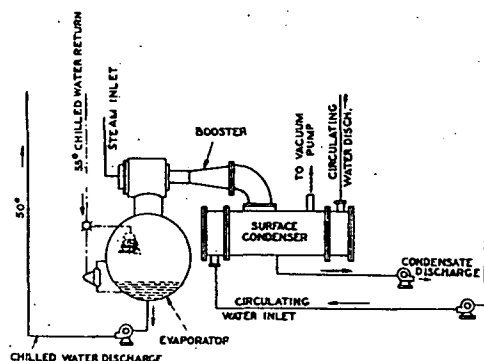


FIG. 4. STEAM EJECTOR COMPRESSION REFRIGERATION SYSTEM

ratio of compression, while the mechanical efficiency remains virtually fixed. Good standard practices for F_{12} compressors are:

Low comp. ratio = 2.5 to 1	High comp. ratio = 5 to 1
Vol eff. reexp. 94 to 96 per cent	88 to 92 per cent
Vol. eff. super. 75 to 85 per cent	73 to 77 per cent
Vol. eff. overall 70 to 81 per cent	64 to 71 per cent
Mech. eff. 75 to 85 per cent	75 to 85 per cent

These values are for one ton or larger compressors. Part of the difference expresses the change with capacity. With other refrigerants and other types of compressors there will be some further variation.

Steam Ejector Refrigeration System

Fig. 4 is a diagrammatic representation of a typical steam ejector refrigeration system. Live saturated steam is supplied to a nozzle which discharges at a high vacuum into the steam ejector. The expansion of the steam imparts a tremendous velocity to it, which entrains vapor from a flash evaporator and compresses it with a conversion from velocity to static pressure. The mixture of vapor is then delivered to a condenser. The condensate is removed by suitable means, some of it being used in some cases to supply make-up water to the evaporator circuit. The condenser is maintained free of air by means of a suitable purge. This may take the form of a water ejector if only a small amount of air is to be removed, as in a completely closed system. If a larger capacity is re-

quired, a two-stage purge is usually used, sometimes a combination steam and water and occasionally a two-stage steam purge. The vapor which is drawn from the evaporator is the result of spraying water which has been warmed by the cooling load into the flash evaporator. Evaporation of a part of this water occurs, which cools the remainder, which is then again returned to the cooling load to receive more heat.

The performance of the steam ejector may be studied theoretically by the use of the temperature-entropy diagram, Fig. 5. Unlike its usual application, however, the amount of working fluid is different for one portion of the cycle than for the other. Dry saturated steam under high pressure, for example 100 lb per square inch gage, at a , is expanded

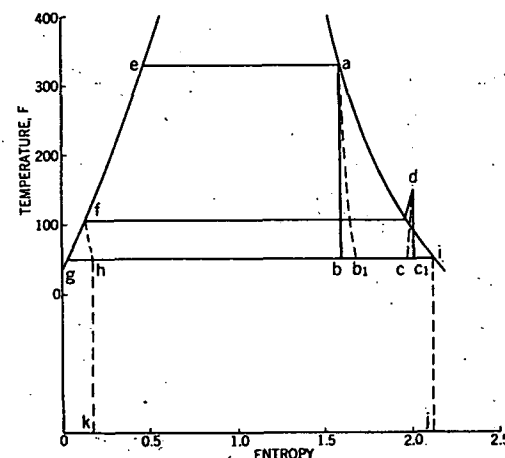


FIG. 5. STEAM EJECTOR TEMPERATURE-ENTROPY DIAGRAM

through the nozzle of the steam ejector. With 100 per cent efficiency, the expansion would occur along isentropic line ab . Actually, however, most nozzles are only about 90 per cent efficient, the real expansion being along the line ab_1 . The resulting velocity is determined from the work area ab_1gea when multiplied by the mechanical efficiency.

At the evaporator pressure or slightly below, the vapor from the nozzle mixes with virtually dry saturated vapor from the evaporator. An impact loss also occurs at this point due to the mixture of vapors at different velocities. This results in bringing the state point of the mixture to c . Compression then occurs along the line cd , the work of compression per pound being $cdfgc$. In computing the work area, however, the point c is not actually known. Therefore, the work area c_1dfgc_1 is used in expressing the efficiency of the ejector, the line c_1d being an isentropic. The losses are expressed by nozzle efficiency, impact loss and diffuser efficiency. The work of compression, however, is performed on the mass of the mixture. Thus, the available work is reduced in proportion to:

$$\frac{M_{\text{primary}}}{M_{\text{mixture}}}$$

The impact loss is commonly determined from the formula:

$$MV_{\text{primary}} + MV_{\text{secondary}} = MV_{\text{mixture}} \quad (8)$$

Common efficiencies for commercial ejectors are: nozzle efficiency 90 per cent, diffuser efficiency 60 to 70 per cent. Customary steam rates in pounds per ton are approximately as follows:

Evaporator temp.	50 F	Steam press.	100 lb	Steam press	12 lb
Condenser temp.	105 F	Steam rate	30 lb per hour per ton	Steam rate	45 lb per hour per ton.
Evaporator temp.	40 F	Steam press.	100 lb	Steam press	12 lb.
Condenser temp.	105 F	Steam rate	40 lb per hour per ton	Steam rate	70 lb per hour per ton.

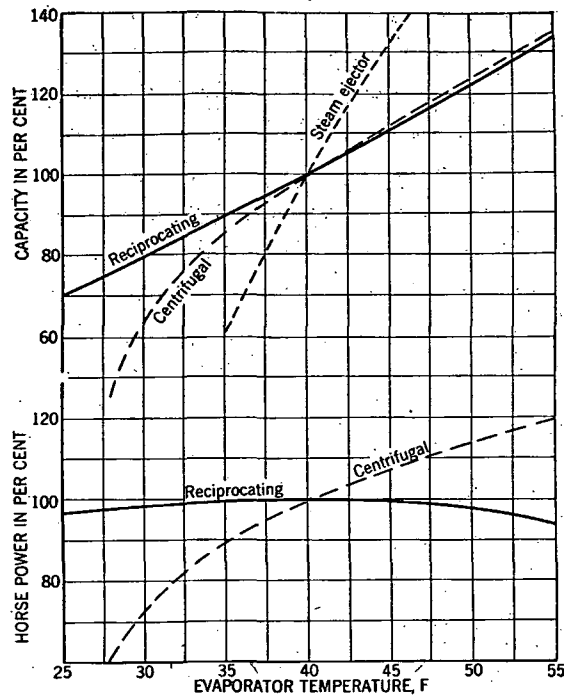


FIG. 6. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

Condenser Temperature 100 F

The steam ejector uses large amounts of condenser water and steam, and therefore this system has relative advantages when large quantities of water and steam are available at reduced costs. Characteristic curves show that the ejector performance is independent of condenser temperature up to a certain temperature, depending upon the particular design, and above this temperature the ejector breaks and loses all capacity. (See also Chapter 10.)

Characteristics of Compression Systems

The different types of compression systems have quite different characteristics of capacity and power with varying evaporator temperature and with varying condenser temperature, as will be seen from curves in Figs. 6 and 7.

The capacity of the reciprocating and rotary compressor varies slowly with a change of evaporator temperature, and the variance of power requirements, in the air conditioning range of operation, is small for a change of evaporator temperature. On the other hand, the capacity and power of the centrifugal machine vary rapidly, and the capacity of the

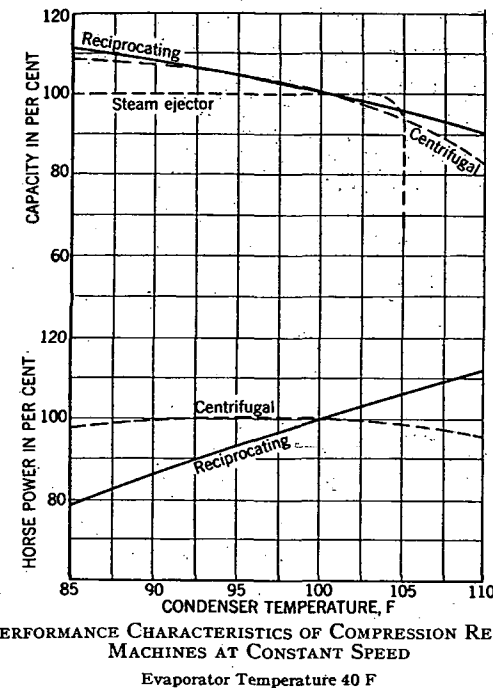


FIG. 7. PERFORMANCE CHARACTERISTICS OF COMPRESSION REFRIGERATION MACHINES AT CONSTANT SPEED

Evaporator Temperature 40 F

steam ejector also varies considerably. Thus, both these latter types tend to be more nearly self-regulating than the reciprocating and rotary compression type. On the other hand, the operating range of the latter near standard capacity is superior. Although the capacity of the reciprocating and rotary compressor is little affected by the condenser temperature, the power of the compressor is greatly affected, while the reverse is true for the centrifugal compressor. As previously indicated, the condenser temperature has no effect on the capacity of the steam ejector type of compressor until a certain point is reached, beyond which the capacity is zero. The steam consumption for the performance characteristic curves shown in Figs. 6 and 7 remains constant for all evaporator and condenser temperatures.

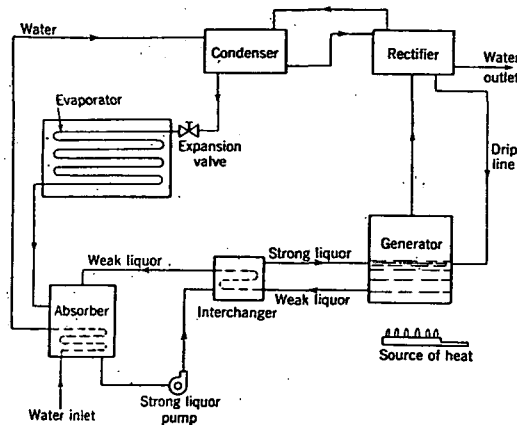


FIG. 8. CLOSED ABSORPTION SYSTEM

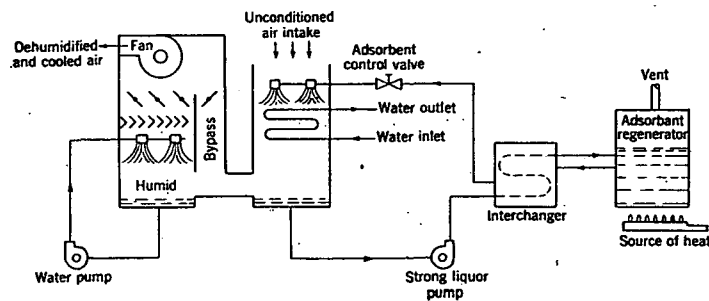


FIG. 9. OPEN LIQUID ADSORPTION SYSTEM

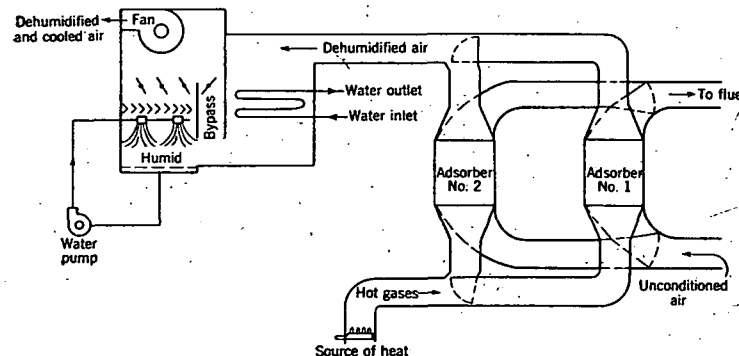


FIG. 10. OPEN SOLID MATERIAL ADSORPTION SYSTEM

CLOSED ABSORPTION SYSTEM

Years ago the closed absorption system was in favor for industrial refrigeration and ice making. It is now of little commercial importance, however, except for very special applications (notably the gas-fired refrigerator). The most usual refrigerant has been ammonia, and in some cases water has been used as an absorbent. However, a long list of refrigerant-absorbent combinations have been proposed and quite a number have been tested, either experimentally or commercially. There are some commercial installations using a dry adsorber instead of a liquid absorbent.

Fig. 8 shows a typical diagram of a closed absorption 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 is 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 residual 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.

Cooling water is ordinarily used in the absorber to remove the heat of absorption and maintain the absorptive power of the absorber at a maximum.

Like the steam ejector system, the absorption system compares most favorably when a cheap source of cooling water and steam or other heat source is available. Unlike the ejector 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.

There are innumerable variations on the arrangement shown in Fig. 8 which include multiple absorption systems, parallel absorption systems, systems using inert gas to raise the pressure in the evaporator and absorber to that of the condenser.

OPEN ADSORPTION SYSTEMS

For air conditioning installations, some experimental work and some commercial work has been done, using open adsorption refrigeration systems. The open liquid adsorption system is illustrated diagrammatically by Fig. 9. Air from the outside and from the conditioned space is drawn through an adsorbent spray which, with the air, is cooled by passing over water cooled surfaces. The adsorbent material adsorbs much of the water vapor from the air. The cooling coil removes the heat of adsorption and may cool the air below entering temperature, if the water is cold. The dehumidified and partially cooled air is then passed to a humidifier where water-vapor is added with further cooling by evaporation. A bypass may be provided so that some of the dry air bypasses the

humidifier in order to prevent the dew point of the mixture from rising too high. The air then passes through a fan and is delivered to the conditioned space. The adsorbent, rich in water vapor, is discharged through an interchanger to a regenerator, from which it again flows to the adsorber.

The adsorbent materials most commonly considered are solids which are put into solution as brines (see also Chapter 10). The difficulty with most adsorbents is that their adsorptive power is not great enough to dehumidify the air sufficiently. Thus, they have to be used inefficiently or do an inadequate job of dehumidification.

Fig. 10 is a diagrammatic representation of an open solid material adsorption system. The spray and cooling coil of the open adsorber is replaced by a bed of adsorptive material and a separate water cooling coil, and a second section of adsorber must be provided for alternate reactivation while the first is adsorbing. A set of dampers or a rotary device must be arranged to connect the beds alternately to the air to be conditioned and to the hot gases for regeneration. Otherwise the liquid adsorption and solid adsorption systems are identical.

The most efficient solid adsorption systems use some internal means of heating the adsorber bed before activation, and cooling it after activation, and other means may also be employed for improving the performance.

These systems are especially affected by the temperature of the cooling water, since if the air leaving the cooling surface is at a higher temperature than the air entering, additional work must be done in the generator or activator in order to compensate for the extra sensible heat of the leaving air. On the other hand, if the leaving temperature is lower than the entering temperature, the reverse is true. When the cooling water temperature is low, similar economies may, however, be obtained with compression systems by the use of an air pre-cooling coil.

From this discussion it may be seen that these systems function to best advantage on a high air temperature, and on one in which the initial moisture content is high. Thus the use of high room temperatures and comparatively large amounts of outside air are encouraged in connection with these systems. In order to offset the effect of high air temperature, some effort is made to keep the humidity lower than usual. The ratio of the heat input to the refrigerant output, where the temperature of the air entering and leaving the adsorber cooler is not widely different, may vary from 4 to 1 for some of the solid adsorption systems, to as low as 1.5 to 1 for some of the liquid adsorption systems.

THE REVERSE CYCLE

The idea of heating by the reverse refrigeration cycle has captured the imagination of many people and has been much discussed. In principle, heat is absorbed in an evaporator from some available source of heat, pumped to a higher temperature and delivered to a condenser. 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 work of compression in relation to the heat delivered is:

$$\frac{T_2}{T_1 - T_2}$$

where

T_1 = absolute temperature of evaporator.

T_2 = 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 4 or 5 times practically, as the work put in. 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 high 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 theoretical 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 at present available for the reverse cycle in comparison with that 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 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.

PROBLEMS IN PRACTICE

1 • Dichlorodifluoromethane (F_{12}) at a saturated temperature of 30 F but superheated 25 F to 55 F is compressed to a saturation temperature of 90 F with a compressor having an overall efficiency of 70 per cent. It leaves the condenser sub-cooled 5 F. What is the: a. Work per pound; b. Refrigerating effect; c. Pounds per minute per ton; d. Horsepower per ton; e. Heat rejected to condenser neglecting radiation; f. Equivalent discharge temperature?

$$a. \frac{1.151}{0.151} \times 43.16 \times 144 \times 0.939 \times \frac{515}{490} \left[\left(\frac{114.3}{43.16} \right)^{\frac{0.151}{1.151}} - 1 \right] \times \frac{1}{0.70} = 9000 \text{ ft-lb.}$$

(By Pressure—Volume Method).

$$\text{Check: } (93.35 - 85.25) \times \frac{1}{0.7} = 11.6 \text{ Btu per pound} = 9000 \text{ ft-lb. (By Heat}$$

Content Method).

$$b. 95.25 - 27.48 = 57.77 \text{ Btu per pound.}$$

$$c. \frac{200}{57.77} = 3.46 \text{ lb per minute per ton.}$$

$$d. 9000 \times \frac{3.46}{33,000} = 0.944 \text{ hp per ton.}$$

$$e. 200 + 0.944 \times 42.5 = 240 \text{ Btu per minute per ton.}$$

$$\text{Check: Heat content leaving compressor} = 85.25 + 11.6 = 96.85 \\ (96.85 - 27.48) 3.46 = 240 \text{ Btu per minute per ton.}$$

$$f. \text{ Heat Content} = 96.85.$$

$$\text{Pressure} = 114.3 \text{ lb per square inch.}$$

$$\text{From Table 3, Temp.} = 146 \text{ F.}$$

2 • If the velocity of vapor in the suction pipe is 50 fps and there are 10 velocity heads lost between evaporator and compressor, what is the saturation temperature at the evaporator?

$$h = \frac{V_s^2}{2g}$$

$$h = 10 \times \left(\frac{50}{2g} \right)^2 = 388 \text{ ft head.}$$

$$\text{Head per degree} = (47.28 - 43.16) \times \left(\frac{0.863 + 0.792}{2} \right) \times \frac{144}{5} = 98.4 \text{ ft.}$$

$$\text{Check: Head per degree} = 9000 \times \frac{0.7}{(90 - 30)} = 105 \text{ ft. (Approx.)}$$

$$\text{Temperature} = 30 + \frac{388}{98.4} = 34 \text{ F.}$$

3 • If the refrigerant is sub-cooled to 70 F, what is the effect on: a. Work per pound; b. Refrigerating effect; c. Horsepower per ton?

a. No effect on work per pound.

$$b. (85.25 - 23.90) = 61.35.$$

$$61.35 - 57.77 = 3.58 \text{ Btu increase.}$$

$$\frac{3.58}{57.77} = 6.2 \text{ per cent increase.}$$

$$c. \frac{200}{61.35} = 3.27 \text{ lb per minute per ton.}$$

$$9000 \times \frac{3.27}{33,000} = 0.891 \text{ hp per ton.}$$

$$0.944 - 0.891 = 0.053 \text{ hp decrease.}$$

$$\frac{0.053}{0.944} = 5.6 \text{ per cent decrease.}$$

4 • What is the approximate change in capacity of the following types of systems per degree at 40 F: a. Reciprocating; b. Centrifugal; c. Ejector?

a. 2.5 per cent.

b. 3.0 per cent.

c. 7.5 per cent.

5 • a. Which type of system will maintain the most uniform evaporator temperature with change of load?

b. Which system will maintain the most uniform load with change of evaporator temperature?

From Fig. 6.

a. Steam ejector.

b. Reciprocating and rotary.

6 • a. What is the velocity of steam expanding from 100 lb per square inch gage, saturated to 0.0178 lb per square inch absolute, corresponding to 50 F if the nozzle has an efficiency of 90 per cent?

b. What is the velocity of the mixture of this steam with one-third the mass of entrained steam moving at 300 fps?

$$a. V = \sqrt{778 \times 2g \times 0.9 \times (1190.7 - 815.4)}$$

$$V = 4110 \text{ fps.}$$

$$b. V_{\text{mix}} = \frac{3 \times 4110 + 1 \times 300}{4} = 3158 \text{ fps.}$$

7 • If air entering an open adsorption system at 80 F and 50 per cent relative humidity is dehumidified and cooled to a temperature of 90 F and 12 per cent relative humidity, how much air is cooled per ton of refrigeration and what is the latent heat of the water which is adsorbed?

Entering Conditions

66.6 F WB.

30.85 Btu Total Heat

76.0 Grains per Pound

Leaving Conditions

59.1 F WB.

25.59 Btu Total Heat

24.5 Grains per Pound

$$\text{cfm} = \frac{200 \times 13.5}{(30.85 - 25.59)} = 513 \text{ cfm per ton.}$$

$$L = \frac{513 \times (76.0 - 24.5) \times 1044}{13.5 \times 7000} = 292.5 \text{ Btu.}$$

$$\text{Check: } L = 200 + \left(\frac{513 \times 0.2415 \times 10}{13.5} \right) = 291.6 \text{ Btu.}$$

8 • If air entering an open adsorption system at 80 F and 50 per cent relative humidity is cooled to 75 F and 12 per cent relative humidity, how much air is required per ton and what is the latent heat of the water which is absorbed?

Entering Conditions

66.6 F WB

30.85 Btu Total Heat

76.0 Grains per Pound

Leaving Conditions

50.3 F WB

20.35 Btu Total Heat

14.4 Grains per Pound

$$\text{cfm} = \frac{200 \times 13.5}{(30.85 - 20.35)} = 257 \text{ cfm per ton.}$$

$$L = \frac{257 \times (76.0 - 14.4) \times 1044}{13.5 \times 7000} = 175.2 \text{ Btu.}$$

$$\text{Check: } L = 200 - \left(\frac{257 \times 0.2415 \times 5}{13.5} \right) = 177.0 \text{ Btu.}$$

9 • Why is it possible for a reversed cycle refrigeration unit to show a better performance in heating operation than when operating during the cooling season?

In normal use a refrigerating machine is arranged to remove heat and the heat removed is thrown away. The driving energy is converted into heat, most of which is added to the heat removed and is also thrown away.

In the reversed refrigeration cycle the heat removed is not thrown away, but, together with the heat converted from the driving energy, it is utilized to heat the building.

10 ● In a large ice plant, for approximately 147,000 Btu equivalent input to the motor driving the ammonia compressor a ton of ice is produced, in the production of which 288,000 Btu are removed from one ton of water at 32 F to transform it to ice at 32 F. What is the ratio of the heat removed in this process to the input of the motor?

In this case the heat removed, or pumped from the water to the air is approximately 1.96 times the heat equivalent of the input to the motor.

11 ● If the cycle is reversed in Question 10 and the heat is transferred or pumped from the air to the water, what is the ratio of the heat removed in the process to the motor input?

In this case 2.96 Btu is put into the water for each Btu equivalent input to the motor, for the total heat put into the water now represents the heat pumped plus the heat converted from the driving energy, or motor input.

Chapter 3

VENTILATION AND AIR CONDITIONING STANDARDS

Vitiation of Air, Heat Regulation in Man, Effects of Heat, Effects of Cold, Temperature Changes, Acclimatization, Warmth and Comfort, Effective Temperature, Comfort Chart, Comfort Line, Comfort Zone, Application of Comfort Chart, A.S.H.V.E. Ventilation Standards, Natural and Mechanical Ventilation, Recirculation, Ultra-Violet Radiation and Ionization, Heat and Moisture Losses

VENTILATION is defined in part as "the process of supplying or removing air by natural or mechanical means to or from any space." (See Chapter 44.) 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 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. More particularly, it includes the simultaneous control of temperature, humidity, movement, and purity of the air. The term is broad enough to embrace whatever other additional factors may be found desirable for maintaining the atmosphere of occupied spaces at a condition best suited to the physiological requirements of the human body.

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. Moreover, according to latest researches¹, there is a marked decrease in both positive and negative ions in the air of occupied rooms.

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are much too small even under the worst conditions. The amount of carbon dioxide in air is often used in ventilation work as an index of odors of human origin, but

¹See A.S.H.V.E. research paper entitled 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. 37, 1931).

the information it affords rarely justifies the labor involved in making the observation². 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 and transpired air is odorous and offensive, and it is capable of producing loss of appetite and a disinclination for physical activity. These reasons alone, whether æsthetic or physiological, are sufficient to warrant a desire for proper air conditions.

A certain part of the dissemination of disease which occurs in confined spaces is caused by the emission of pathogenic bacteria from infected persons. Infections by droplets from coughing and sneezing constitute a limited mode of transmission in the immediate vicinity of the infected person. Experiments have shown that the mouth spray is a coarse rain which settles down quickly. The contamination is local and the problem is considered to be largely one of contact infection rather than air-borne infection.

The primary factors in air conditioning work, in the absence of any specific contaminating source, are temperature, humidity, air movement and body odors. As compared with these physical factors, the chemical factors are, as a general rule, of secondary importance.

HEAT REGULATION IN MAN

The importance of temperature, humidity and air movement arises from the profound influence which these factors exert upon body temperature, comfort and health. Body temperature is a resultant of the balancing action between its heat production and its heat loss. The heat resulting from the combustion of food within the body maintains its temperature well above that of the surrounding air. At the same time, heat is constantly lost from the body by radiation, conduction 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 healthy persons this takes place automatically by the action of the heat regulating mechanism.

According to the general view, special areas in the skin are sensitive to temperature. Nerve courses 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 mechanisms of adjustment are complex and little understood at the present time. Coordination of these different mechanisms seems to vary greatly with different air conditions.

¹Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).

With rising air temperatures up to 75 F or 80 F, metabolism, or internal heat production, is decreased³, probably by an inhibitory action on heat producing organs, especially the adrenal glands, which seem to exert the major influence on basic combustion processes in the body. The blood capillaries in the skin become dilated by reflex action of the vasomotor nerves, allowing more blood to flow into the skin, and thus increase its temperature and consequently its heat loss. The increase in peripheral circulation is at the expense of the internal organs. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin, where it is evaporated. This method of cooling is the most effective of all, as long as the humidity of the air is sufficiently low to allow for evaporation. In high humidities, where the difference between the dew-point temperature of the air and body temperature is not sufficient to allow rapid evaporation, equally good results may be obtained by increasing the air movement, and hence the heat loss by conduction and evaporation.

In cold environments, in order to keep the body warm there is an actual increase in metabolism brought about partly by voluntary muscular contractions (shivering) and partly by an involuntary reflex upon the heat producing organs. The surface blood vessels become constricted, and the blood supply to the skin is curtailed by vasomotor shifts to the internal organs in order to conserve body heat.

EFFECTS OF HEAT

Although the human organism is capable of adapting itself to variations in environmental conditions, its ability to maintain heat equilibrium is limited. The heat regulating center fails, for instance, if the external temperature is so abnormally high that bodily heat cannot be eliminated as fast as it is produced. Part of it is retained in the body, causing a rise in skin and deep tissue temperature, an increase in the heart rate, and accelerated respiration. (See Table 1.) In extreme conditions, the metabolic rate is markedly increased owing to the excessive rise in body temperature⁴, and a vicious cycle results which may eventually lead to serious physiologic damage.

Examples of this are met with in unusually hot summer weather and in hot industries where the radiant heat from hot objects renders heat loss from the body by radiation and convection impossible. Consequently, the workers depend entirely on evaporation for the elimination of body heat. They stream with perspiration and drink liquids abundantly to replace the loss.

One of the most deleterious effects of high temperatures is that 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 believed that the feeling of lassitude and discomfort experienced is due to the anæmic condition of the brain.

³Heat 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).

⁴Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (*The American Journal of Physiology*, Vol. 88, No. 3, April, 1929).

TABLE 1. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK^a

EFFECTIVE TEMP.	ACTUAL CREEK TEMP (DEG FAHR)	MEN AT REST			MEN AT WORK 90,000 FT-LB OF WORK PER HOUR			
		Rise in Rectal Temp (Deg Fahr per Hour)	Increase in Pulse Rate (Beats per Min per Hour)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft-lb)	Rise in Body Temp (Deg Fahr 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	103 ^b	2.7
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

^aData by A.S.H.V.E. Research Laboratory.^bComputed value from exposures lasting less than one hour.

The stomach loses some of its power 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 tract⁵. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather. The victim may lose appetite and suffer from indigestion, headache and general enervation, which may eventually lead to a premature old age.

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 lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, Moss⁶ recommends the addition of 6 grams of sodium chloride and 4 grams of potassium chloride to a gallon of water.

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show clearly 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.

EFFECTS OF COLD

The action of cold on human beings is not well known. Cold affects the human organism in two ways: (1) through its action on the body as a whole, and (2) through its action on the mucous membranes of the upper respiratory tract. Little exact information is available on the latter.

On exposure to cold, the loss of heat is increased considerably and only

⁵Influence of Effective Temperature upon Bactericidal Action of Gastro-Intestinal Tract, by Arnold and Brody (*Proceedings Society Exp. Biol. Med.* Vol. 24, 1927, p. 832).

⁶Some Effects of High Air Temperatures upon the Miner, by K. N. Moss (*Transactions Institute of Mining Engineers*, Vol. 66, 1924, p. 284).

within certain limits is compensation possible by increased heat production and decreased peripheral circulation. The rectal temperature often rises upon exposure to cold but the pulse rate and skin temperature fall. The blood pressure increases, owing to constriction in the peripheral vessels and to thickening of the blood. The subcutaneous tissues and muscles form reservoirs for storing the water which leaves the blood. In extremely cold atmospheres compensation becomes inadequate. The body temperature falls and the reflex irritability of the spinal cord is markedly affected. The organism may finally pass into an unconscious state which ends in death.

Cannon showed that excessive loss of heat is associated with increased activity of the adrenal medulla⁷. The extra output of adrenin hastens heat production which protects the organism against cooling. Bast⁸ found a degeneration of thyroid and adrenal glands upon exposure to cold.

Effects of Temperature Changes

A moderate amount of variability in temperature is known to be beneficial to health, comfort, and the performance of physical and mental work. On the other hand, extreme changes in temperature, such as those experienced in passing from a warm room to the cold air out of doors, appear to be harmful to the tissues of the nose and throat which are the portals for the entry of respiratory diseases.

Experiments show that chilling causes a constriction of the blood vessels of the palate, tonsils and throat, which is accompanied by a fall in the temperature of the tissues. On rewarming, the palate and throat do not always regain their normal temperature and blood supply. This anæmic condition favors bacterial activity and it is believed to play a part in the inception of the common cold and other respiratory diseases. It is believed that the lowered resistance is due to a diminution in the number and phagocytic activity of the leucocytes (white blood cells) brought about by exposure to cold and by changes in temperature.

Sickness records in industries seem to strengthen this belief. The Industrial Fatigue Research Board of England⁹ found that in workers exposed to high temperatures and to changes in temperature, namely, steel melters, puddlers, and tin-plate millmen, there is an excess of all sickness, the excess among the puddlers being due chiefly to respiratory diseases and rheumatism. The causative factor was not the heat itself but the sudden changes in temperature to which the workers were exposed. The tin-plate millmen who were not exposed to chills, since they work almost continuously throughout the shift, had no excess of rheumatism and respiratory diseases. On the other hand, the blast-furnacemen, who work mostly in the open, showed more respiratory sickness than the steel workers. This experience in British factories is well in accord with the findings in American industries¹⁰. According to these data the highest

⁷Studies on the Condition of Activity of Endocrine Glands, by W. B. Cannon, A. Guerido, S. W. Britton and E. M. Bright (*American Journal of Physiology*, Vol. 79, 1926, p. 466).

⁸Studies in Exhaustion Due to Lack of Sleep, by T. H. Bast, J. S. Supernaw, B. Lieberman and J. Munro (*American Journal of Physiology*, Vol. 85, 1928, p. 135).

⁹Fatigue and Efficiency in the Iron and Steel Industry, by H. M. Vernon (*Industrial Fatigue Research Board, Report No. 5*, 1920, London).

¹⁰Iron Foundry Workers Show Highest Percentage of Deaths from Pneumonia (*Statistical Bulletin, Metropolitan Life Insurance Company*, 1928).

pneumonia death rate is associated with dust, extreme heat, exposure to cold, and to sudden changes in temperature.

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 the present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and it 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 averaging 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. All this seems to indicate that adaptation to a climate averaging between 60 and 80 F is a very primitive trait¹¹.

Within these limits, however, there does occur a definite adaptation to external temperature level. People and animals raised under conditions of tropical moist heat have a lower rate of heat production than do those who grow up in cooler environments. This causes them to stand chilling poorly as they are unable to quickly 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 adrenal activity has slowed down. Within a couple of years, however, they find themselves standing the heat much better and disliking cold. They become acclimated by a definite change in the combustion level within the body¹².

In certain individuals the psychologic factor is more powerful than acclimatization. A fresh air fiend may suffer in a room with windows closed regardless of the quality of the air. As a matter of fact, instances are known in which paid subjects refused to stay in a windowless but properly conditioned experimental chamber because the atmosphere felt suffocating to them upon entering the room.

WARMTH AND COMFORT

The temperature, humidity, and motion of the air, and the radiation between a person and surrounding hot or cold surfaces, taken together, determine his feeling of warmth and influence his elimination of body heat. In other words, the temperature sensations of the human body depend not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by

¹¹Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1924.

¹²Air Conditioning in its Relation to Human Welfare, by C. A. Mills, M.D. (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934).

a wet-bulb thermometer. 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.

On the other hand, in cold environments an increase in humidity produces a cooler sensation. The dividing line at which humidity has no effect upon comfort varies with the air velocity and is about 46 F (dry-bulb) for still air and about 51, 56 and 59 F for air velocities of 100, 300 and 500 fpm, respectively.

Thermo-Equivalent Conditions

Combinations of temperature, humidity and air movement which produce the same feeling of warmth are called thermo-equivalent conditions. A series of tests^{13, 14, 15} has been carried out in the psychrometric rooms of the A.S.H.V.E. Research Laboratory, Pittsburgh, in order to determine the equivalent conditions met with in general air conditioning work. These show that this newly-developed scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also determines the physiological effects on the body induced by heat and cold. For this reason, it is called the *effective temperature* scale or index.

Effective temperature is an index of warmth or cold. It is not in itself an index of comfort, as it is often assumed to be, nor are the effective temperature lines necessarily lines of equal comfort. This is true because, in determining this index, the subjects compared not the relative comfort, but rather the relative warmth or cold of various air conditions. Moist air at a comparatively low temperature, and dry air at a higher temperature may each feel as warm as air of an intermediate temperature and humidity, but the *comfort* experienced in the three air conditions would be different, although the effective temperature is the same.

Under extreme humidity conditions there seems to be a difference between sensations of absolute comfort and of the proper degree of warmth. In other words, human beings are not necessarily comfortable when the air is neither *too warm* nor *too cold*. Air of proper warmth may, for instance, contain excessive water vapor, and in this way interfere with the normal physiologic loss of moisture from the skin, leading to damp skin and clothing and producing more or less discomfort; or the air may be excessively dry, producing appreciable discomfort to the mucous membrane of the nose and to the skin which dries up and becomes chapped from too rapid loss of moisture. According to the comfort experiments first conducted at the A.S.H.V.E. Laboratory¹⁶ in the U. S. Bureau of Mines, Pittsburgh, and later studies at the Harvard School of Public Health¹⁷ in Boston, effective temperature appears to be a fair index of comfort also, particularly within a humidity range of 30 to 60 per cent, approximately.

¹³Determining Lines of Equal Comfort, by F. C. Houghten and C. P. Yagloglou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361).

¹⁴Cooling Effect on Human Beings by Various Air Velocities, by F. C. Houghten and C. P. Yagloglou (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, p. 193).

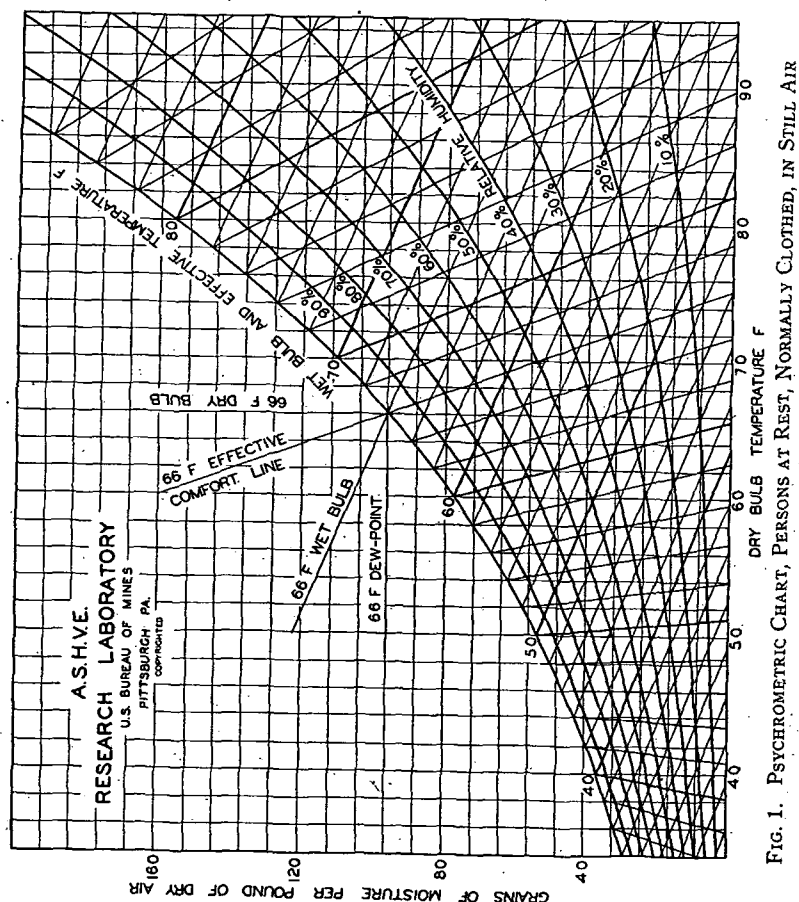
¹⁵Effective Temperature with Clothing, by C. P. Yagloglou and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 31, 1925, p. 89).

¹⁶Determination of the Comfort Zone With Further Verification of Effective Temperatures Within This Zone, by F. C. Houghten and C. P. Yagloglou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361).

¹⁷The Summer Comfort Zone; Climate and Clothing, by C. P. Yagloglou and Philip Drinker (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

Definition of Effective Temperature

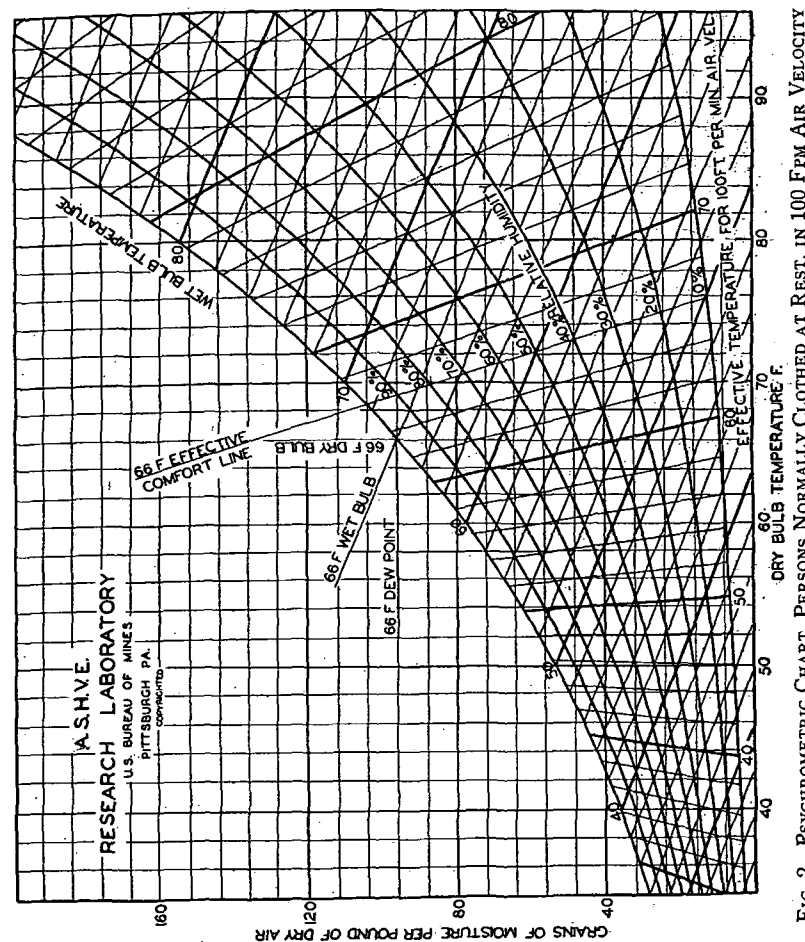
Briefly, *effective temperature* may be defined as an arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is not a true temperature of the air but an index which combines temperature, humidity and air motion in a single value. The numerical value of the



effective temperature index for any given air condition is fixed by the temperature of saturated air which, at a velocity or turbulence of 15 to 25 fpm, induces a sensation of warmth or cold like that of the given condition. Thus, any air condition has an effective temperature of 65 deg when it induces a sensation of warmth like that experienced in practically still air at 65 F saturated with moisture.

In all reports of the A.S.H.V.E. Research Laboratory, the term *still air*

signifies the minimum air movement it was possible to obtain in the Laboratory's psychrometric chamber. Actually, the air motion was between 15 and 25 fpm in all experiments, without qualification, as measured by the Kata thermometer. This was not a linear movement of air but it represented the turbulence or eddy currents produced by the air



change. Even in tightly sealed rooms, the natural air movement is not likely to fall below 10 fpm so long as there is a temperature or pressure difference between the air inside and that outside the room.

The relation of temperature and humidity to comfort for persons normally clothed and at rest in still air is given in Fig. 1, while the effect of various air motions upon comfort for persons normally clothed at rest for velocities of 100 and 300 fpm are given in Figs. 2 and 3. The difference

between the effective temperature for still air and for moving air, of any velocity, represents the cooling resulting from that air velocity. These charts apply to average normal and healthy persons adapted to American living and working conditions. They are limited to sedentary or light muscular activity, and to rooms heated by the usual American convection methods (warm air, central fan and direct hot water and steam heating

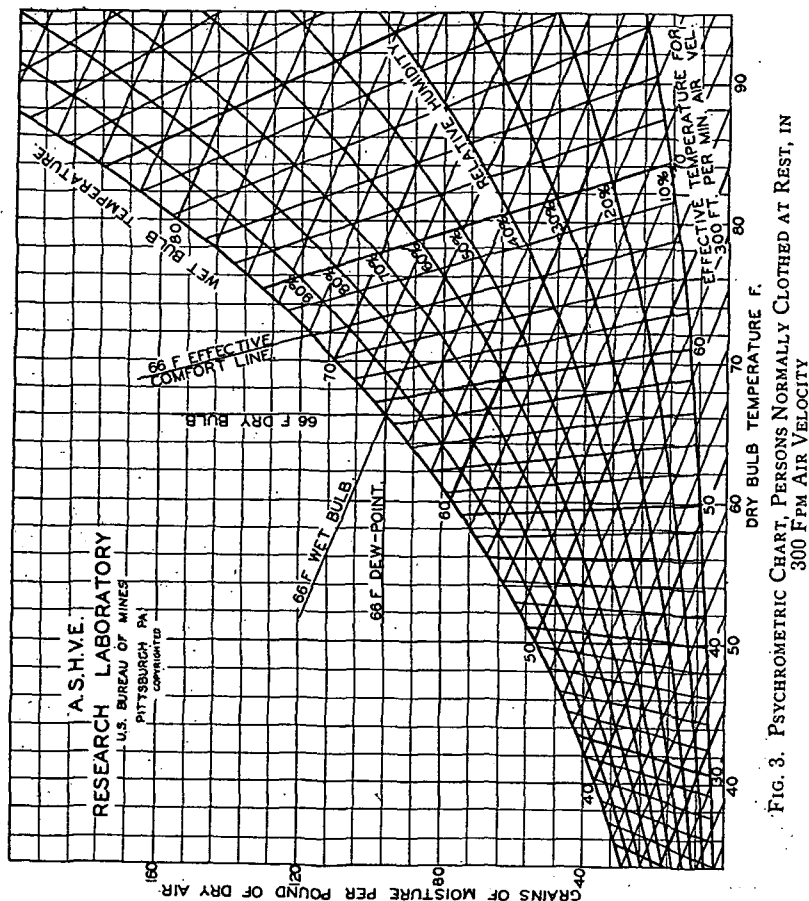


FIG. 3. PSYCHROMETRIC CHART, PERSONS NORMALLY CLOTHED AT REST, IN 300 FPM AIR VELOCITY

systems) in which the difference between the air and wall surface temperatures may not be great. The charts do not apply to rooms heated by radiant methods such as the British panel system, open coal fires, and the like. They will probably not apply with adequate accuracy to races other than the white or perhaps to inhabitants of other countries where the living conditions, climate, heating methods, and clothing are materially different from those of the subjects employed in experiments at the Research Laboratory.

If an occupant of a room loses heat by radiation to large wall or glass surfaces at lower temperatures, the air within the room must be maintained at a higher temperature to compensate for this effect in order to give the same feeling of warmth. The results of a recent study¹⁸ by the A.S.H.V.E. Laboratory, shown in Fig. 4, indicate that in poorly insulated buildings this effect may become of considerable importance. Thus an occupant of a room having inside wall surface temperatures of 55 F on three sides will require an air temperature of 74 F to have the same feeling of warmth he would experience in a warm-wall room with air at 70 F. A

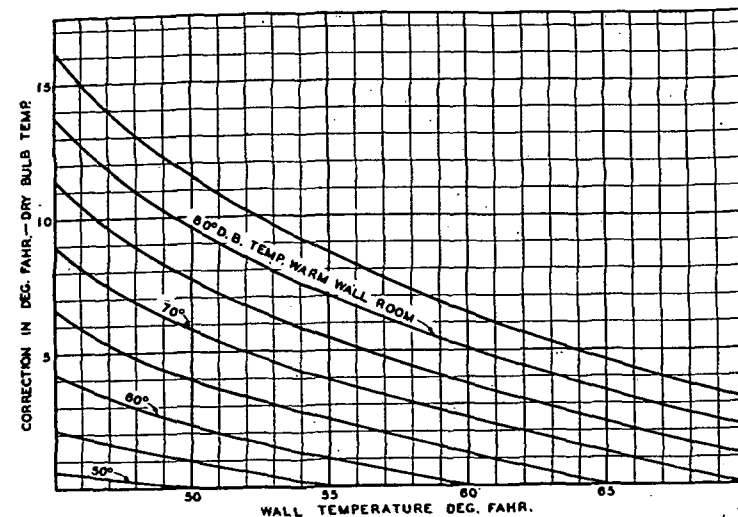


FIG. 4. CORRECTION TO VARIOUS DRY-BULB TEMPERATURES IN A WARM-WALL ROOM FOR THE SAME FEELING OF WARMTH IN ROOMS HAVING THREE COLD WALLS. TEMPERATURES INDICATED BY SHIELDED THERMOMETERS 30 IN. ABOVE THE FLOOR

wall consisting of 8-in. brick and plaster, with 16 F outside air temperature and 70 F inside air temperature, will have an inside surface temperature of 55 F. The reverse effect will be experienced by occupants of rooms having extensive high-temperature surfaces in them. In such cases, a lower air temperature is required to compensate for heat radiated to the occupant.

The effective temperature index for persons doing medium or heavy muscular work, in still air, has also been determined at the A.S.H.V.E. Research Laboratory¹⁹.

OPTIMUM AIR CONDITIONS

No single comfort standard can be laid down which would meet every need. There is an inherent individual variation in the sensation of

¹⁸Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1933, p. 53).

¹⁹Effective 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).

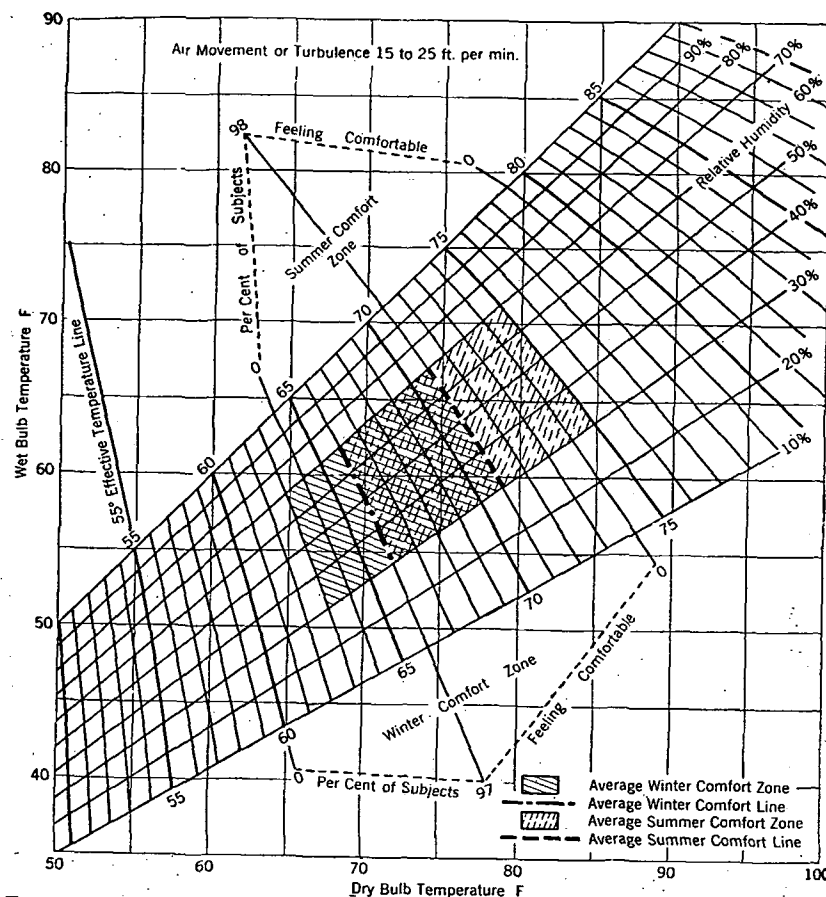


FIG. 5. A.S.H.V.E. COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)²¹

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.

warmth or comfort felt by persons when exposed to an identical atmospheric condition. The state of health, age, sex, clothing, activity, and the degree of acquired adaptation seem to be the important factors affecting the comfort standards.

Since the prolonged effects of temperature, humidity and air movement on health are not known to the same extent as their effects on comfort, the optimum conditions for health may not be identical with those for comfort. On general physiologic grounds, however, the two do not

²¹How to Use the Effective Temperature Index and Comfort Charts (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

differ greatly since this is in accordance with the efficient operation of the heat regulating mechanism of the body. This belief is strengthened by results of studies on premature infants over a four-year period²⁰. By adjusting the temperature and humidity so as to stabilize the body temperature of these infants, the incidence of diarrhoea and mortality was decreased, gains in body weight increased and infections were reduced to a minimum.

Comfort Chart; Comfort Line; Comfort Zone

Fig. 5 shows a comfort chart, developed at the A.S.H.V.E. Laboratory, on which the average and extreme comfort zones have been superimposed. The *extreme comfort zone* includes air conditions in which one or more of the experimental subjects were comfortable. The *average comfort zone* includes those air conditions in which the majority of the subjects (50 per cent or more) were comfortable. That particular effective temperature at which the maximum number of subjects was comfortable was called the *comfort line*.

The *average winter comfort zone* as determined at the A.S.H.V.E. Laboratory ranges from 63 deg to 71 deg ET (effective temperature). In winter while at rest, a large percentage of persons normally clothed were found to be comfortable at 66 deg ET and this temperature has been accepted by a committee of the Society²¹ as the *winter comfort line* or *optimum effective temperature*.

The comfort line separates the cool air conditions to its left from the warm air conditions to its right. Under the air conditions existing along or defined by the comfort line, the body is able to maintain thermal equilibrium with its environment with the least conscious sensation to the individual, or with the minimum physiologic demand on the heat regulating mechanism. This environment involves not only the condition of the air with respect to temperature and humidity, but also the condition of the surrounding objects and wall surfaces. The comfort zone tests were made in rooms with wall surface temperatures approximately the same as the room dry-bulb temperature. For walls of large area having unusually high or low surface temperatures, however, a somewhat lower or higher range of effective temperature is required to compensate for the increased gain or loss of heat to or from the body by radiation²².

The *average summer comfort zone* for exposures of 3 hours or more ranges from about 66 deg to 75 deg ET, based on studies made at the Harvard School of Public Health²³. The probable optimum effective temperature (for exposures of 3 hours or more) is 71 deg. These effective temperatures average about 4 deg higher than those found in winter when customary winter clothing was worn. The variation from winter to summer is probably due partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons.

The best effective temperature (for exposures lasting 3 hours or more) was found to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained at approxi-

²⁰Application 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).

²²Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghton and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).

mately the same value in July, August and September, and although the average monthly temperature did not vary much, the prevailing outdoor temperature ranged from 70 F to 99.5 F. A decrease in the optimum temperature became apparent only when the prevailing outdoor temperature fell to 66 F, which is below the customary room temperature in the United States for summer and winter.

Young men as a general rule prefer conditions in the cool region of the comfort zone, and women and older people in the warm region of the comfort zone. Crowding the experimental chamber lowered the optimum effective temperature from 70.8 deg when the gross floor area per occupant was 44 sq ft and the air space 380 cu ft, to 69.4 deg when the floor area was reduced to 14 sq ft and the air space to 120 cu ft per occupant.

In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the Harvard School of Public Health, a relative humidity of 70 per cent was found to be *somewhat humid* in winter, by about half of the subjects who were stripped to the waist, even when the dry-bulb temperature was 70 F or less. In summer, a relative humidity of 30 per cent was pronounced as a *little too dry* by about a third of the subjects wearing warm-weather clothing. So long as the temperature was kept within proper limits, the majority of the subjects were unable to detect sensations of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 and 60 per cent. This is in accord with studies by Howell²², Miura²⁴ and others.

Dry air produces an excessive loss of moisture from the skin and respiratory tract. Owing to the cooling effect of evaporation, higher temperatures are necessary, and this condition leads to discomfort and lassitude. Moist air, on the other hand, interferes with the normal evaporation of moisture from the skin, and again may cause a feeling of oppression and lassitude, especially when the temperature is also high.

Just what the optimum range of humidity is, is a matter of conjecture. There seems to exist a general opinion, supported by some experimental and statistical data, that warm, dry air is less pleasant than air of a moderate humidity, and that it dries up the mucous membranes in such a way as to increase susceptibility to colds and other respiratory disorders^{25, 26, 27}.

For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth²⁸, and according to Huntington²⁹, this seems to be the case for adults also. All of these

²²Humidity and Comfort, by W. H. Howell (*The Science Press*, April, 1931).

²⁴Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal of Hygiene*, Vol. 13, 1931, p. 432).

²⁵Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surface, by Mudd, Stuart, et al (*Journal Experimental Medicine*, 1921, Vol. 34, p. 11).

²⁶Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surfaces, by A. Goldman, et al and Concurrent Study of Bacteriology of Nose and Throat (*Journal Infectious Diseases*, 1921, Vol. 29, p. 151).

²⁷The Etiology of Acute Inflammations of the Nose, Pharynx and Tonsils, by Mudd, Stuart, et al (*Am. Otol., Rhinol., and Laryngol.*, 1921).

²⁸Application 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).

²⁹Weather and Health, by Ellsworth Huntington (*Bulletin of the National Research Council* No. 75. The National Academy of Science, Washington, D. C., 1930).

studies indicate that the optimum humidity must always be considered in combination with temperature.

Until more exact information is secured, it would be desirable to restrict the comfort zones to the range of relative humidity employed in the comfort zone experiments, namely, 30 to 70 per cent. Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort, so long as extremely low humidities are avoided. From the standpoint of health, however, the consensus seems to favor a relative humidity between 40 and 60 per cent. In mild weather such comparatively high relative humidities are entirely feasible, but in cold or sub-freezing weather they are objectionable on account of condensation and frosting on the windows. They may even cause serious damage to certain building materials of the exposed walls by condensation and freezing of the moisture accumulating inside these materials. Unless special precautions are taken to properly insulate the affected surfaces, it will be necessary to reduce the degree of artificial humidification in sub-freezing weather to less than 40 per cent, according to the outdoor temperature. Information on the prevention of condensation on building surfaces is given in Chapter 7. The principles underlying humidity requirements and limitations are discussed more fully elsewhere³⁰.

The comfort chart (Fig. 5) applies to adults between 20 and 70 years of age living in the northeastern parts of the United States. For prematurely born infants, the optimum temperature varies from 100 F to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 F to 68 F with natural indoor humidities. For school children, the studies of the *New York State Commission on Ventilation* place the optimum air conditions at 66 F to 68 F temperature with a moderate humidity (not specified) and a moderate but not excessive amount of air movement (not specified)³¹.

Satisfactory comfort conditions are found to vary from 40 deg to 70 deg ET, depending upon the rate of work and amount of clothing worn. The effective temperatures giving maximum comfort for persons working have been determined by the A.S.H.V.E. Research Laboratory³² for a rate of work which is considered hard labor. For this degree of work, 50 per cent were fairly comfortable for temperatures ranging from 46 to 64 deg ET, while the greatest percentage found maximum comfort at 53 deg ET. In hot industries, 80 deg ET is considered the upper limit compatible with efficiency, and, whenever possible, this should be reduced to 70 deg ET or less.

APPLICATION OF COMFORT CHART

The average winter comfort line (66 deg ET) applies to average American men and women living inside the broad geographic belt across

³⁰Humidification for Residences, by A. P. Kratz (*University of Illinois Engineering Experiment Station Bulletin* No. 230, July 28, 1931).

³¹Ventilation, Report of the New York State Commission on Ventilation, 1923.

³²A.S.H.V.E. research paper entitled 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).

the United States in which central heating of the convection type is generally used during four to eight months of the year. It does not apply to rooms heated by radiant energy, or to rooms with excessive glass area or rooms with poorly insulated or cold walls, and it has not been advocated officially for use in foreign countries where the climate, heating methods, and general living conditions are materially different from those in the United States, although several foreign workers have attempted to show that it cannot be so applied. Even in the warm south and southwestern climates, and in the very cold north-central climate of the United States, the comfort chart would probably have to be modified according to climate, living and working conditions, and the degree of acquired adaptation.

In densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures are necessary than those indicated by the comfort line on account of counter-radiation between the bodies of occupants²² in close proximity. In rooms in which the average wall surface temperature is considerably below the air temperature, higher air temperatures are necessary. The reverse holds true in radiant or panel heating methods. (See Chapter 38.)

The sensation of comfort, in so far as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated by the comfort line, it should not be expected that all the occupants of a room will feel perfectly comfortable. When the winter comfort line is applied in accordance with the foregoing recommendations, the majority of the occupants will be perfectly comfortable, but there will always be a few who would feel *a bit too cool* and a few *a bit too warm*. These individual differences among the minority should be counteracted by suitable clothing.

Air conditions lying outside the average comfort zone but within the extreme comfort zone may be comfortable to certain persons. In other words, it is possible for half of the occupants of a room to be comfortable in air conditions outside the *average* comfort zone, but in the majority of cases, if not in all, these conditions will be well within the extreme comfort zone as determined experimentally.

Strictly speaking, the only authoritative comfort zone on which accurate data are available, is that for 15 to 25 fpm air movement or turbulence (often referred to as still air). In the past, the winter comfort zone has often been superimposed on effective temperature charts for various air velocities, on the assumption that air conditions of equal warmth are approximately equally comfortable. This may hold in hot industries where the workers are adapted to high temperatures and strong air currents, but it does not apply to sedentary conditions. To ascertain approximately whether a given industrial condition is reasonably comfortable, it would be necessary first to compute the effective temperature from Figs. 1, 2 or 3, and then to refer this effective temperature to the comfort chart (Fig. 5), or to refer directly to a chart or table for the proper air velocity.

The summer comfort line (71 deg ET) is applicable to the same geographic area as the winter comfort line. It is further restricted to cases in which the human body has reached thermal equilibrium with its environment. As a general rule this takes place after $1\frac{1}{2}$ to 3 hours' exposure.

When a person from outdoors enters a room cooled to 71 deg ET on a hot day (95 F or over) an intense chill is likely to be experienced which is unpleasant. However, after remaining in the room for about 2 hours, this fundamental optimum condition will prove satisfactory to the average person. The summer comfort zone, as well as the comfort line, makes proper allowance for these adaptive changes in the body, and thus applies to homes, offices, schools and other similar places where persons of sedentary occupations spend from 3 to 8 or more hours daily.

In artificially cooled theaters, department stores, restaurants, and other public buildings where the period of occupancy is short, the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the temperature and humidity to be maintained. The object of cooling such places in the summer is not to reduce the temperature to the

TABLE 2. DESIRABLE INDOOR AIR CONDITIONS IN SUMMER CORRESPONDING TO OUTDOOR TEMPERATURES

Applicable to Exposures Less Than 3 Hours

OUTDOOR TEMPERATURE (DEG FAHR)	INDOOR AIR CONDITIONS WITH DEW POINT CONSTANT AT 57 F		
	DRY-BULB	WET-BULB	EFFECTIVE TEMP
95	80.0	65.0	73
90	78.0	64.5	72
85	76.5	64.0	71
80	75.0	63.5	70
75	73.5	63.0	69
70	72.0	62.5	68

optimum degree, but to maintain therein a temperature which is temporarily comfortable to the patrons who thus avoid sensations of chill and intense heat on entering and leaving the building. The relative humidity should be low enough (about 50 per cent or less) to give a sense of comfort without chill and to induce a rate of evaporation which will keep clothing and skin dry. For exposures less than 3 hours, desirable indoor conditions in summer corresponding to various outdoor temperatures are given in Table 2.

It should be kept in mind that southern people, with their more sluggish heat production and lack of adaptability, will demand a comfort zone several degrees higher than those given here for the more active people of northern climates. Instead of the summer comfort line standing at 71 deg as here given, it was found to be much higher for foreigners in Shanghai where climatic conditions are similar to those of our gulf states. This difference in basic metabolic level of people forms a very real problem for air conditioning engineers, which they must recognize in their efforts to give proper conditions of comfort. Cooling of theaters, restaurants, and other public buildings in southern climates cannot be based on northern standards without considerable modification.

A.S.H.V.E. VENTILATION STANDARDS³³

It is the intent of the Committee in presenting this report to confine itself to a statement of those requirements which, based on present day knowledge, will provide adequate ventilation for spaces intended for human occupancy. The following standards shall apply to all spaces occupied by human beings in all buildings for which ventilation regulations are to be established.

SECTION I—AIR TEMPERATURE AND HUMIDITY

The temperature and humidity of the air in such occupied spaces, and in which the only source of contamination is the occupant, shall be maintained at all times during occupancy at an *Effective Temperature*, as hereinafter stated.

The relative humidity shall be not less than 30 per cent, nor more than 60 per cent in any case. The Effective Temperature shall range between 64 deg and 69 deg when heating or humidification is required, and between 69 deg and 73 deg when cooling or dehumidification is required.

These Effective Temperatures shall be maintained at a level of 36 in. above the floor. (See Appendix, Tables A and B).

SECTION II—AIR QUALITY

The air in such occupied spaces shall at all times be free from toxic, unhealthful or disagreeable gases and fumes and shall be relatively free from odors and dust.

In every space coming within the provisions of these requirements and in which the quality of the air is below the standards prescribed by good medical and engineering practices, due to toxic substances, bacteria, dust, excessive temperature, excessive humidity, objectionable odors, or other similar causes, means for ventilating shall be provided so that the quality of the air shall be raised to these standards.

SECTION III—AIR MOTION

The air in such occupied spaces shall at all times be in constant motion sufficient to maintain a reasonable uniformity of temperature and humidity, but not such as to cause objectionable drafts in any occupied portion of such spaces.

The air motion in such occupied spaces, and in which the only source of contamination is the occupant, shall have a velocity of not more than 50 ft per minute, measured at a height of 36 in. above the floor.

SECTION IV—AIR DISTRIBUTION

The air in all rooms and enclosed spaces shall, under the provisions of these requirements, be distributed with reasonable uniformity, and the variation in the carbon dioxide content of the air shall be taken as a measure of such distribution.

The air in a space ventilated in accordance with these requirements, and in which the only source of contamination is the occupant, shall be distributed and circulated so that the variation in the concentration of carbon dioxide, when measured at a height of 36 in. above the floor, shall not exceed one part in 10,000.

SECTION V—AIR QUANTITY

The quantity of air used to ventilate the given space during occupancy shall always be sufficient to maintain the standards of air temperature, air quality, air motion and air distribution as herein required. Not less than 10 cu ft per minute per occupant of the total air circulated to meet these requirements shall be taken from an outdoor source.

APPENDIX

Definitions

For the purposes of these standards the terms used shall be defined as follows:—

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*).

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, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

³³Report of A.S.H.V.E. Committee on Ventilation Standards consisting of W. H. Driscoll, *Chairman*, J. J. Aeberly, F. Paul Anderson, L. A. Harding, D. D. Kimball, J. R. McColl, C. L. Riley, W. A. Rowe, Perry West and A. C. Willard, presented at the Semi-Annual Meeting of the Society, Milwaukee, Wis., June, 1932, and adopted by the Society in August, 1932.

CHAPTER 3—VENTILATION AND AIR CONDITIONING STANDARDS

Dry-Bulb Temperature: The temperature of the air which is indicated by any type of thermometer which is not affected by the water vapor content or relative humidity of the air.

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, such as are blown on a windy day, the average diameter of which is approximately 0.01 centimeter, may be called dust.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Humidity: The water vapor (either saturated or superheated steam) occupying any space, which may or may not contain other vapors and gases at the same time.

Relative Humidity: A ratio, although usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space, which depends merely on the temperature and partial pressure of the vapor.

Spaces in Which the Only Source of Contamination is the Occupant: Spaces in which the atmospheric contamination results entirely from the respiratory processes of the occupant, including heat, moisture, and odors given off by the body. No manufacturing or industrial processes or other sources of atmospheric contamination, including heat and moisture, than people are considered under this title.

TABLE A. EFFECTIVE TEMPERATURES RANGING FROM 64 DEG TO 69 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE^a

(For use when heating or humidification is required)

DRY-BULB TEMPERATURES (DEG FAHR)	RELATIVE HUMIDITIES (PER CENT)						
	30	35	40	45	50	55	60
	EFFECTIVE TEMPERATURES (DEGREES)						
67							
68							
69	64.1	64.4	64.0	64.2	64.5	64.0	64.3
70	64.8	65.1	65.4	65.8	66.2	65.7	65.1
71	65.5	65.8	66.2	66.6	67.0	66.5	66.0
72	66.2	66.5	66.9	67.3	67.7	67.3	66.8
73	67.0	67.3	67.7	68.1	68.5	68.1	67.7
74	67.7	68.0	68.4	68.8			
75	68.4	68.7					
76	69.0						

^aSee Fig. 5.

TABLE B. EFFECTIVE TEMPERATURES RANGING FROM 69 DEG TO 73 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE^a

(For use when cooling or dehumidification is required)

DRY-BULB TEMPERATURES (DEG FAHR)	RELATIVE HUMIDITIES (PER CENT)						
	30	35	40	45	50	55	60
	EFFECTIVE TEMPERATURES (DEGREES)						
73							
74							
75							
76	69.0	69.4	69.1	69.5	69.3	69.7	69.3
77	69.7	70.2	70.7	71.2	70.0	71.5	70.1
78	70.4	70.9	71.4	71.9	70.8	71.3	71.8
79	71.1	71.6	72.2	72.6	71.6	72.1	72.6
80	71.8	72.4	72.9		72.4	73.0	
81	72.5						

^aSee Fig. 5.

^bThis table applies primarily to cases in which the human body has reached equilibrium with the surrounding air. A higher plane of summer effective temperatures is required in places of public assembly where the period of occupancy is short, than is required for offices and industrial plants where the period of occupancy is of longer duration. When the period of occupancy is two hours or less, the dry-bulb temperature shall be 72 F plus one-third of the difference between the outside dry-bulb temperature and 70 F, and the relative humidity shall not exceed 60 per cent. (See also Table 2.)

FACTORS INFLUENCING APPLICATIONS

The conditions and limitations outlined under the heading Application of Comfort Chart should be noted in applying the temperatures and relative humidities specified in Tables A and B of the preceding A.S.H.V.E. Ventilation Standards.

Air Quality

In occupied spaces in which the vitiation is entirely of human origin, the chemical composition of the air, the dust, and bacteria content may be dismissed from consideration so that the problem consists in maintaining a suitable temperature with a moderate humidity, and in keeping the atmosphere free from objectionable odors. Such unpleasant odors, human or otherwise, can be easily detected by persons entering the room from clean, odorless air. A further discussion of air quality will be found in Chapters 15 and 16.

Air Motion

As a result of studies by Baetjer³⁴ and work carried on by the A.S.H.V.E. Research Laboratory, it is now recognized that the importance of air motion in air conditioning ranks only second to temperature. Air in an occupied space having all the other essential qualities but lacking in air motion feels stagnant, stuffy, and depressing, because the vitiated air next to the body is not replaced by the surrounding air possessing the satisfactory qualities. Hence, air motion is absolutely essential that an occupant may realize the other desired qualities of the atmosphere. Possible limits in variation in air motion may range from 5 fpm to 50 fpm, as measured by the Kata thermometer. (See Chapter 43.) However, satisfactory results are more likely to be insured by air velocities ranging from 15 to 30 fpm. The limit of 5 fpm may be taken as the minimum during the heating season, and 50 fpm as the maximum for the cooling season.

Air Distribution

Variation in concentration of carbon dioxide in different parts of an occupied room has been used as a measure of satisfactory distribution of the outside or conditioned air supply. For satisfactory air distribution, the carbon dioxide concentration at the 36-in. level should not vary by more than one part in 10,000 parts of air. Recent work² by the A.S.H.V.E. Research Laboratory demonstrates that variations in dry-bulb temperature, wet-bulb temperature, or moisture content of the air are equally good indices of air distribution. This work also indicates that the presence of satisfactory air motion within the room (15 to 30 fpm as measured by the Kata thermometer) insures satisfactory distribution. Because of the laborious and exacting technique involved in making carbon dioxide determinations, it is recommended that satisfactory distribution can be amply insured by the presence of such air velocities in all parts of the room together with dry-bulb temperature variations of not to exceed 3 deg at the 36-in. level.

³⁴Threshold Air Currents in Ventilation (*American Journal of Hygiene*, Vol. IV, No. 6, p. 650, 1924).

Air Quantity

The quantity of air to be circulated through an occupied space, whether by natural or mechanical means, or whether the air is conditioned or not, must in all cases be sufficient to maintain the required standards of air temperature, quality, motion and distribution. The factors which determine air quantity include the type and nature of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and last but not least, the method of distribution.

The quantity of air supplied to a room by an air conditioning or ventilating system serves two purposes: *First*, the supply of sufficient outside air for the needs of the occupants; and *second*, the setting up of circulation or air motion within the room. Until recently it was considered that 30 cfm were necessary in any occupied space, particularly in a classroom. It has since been demonstrated that 10 cfm of outside air per person is frequently sufficient to remove body heat, insure against body odors, and provide the chemical needs of respiration. However, it is found that a greater volume should be circulated in the average room in order to provide the required air motion. It is now customary to supply the minimum amount of outside or conditioned air required for removing heat and odors, and to recirculate the additional volume.

In offices and small rooms where the occupants smoke, from 6 to 7 cfm of outside air per occupant will be necessary to eliminate the nuisance effects of the smoke; this quantity of air, however, may be a part of that necessary for other ventilation requirements. Restaurants which permit smoking, because of the exposed food and the necessity that restaurant air seem very clean, need from 10 to 12 cfm of outside air per occupant to care for the smoke condition. This air, likewise, need not be in addition to that required for other ventilation purposes.

Temperature Rise

The total quantity of air introduced is governed largely by the needs for controlling temperature and humidity when either heating or cooling is required. As a rule, the introduction and distribution of warm air into an occupied space does not present as many difficulties as does the introduction of cold air. The former is determined from the amount of heat to be given up to the space, and the latter is determined from the amount of heat to be removed from the space, using a temperature rise that will produce uniform distribution without the production of disagreeable drafts.

Fig. 6 shows the changes in carbon dioxide concentration and moisture content resulting from occupation, in the atmosphere of a room supplied with various volumes of outside air. Data are given for an adult, 5 ft 8 in. in height weighing 150 pounds and having a body surface area of 19.5 sq ft, and for a child, 12 years of age, 4 ft 7 in. in height, weighing 76.6 pounds and having a body surface area of 12.6 sq ft. It is a recognized fact that the dissipation of heat and moisture to the atmosphere, the addition of carbon dioxide, and all metabolic changes take place in proportion to the surface area of the individual. Hence, data for persons of other sizes may be obtained by interpolating among the curves given. The rate of sensible heat production is given in Fig. 9. Fig. 6 also gives

the temperature of the incoming air necessary to maintain a room temperature of either 70 or 80 F as indicated, assuming that there is no heat gain or loss to the room by transmission through the walls, solar radiation or other sources.

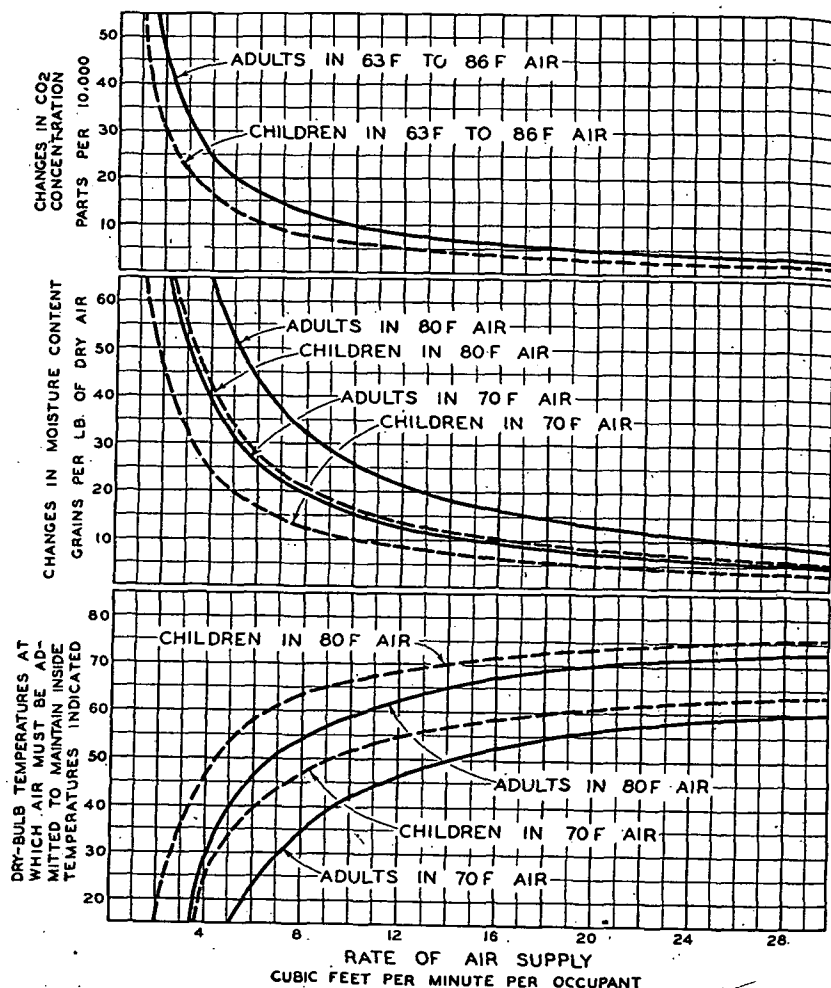


FIG. 6. RELATION AMONG RATE OF AIR CHANGE PER OCCUPANT, CARBON DIOXIDE CONCENTRATION AND MOISTURE CONTENT OF ENCLOSURE, AND DRY-BULB TEMPERATURE OF INCOMING AIR

Two of the most important factors on which the *temperature rise* depends are (1) the method of distribution and (2) the most economical temperature rise for the conditions involved. Some systems of distri-

bution produce drafts with but a few degrees temperature rise, while other systems operate successfully with a temperature rise as high as 35 deg. The total air quantity introduced in any particular case is inversely proportional to the temperature rise, and depends largely upon the judgment and ingenuity of the engineer in designing the most suitable system for the particular conditions.

Outside Air. In order to provide uniform temperature conditions, it is necessary to maintain a pressure of about 0.1 in. of water in the room or space to be ventilated or conditioned. This usually requires the introduction of a certain amount of outside air which depends on the particular conditions involved, and may vary over a considerable range.

NATURAL AND MECHANICAL VENTILATION

Under favorable conditions natural ventilation methods properly combined with means for heating may be sufficient to provide for the foregoing standards. As a rule, in instances in which the only source of contamination is the occupant, the requirements may be fulfilled when the following conditions prevail:

1. At least 50 sq ft of floor area for each occupant.
2. At least 500 cu ft of air space per occupant.
3. Effective openings in windows and skylights equal to at least 5 per cent of the floor area.

Whenever natural means are not sufficient to maintain the standards, resort must be made to whatever modifications or mechanical apparatus are necessary to secure such standards.

In large offices, large school rooms, and in public and industrial buildings, natural ventilation is uncertain and makes heating difficult. The chief disadvantage of natural methods is the lack of control; they depend largely on weather and upon the velocity and direction of the wind. Rooms on the windward side of a building may be difficult to heat and ventilate on account of drafts, while rooms on the leeward side may not receive an adequate amount of air from out of doors. The partial vacuum produced on the leeward side under the action of the wind may even reverse the flow of air so that the leeward half of the building has to take the *drift* of the air from the rooms of the windward half. Under such conditions no outdoor air would enter through a leeward window opening, but room air would pass out.

In warm weather natural methods of ventilation afford little or no control of indoor temperature and humidity. Outdoor smoke, dust and noise constitute other limitations of natural methods.

RECIRCULATION AND OZONE

The saving in operating costs due to recirculation of the air, while very considerable, must not be obtained at the expense of air quality. The percentage of recirculated air may be varied to suit the seasonal changes so as to conserve heat in winter and refrigeration in summer, but at no time during occupancy should there be taken from out of doors less than

10 cfm for each occupant. As a general rule, recirculation impairs the quality of the air by excessive humidity (if not conditioned), excessive odors, or both, and it tends to deprive the air of its ionic content, but the influence of this factor on comfort and health is at present a matter of speculation.

Ozone has been used as an adjunct to ventilation as a deodorant, particularly where a portion of the air is recirculated in order to effect a saving in both heating and refrigeration. At proper concentrations ozone exerts practically no effect on pathogenic air borne organisms, but will eliminate most of the undesirable odors usually present in recirculated air. It is very important to use only equipment which provides for positive control of ozone output, as higher concentrations produce a pungent and

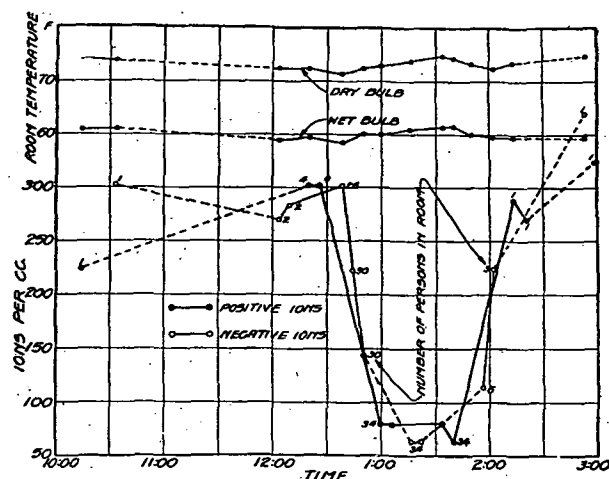


FIG. 7. INFLUENCE OF ROOM OCCUPANCY ON IONIC CONTENT³⁵
(Cubical Contents of Room, 10,000 Cu Ft; Number of Occupants, 34)

unpleasant odor, with considerable discomfort to human beings so exposed.

Toilets, kitchens, and similar rooms, in buildings using recirculation, should be separately mechanically ventilated by exhausting the air from them in order to prevent objectionable odors from diffusing into other parts of the building.

ULTRA-VIOLET RADIATION AND IONIZATION

In spite of the rapid advances made in the field of air conditioning during the past few years, the secret of reproducing, in indoor spaces, atmospheres of as stimulating qualities as those existing outdoors in the country, under ideal weather conditions, has not as yet been found. In fact, extensive studies have failed to elucidate the cause of the stimulating quality of outdoor country air, qualities which are lost when such air is brought indoors and particularly when it is handled by mechanical

TABLE 3. RELATION BETWEEN METABOLIC RATE AND ACTIVITY³⁵

ACTIVITY	METABOLIC RATE BTU PER HOUR FOR AVERAGE MAN (19.5 Sq. Ft. SURFACE AREA)	AUTHORITY
Seated at rest.....	384	Research Laboratory, American Society of Heating and Ventilating Engineers.
Standing at rest.....	431	Research Laboratory, American Society of Heating and Ventilating Engineers.
Walking 2 mph.....	761	Average values from Douglas, Haldane, Henderson and Schneider; and Henderson and Haggard.
Walking 3 mph.....	1049	Douglas, Haldane, Henderson and Schneider
Walking 4 mph.....	1388	Average values from Douglas, Haldane, Henderson and Schneider; and Henderson and Haggard.
Walking 5 mph.....	2530	Douglas, Haldane, Henderson and Schneider
Slow run.....	2285	Henderson and Haggard
Very severe exercise.....	2555	Benedict and Carpenter
Maximum exertion.....	3333 to 4762+	Henderson and Haggard
Tailor.....	482	Becker and Hamalainen
Bookbinder.....	626	Becker and Hamalainen
Shoemaker.....	661	Becker and Hamalainen
Carpenter.....	762 to 963	Becker and Hamalainen
Metal worker.....	862	Becker and Hamalainen
Painter (of furniture).....	876	Becker and Hamalainen
Stonemason.....	1488	Becker and Hamalainen
Man sawing wood.....	1797	Becker and Hamalainen

means. It is true that many suggestions have been advanced to account for the stimulating quality of outdoor air, such as ultra-violet light and ionization. At the present time neither of these suggestions has received any degree of scientific confirmation.

It is generally recognized that total outdoor solar radiation has marked curative value in certain diseases and is also a powerful germicidal agent. A critical review of the literature, however, does not substantiate the theory that ultra-violet radiation is of importance in air conditioning, since the use of ultra-violet sources fails to produce indoors, the previously mentioned stimulating qualities found in outdoor air.

Experiments³⁶ show that in occupied rooms there is a marked decrease in both positive and negative small ions. As shown in Fig. 7, soon after the occupants assembled the ionic content fell abruptly to a very low level which was maintained until the occupants left the room. Both positive and negative ions began to rise again as soon as the occupants departed.

The effects of the decrease in the ionic content of indoor air on comfort and health have not yet been subjected to sufficient scientific investigation. It would appear, however, from the evidence at hand, that comfort is not associated with a high ion content—but this must be considered, at least for the time being, as still a subject for further study.

³⁵A.S.H.V.E. research paper entitled 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. 37, 1931). 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). 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). The Nature of Ions in Air and their Possible Physiological Effects, L. B. Loeb (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934).

HEAT AND MOISTURE LOSSES

In order to solve air conditioning problems involving the human body it is necessary to know the rate at which sensible and latent heat are given up by the body under various conditions of temperature and activity. Research at the A.S.H.V.E. Laboratory^{32, 36} has resulted in the data given in Figs. 9, 10, and 11. Table 3 gives the metabolic rates for various degrees of activity.

The experimental data from which the curves were drawn show that

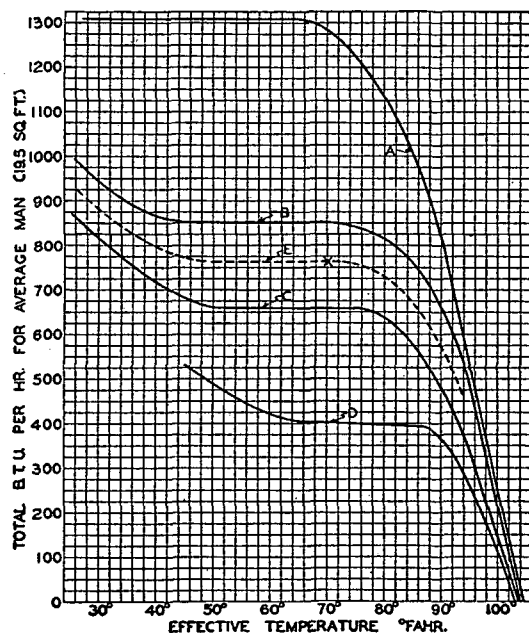


FIG. 8. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR^a

^aCurve A—Men working 66,160 ft-lb per hour. Curve B—Men working 33,075 ft-lb per hour. Curve C—Men working 16,538 ft-lb per hour. Curve D—Men seated at rest. Curves A and C drawn from data at an effective temperature of 70 deg only and extrapolating the relation between curves B and D, which were drawn from data at many temperatures.

total heat loss does not vary appreciably within the comfort zone (see Fig. 8). Above or below this range the variation is approximately a function of effective temperature. Sensible and latent heat losses (Figs. 9 and 11) on the other hand, vary greatly within the comfort zone, the variation following closely the dry-bulb temperature.

Although total heat loss and sensible and latent heat losses are not exact functions of effective and dry-bulb temperature, respectively, for all

^aThermal 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, No. 2, March, 1931, pp. 415-431).

conditions of humidity and air motion, they are plotted as such in the curves. This is accomplished by approximations which are sufficiently accurate for application to practical problems. Comparison of Figs. 9 and 10 shows how the cooling load may vary between sensible and latent heat elimination for different atmospheric conditions and activities of occupants.

An atmospheric condition resulting in sensible perspiration is to be

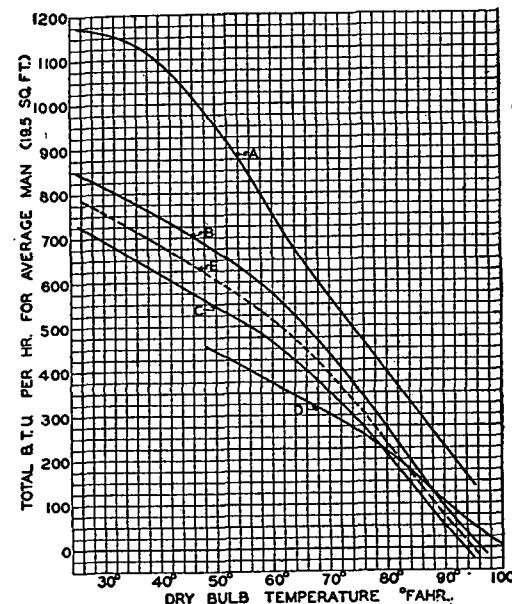


FIG. 9. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR^a

^aCurve A—Men working 66,150 ft-lb per hour. Curve B—Men working 33,075 ft-lb per hour. Curve C—Men working 16,538 ft-lb per hour. Curve D—Men seated at rest. Curves A and C drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves B and D which were drawn from data at many temperatures.

avoided for obvious reasons. Tables 4 and 5 give the approximate effective temperatures at which perspiration is noticeable in different degrees for 95 per cent and 20 per cent relative humidity.

In theaters, auditoriums, department stores and other crowded enclosures, the amount of heat and moisture given off by the people is so large that normal changes in outside temperature and humidity have relatively little effect on indoor air conditions. The principal object of air conditioning in such places is to remove excessive heat and moisture by supplying a sufficient quantity of properly conditioned air. The indoor air conditions, however, must be varied according to the outside temperature, as has been pointed out.

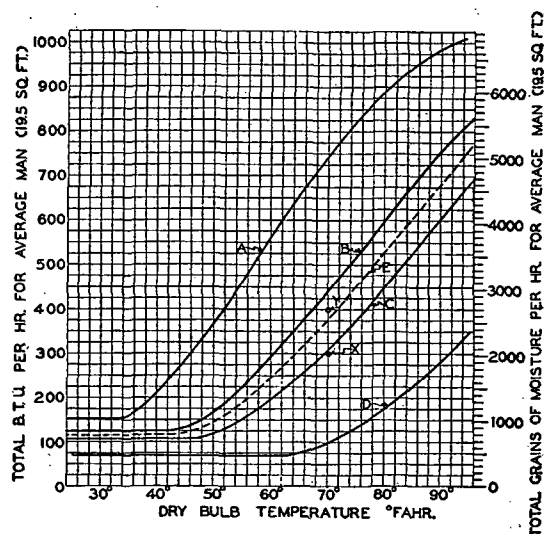


FIG. 10. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS^a

^aCurve A—Men working 66,150 ft-lb per hour. Curve B—Men working 33,075 ft-lb per hour. Curve C—Men working 16,538 ft-lb per hour. Curve D—Men seated at rest. Curves A and C drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between Curves B and D which were drawn from data at many temperatures.

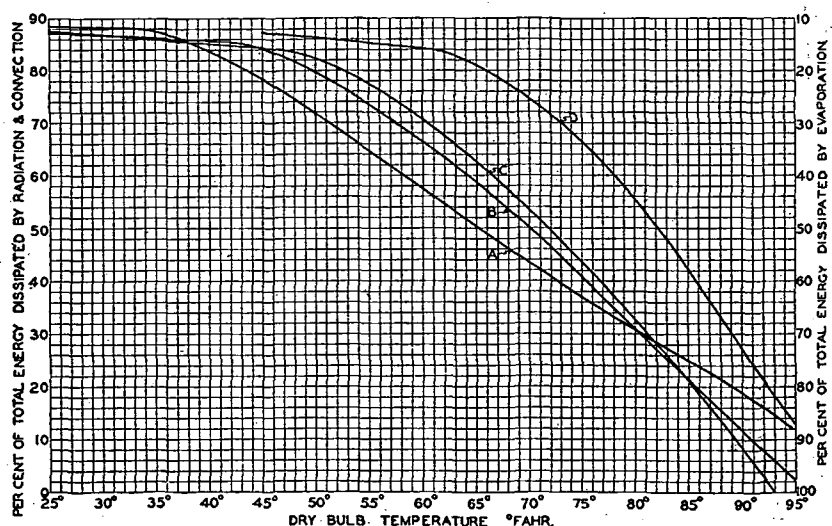


FIG. 11. HEAT LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION AND CONVECTION IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS^a

^aCurve A—Men working 66,150 ft-lb per hour. Curve B—Men working 33,075 ft-lb per hour. Curve C—Men working 16,538 ft-lb per hour. Curve D—Men seated at rest. Curves A and C drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between Curves B and D which were drawn from data at many temperatures.

Although heat and moisture from the human body constitute the major portion of the cooling load, in most cases where air conditioning is provided for comfort and health other factors must also be considered. These include heat from lights, machinery, and processes, as well as the transmission and infiltration of heat through the building structure. The computations for these factors may be made in accordance with data given in Chapters 5 and 7.

TABLE 4. CONDITION OF SENSIBLE PERSPIRATION FOR PERSONS SEATED AT REST UNDER VARIOUS ATMOSPHERIC CONDITIONS^a

DEGREE OF PERSPIRATION ^b	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E. T.	D. B.	W. B.	E. T.	D. B.	W. B.
Forehead clammy.....	73.0	73.6	72.4	75.0	87.0	60.7
Body clammy.....	73.0	73.6	72.4	75.0	87.0	60.7
Body damp.....	79.0	79.7	78.4	81.0	97.5	67.5
Beads on forehead.....	80.0	80.8	79.4	87.0	109.4	75.2
Body wet.....	84.5	85.4	84.0	86.5	108.5	74.6
Perspiration on forehead runs and drips.....	88.0	89.0	87.6	94.0	125.2	85.4
Perspiration runs down body.....	88.5	89.5	88.1	90.0	116.0	79.5

^aForty per cent of subjects registered degree of perspiration equal to or greater than indicated.

TABLE 5. CONDITION OF SENSIBLE PERSPIRATION FOR PERSONS AT WORK UNDER VARIOUS ATMOSPHERIC CONDITIONS

DEGREE OF PERSPIRATION ^a	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E. T.	D. B.	W. B.	E. T.	D. B.	W. B.
Forehead clammy.....	59.0	59.4	58.3	69.5	80.5	56.5
Body clammy.....	50.0	50.2	49.3	57.0	61.6	44.2
Body damp.....	60.0	60.3	59.3	62.5	69.6	49.5
Beads on forehead.....	68.0	68.5	67.5	76.0	91.0	63.4
Body wet.....	69.0	69.6	68.5	71.0	82.8	53.0
Perspiration on forehead runs and drips.....	78.5	79.3	78.0	82.0	100.5	70.2
Perspiration runs down body.....	79.0	79.8	78.5	81.0	99.8	69.0

^aForty per cent of subjects registered degree of perspiration equal to or greater than indicated.

In many cases, allowance must also be made for sun effect and for heat capacity of the building structure in accordance with studies by the A.S.H.V.E. Research Laboratory³⁷. Another item to be considered is the radiant heat received by the body from high temperature wall and ceiling surfaces.

³⁷Heat 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).

PROBLEMS IN PRACTICE

- 1 • Determine the amount of sensible and latent heat given off by each person on a dance floor if the dry-bulb temperature is 75 F.

First select from Table 3 an occupation which would seem approximately as strenuous as dancing. Let us assume that the average dancer will have the same metabolic rate as a person walking 3 miles per hour. Then the total heat emission will be 1049 Btu per hour per person. If we assume that conditions in the dance hall are within the comfort zone, then we can locate our point with reference to the curves in Fig. 8. We find that 1049 Btu falls between curves A and B, and slightly closer to B.

Knowing this, the amounts of sensible and latent heat may be determined from Figs. 9 and 10, or the relative amounts may be determined from Fig. 11.

Using Fig. 11, we find that for a 75 F dry-bulb temperature, approximately 61 per cent of the heat will be by evaporation and 39 per cent will be sensible heat.

Latent heat = $1049 \times 0.61 = 640$ Btu per hour.

Sensible heat = $1049 \times 0.39 = 409$ Btu per hour.

- 2 • What is the purpose and method of conditioning the air of occupied rooms?

Chiefly comfort, and the method is to control the temperature, humidity, and air distribution, and to prevent the accumulation of excessive body odors in the air. Other factors have yet to be studied.

- 3 • What are the most comfortable air conditions?

Comfort standards are not absolute, but they are greatly affected by the physical condition of the individual, and the climate, season, age, sex, clothing, and physical activity. For the northeastern climate of the United States, the conditions which meet the requirements of the majority of people consist of temperatures between 68 and 72 F in winter and between 70 and 85 F in summer, the latter depending largely upon the prevailing outdoor temperature. The most desirable relative humidity range seems to be between 30 and 60 per cent.

- 4 • Are the optimum conditions for comfort identical with those for health?

There are no absolute criteria of the prolonged effects of various air conditions on health. For the present it can be only inferred that bodily discomfort may be an indication of conditions that may produce poor health.

- 5 • Given dry-bulb and wet-bulb temperatures of 75 and 68 F, respectively, first, what is the effective temperature? Second, is this condition warmer or cooler than 80 F dry-bulb and 60 F wet-bulb?

The first condition is given by the intersection of the 75 F dry-bulb line and the 68 F wet-bulb line (Fig. 5). The effective temperature of 72.1 deg is given by the numerical value of the effective temperature line passing through this point and indicated by the scale along the saturation curve. The second condition is given by the intersection of 80 F dry-bulb and 60 F wet-bulb and is 71.8 deg ET. It is therefore 0.3 deg ET cooler than the first condition.

- 6 • Given 76 F dry-bulb and 61 F wet-bulb, how many degrees difference are there between this condition and the winter comfort line or 66 deg ET?

The effective temperature for this condition is given by the intersection of the 76-F dry-bulb and 61-F wet-bulb lines and is 70 deg ET, which is 4 deg ET warmer than the comfort line.

- 7 • Assume that the design of an air conditioning system for a theater is to be based on an outdoor dry-bulb temperature of 95 F and a wet-bulb temperature of 78 F with an indoor relative humidity of 50 per cent. According to Table 2, the dry-bulb temperature in the auditorium should be 80 F. Estimate the sensible and latent heat given up per person.

The sensible heat given up per person per hour under this condition may be obtained from Fig. 9. With an abscissa value of 80 F, Curve D for men seated at rest gives a value

on the ordinate scale of 220 Btu per person per hour as the sensible heat loss. The latent heat given up by a person seated at rest per hour may be obtained from Fig. 10. With an abscissa value of 80 F, Curve D indicates a latent heat loss of 175 Btu per hour (left hand scale) or a moisture loss of 1190 grains per hour (right hand scale).

- 8 • How much sensible heat, how much latent heat and how much water vapor will be added per hour to the atmosphere of an auditorium by an audience of 1000 adults, when the dry- and wet-bulb temperatures are 75 F and 63.5 F, respectively?

From Curve D, Fig. 9, find the sensible heat loss per person for a dry-bulb temperature of 75 F and still air to be 265 Btu per hour. From Fig. 10 find the latent heat loss per person for a dry-bulb temperature of 75 F to be 134 Btu per hour and the moisture added to be 905 grains per hour. Sensible heat = $1000 \times 265 = 265,000$ Btu. Latent heat = $1000 \times 134 = 134,000$ Btu. Water vapor added per hour to the air in the auditorium = $1000 \times 905 = 905,000$ grains or 129 lb.

The sensible and latent heat added to the air may also be found as follows: The effective temperature for dry- and wet-bulb temperatures of 75 F and 63.5 F, respectively, is 70.3 deg. From Curve D, Fig. 8, find 403 Btu as the total heat added to the air by a person for an effective temperature of 70.3 deg. From Fig. 11 find the percentage of sensible and latent heat at a dry-bulb temperature of 75 F to be 66.5 per cent and 33.5 per cent. The sensible heat added to the air in the auditorium is $1000 \times 0.665 \times 403 = 267,995$ Btu per hour. The latent heat added is $1000 \times 0.335 \times 403 = 135,005$ Btu per hour.

- 9 • If the dry- and wet-bulb temperatures of the auditorium were 85 F and 63 F, respectively, how much heat and moisture would be dissipated to the atmosphere?

From Figs. 9 and 10, respectively, the sensible and latent heat losses per person for a dry-bulb temperature of 85 F are found to be 164 and 225 Btu per hour. The water vapor added to the atmosphere is 1520 grains per hour. The audience will then add 164,000 Btu sensible heat, 225,000 Btu latent heat and 1,520,000 grains or 217 lb of water vapor to the air in the auditorium per hour.

- 10 • Neglecting the gain or loss of heat to an auditorium by transmission or infiltration through the walls, windows and doors, how many cubic feet of outside air, with dry- and wet-bulb temperatures of 65 F and 59 F, respectively, (63.1 deg ET) must be supplied per hour to an auditorium containing 1000 people in order that the inside shall not exceed 75 F (dry-bulb) and 65 F (wet-bulb), respectively?

Figs. 9 and 10 give 265 Btu sensible heat and 905 grains of moisture as the additions per person with a dry-bulb temperature of 75 F in the auditorium. Therefore, 265,000 Btu of sensible heat and 905,000 grains of moisture will be added to the air in the auditorium per hour.

Taking 0.24 as the specific heat of air, 2.4 Btu per pound of air will be required to raise the dry-bulb temperature from 65 to 75 F and $\frac{265,000}{2.4} = 110,400$ lb of air or 110,400 $\times$

$13.4 = 1,479,000$ cfh of air will be required. This is equivalent to $\frac{1,479,000}{1000 \times 60} = 24.7$ cfm per person.

The moisture content of the inside air as taken from a psychrometric chart is 76 grains per pound of dry air and that of the outside condition is 65 grains. The increase in moisture content will therefore be 11 grains per pound of dry air. Hence $\frac{905,000}{11.0} = 82,300$ lb of air at the specified condition will be required. This is equivalent to $82,300 \times 13.4 = 1,103,000$ cfh of air or $\frac{1,103,000}{1000 \times 60} = 18.4$ cfm of air per person.

The higher volume of 24.7 cfm per person will be required to keep the dry-bulb temperature from rising above the 75 F specified. The wet-bulb temperature will therefore not rise to the maximum of 65 F.

11 • The characteristics of air supplied to ventilate a room are:

Carbon dioxide concentration.....	4 parts per 10,000
Wet-bulb temperature.....	45.2 F
Dry-bulb temperature.....	55.0 F
Moisture content.....	29.0 grains per pound of dry air

a. What will be the dry-bulb temperature of the air in the room if it is occupied by five adults, if the air change, including both ventilation and infiltration, is 50 cu ft per minute, and assuming that there is no heat gain or loss to the room from any source other than from the occupants?

b. What will be the carbon dioxide concentration of the air in the room under these conditions?

c. What will be the moisture content of the air in the room under these conditions?

d. What will be the wet-bulb temperature and the relative humidity of the air in the room under these conditions?

e. What would the temperature of the incoming air have to be to give a room a dry-bulb temperature of 70 F?

a. The air change is 10 cu ft per minute per occupant. From the bottom chart of Fig. 6 at the intersection of an incoming air dry-bulb temperature of 55.0 F and a rate of air supply of 10 cu ft per minute per occupant, find by interpolation between the 70 F and 80 F adult curves the dry-bulb temperature of the air in the room to be 78.0 F.

b. From the top chart of Fig. 6 find the increase in CO₂ concentration to be 10 parts of CO₂ per 10,000 parts of air. Therefore, the air in the occupied room will contain 14 parts of CO₂ per 10,000.

c. From the center chart in Fig. 6 find by interpolation between the 70 F and 80 F adult curves the increase in moisture content to be 23 grains per pound of dry air for adults in 78 F air. This gives a resultant moisture content of the air in the room of 52 grains per pound of dry air.

d. From the psychrometric chart, Fig. 5, find the resulting wet-bulb temperature and relative humidity for 78 F dry-bulb and 52 grains of moisture to be 61.0 F and 37 per cent, respectively.

e. From the bottom chart, Fig. 6, find the required incoming air temperature to be 42 F dry-bulb.

12 • Referring to the A.S.H.V.E. Comfort Chart (Fig. 5), list the conditions (dry-bulb, wet-bulb, effective temperature, and humidity) which will produce comfort at each corner of the average winter comfort zone and of the average summer comfort zone.

Average winter comfort zone:

WET-BULB F	DRY-BULB F	RELATIVE HUMIDITY
58.5	64.5	70 per cent
51.5	67.5	30 per cent
59.0	79.0	30 per cent
67.0	74.0	70 per cent

Average summer comfort zone:

WET-BULB F	DRY-BULB F	RELATIVE HUMIDITY
62.0	68.0	70 per cent
54.0	72.0	30 per cent
63.5	85.0	30 per cent
71.5	78.5	70 per cent

Chapter 4

NATURAL VENTILATION

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces, supplemented in certain cases with mechanical forces, finds extensive application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

The natural forces available for the displacement of air in buildings are the wind and the difference in temperature of the air inside and outside the building. The arrangement and control of ventilating openings should be such that the two forces act coöperatively and not in opposition.

Wind Forces

In considering the use of natural wind forces for the operation of a ventilating system, account must be taken of (1) average and minimum wind velocities, (2) wind direction, (3) seasonal, daily and hourly variations in wind velocity and direction, and (4) local wind interference by buildings and trees.

Table 1, Chapter 8, gives values for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 7, 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 average wind velocities are seldom below 5 mph, there are many hours in each month during which the wind velocity is from 3 to 5 mph, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the hourly wind velocity falls much below 3 mph for more than 10 daylight hours per month. Usually a natural ventilating system should be designed to operate satisfactorily with a wind velocity of 3 to 6 mph, depending on locality.

The following formula 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:

$$Q = EA V \quad (1)$$

where

Q = air flow in cubic feet per minute.

A = free area of inlet (or outlet) openings in square feet.

V = wind velocity in feet per minute,

= miles per hour $\times 88$.

E = effectiveness of openings.

(E should be taken at from 50 to 60 per cent if the inlet openings face the wind and from 25 to 35 per cent if the inlet openings receive the wind at an angle.)

If outlet openings, where air leaves a building, are smaller than inlet openings, where air enters a building, the air will be less effective than indicated by the constant E .

The accuracy of the results obtained by the use of Formula 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-edge 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 following four places:

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).
3. In a monitor on the side opposite from the wind.
4. In roof ventilators or stacks exposed to the full force of the wind¹.

Forces due to Stack Effect²

The stack effect produced within a building is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4 \cdot A \sqrt{H (t_1 - t_2)} \quad (2)$$

where

Q = air flow in cubic feet per minute.

A = free area of inlets or outlets (assumed equal) in square feet.

H = height from inlets to outlets, in feet.

t_1 = average temperature of indoor air in height H , in degrees Fahrenheit.

t_2 = temperature of outdoor air, in degrees Fahrenheit.

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.

The height between inlets and outlets should be the maximum which the building construction will allow.

In some cases the necessary air flow will be known from the requirements of the building occupancy, and the area necessary for certain assumed temperature differences may be calculated. Or the areas may be fixed by the building construction, and the maximum air flow for various differences between indoor and outdoor temperatures may be calculated. In any case, the conditions which give the minimum air flow are those which control the design, as the system must have ample capacity even under the most unfavorable conditions which are those of mild or warm weather.

TYPES OF OPENINGS

The engineering problems of a natural ventilation system consist of the design, location, and control of ventilating openings to best utilize the

¹See *Airation of Industrial Buildings*, by W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).
²See *Neutral Zone in Ventilation*, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926), and *Predetermining Airation of Industrial Buildings*, by W. C. Randall and E. W. Conover (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

natural ventilation forces, in accordance with the requirements of building occupancy. The types of openings may be classified as:

1. Windows, doors, monitor openings, and skylights.
2. Roof ventilators.
3. Stacks connecting to registers.
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

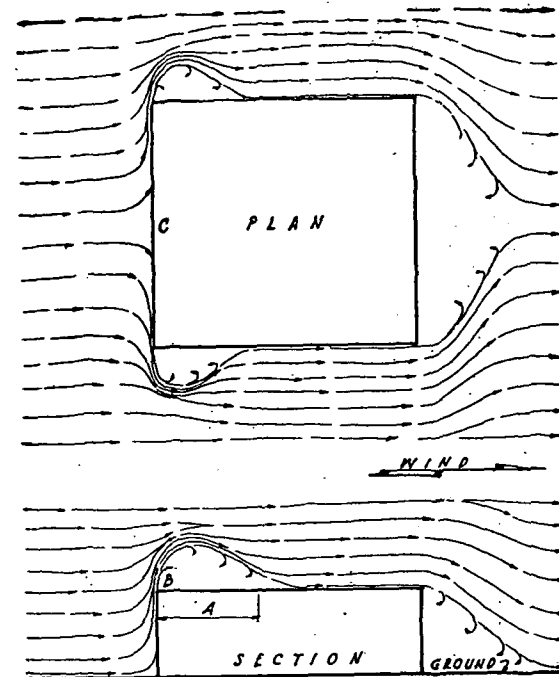


FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (A—LENGTH OF SUCTION AREA; B—POINT OF MAXIMUM INTENSITY OF SUCTION; C—POINT OF MAXIMUM PRESSURE)

various ways; they may open by sliding as in the ordinary double-hung windows, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top or bottom. Whatever the form and type of window used, the amount of clear area that can be made available is the factor of greatest importance in ventilation.

All types of sash (double-hung, top, center or bottom horizontal pivoted, or vertical pivoted) have about the same air flow capacity for the same clear area. Air leakage through closed windows is important during high winds (Chapter 6).

The proper distribution of air in occupied spaces is an element almost as important as that of sufficient air quantity. Advantageous pivoting of sash is very useful for securing good air distribution. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Door openings are seldom included in the ventilation calculations, though they may be of great value for extreme summer conditions, and should be considered in this connection as well as in garage design.

Skylight and monitor openings are of importance as these and the roof ventilators are outlets, while the lower windows are usually inlets on the windward side and outlets on the leeward side. In general the areas of inlets and of outlets should be about equal. It is important to make a check on this ratio in any installation, as any great excess of area of one set of openings over another means waste opening area. The operating devices used for sash, monitors, skylights and roof ventilators should be well selected as poor operating devices may defeat the entire design.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet, which is sensitive to wind action for producing additional flow capacity, and at the same time is subject to manual or automatic control by suitable dampers. The capacity of a ventilator at a constant wind velocity and temperature difference, depends upon four things: (1) its location on the roof, (2) the resistance it offers to air flow, (3) the area and location of openings provided for air inflow at a lower level, and (4) the ability of the ventilator head to utilize the kinetic energy of the wind for inducing flow by centrifugal or ejector action. Frequently one or more of these capacity factors is overlooked in a ventilator installation.

For maximum flow induction, a ventilator should be located on that part of the roof which receives the full wind without interference. (See Fig. 1.) This does not mean that no ventilators are to be installed within the suction region created by the wind jumping over the building, or in a light court, or on a low building between two high buildings. Ventilators are highly effective in such low-pressure areas, but their ejector action, caused by wind velocity, is of little importance in these locations, and hence their size should be increased proportionally.

Ventilator resistance depends on (1) type of inlet, (2) area of openings and passages, and (3) number of turns or changes of direction of the air flow. The inlet grille, if any, should have ample free area, and the ventilator should always be provided with a taper-cone inlet in order to produce the effect of a bell-mouth nozzle (flow coefficient 0.97) rather than that of a square-entrance orifice (flow coefficient 0.60). In other words, the grilles should be oversize as compared with the ventilator, and they should be connected by tapering collars. If the ventilator head construction produces changes in the direction of air flow, the area of the flow passages should be increased accordingly.

Air inlet openings at lower levels in the building are of course necessary for the economical use of ventilator capacity. The inlet openings should be at least equal to, and preferably twice as great as the combined throat areas of all roof ventilators. The air discharged by a roof ventilator

depends on wind velocity and temperature difference, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Several types of roof ventilators are shown in Figs. 2 to 11. These may be classified as *stationary*, Figs. 2 to 6, *pivoted* or *oscillating*, Figs. 7 to 9, or *rotating*, Figs. 10 and 11. When selecting unit ventilators, some attention should be paid to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibilities of noise from dampers or other moving parts, and possible maintenance costs.

It should be kept in mind that a suitable combination of roof ventilators with mechanical ventilation frequently offers the best solution of a ventilating problem. The natural ventilation units may be used to supplement power driven supply fans, and under favorable weather conditions it may be possible to shut down the power driven units. Where low operating costs are very important, such a combination has great advantages. Roof ventilators with built-in electric fans are attracting increased attention because they combine the advantages of low installation and operating cost with those of continuous service.

Controls

In connection with any combination between natural and fan ventilation, the controls are of importance. Both the fans and the ventilator dampers may be controlled by some combination of three methods: (1) hand operation, (2) thermostat operation, and (3) control by wind velocity. 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 are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference (the so-called *gravity* action). While their openings projecting above the roof are not provided with any special construction for developing suction by the action of the wind, the plain vertical opening is also effective in this respect. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction.

Stacks are applicable particularly in the case of schools, apartments, residences and small office buildings. Partitions interfere with general air circulation, and some type of outlet from each room is necessary. If the building is not too tall, and the requirements of occupancy are moderate, a system of stacks with registers in each room may be more economical than a system of mechanical ventilation employing fans. In making the comparison, however, the building space occupied by the stacks should be considered.

With little or no wind, chimney effect or temperature difference will produce outflow through the stacks and an equal inflow through windows

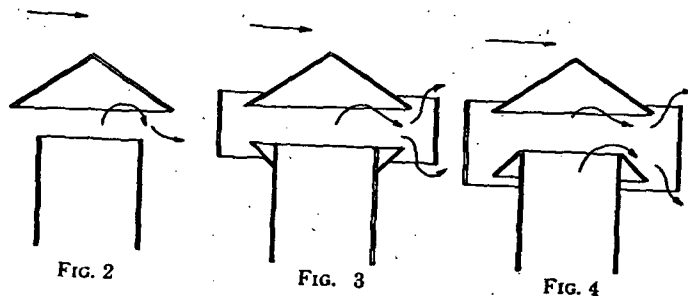


FIG. 5
SIX COMMON TYPES OF STATIONARY VENTILATORS

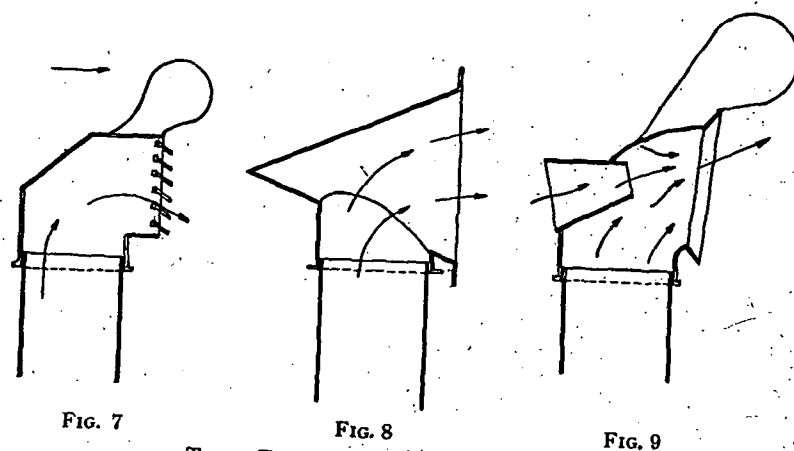
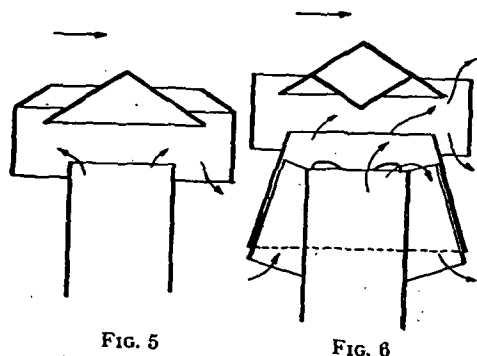


FIG. 8
THREE TYPICAL OSCILLATING VENTILATORS

in all sides of the building. With wind, the inductive force at the top of ventilating shafts is more powerful than that on the leeward side of the building, so that air is drawn in through leeward openings by a combination of the forces of wind and temperature difference. On the windward side, the direct forcing pressure of the wind is of course added to the temperature difference effect. Thus forces are available for causing inflow at practically every window of such a building. Adequacy of stack size must, of course, be provided.

PRINCIPLES OF AIR FLOW CONTROL

The air flow through a ventilation opening depends on the two factors already discussed, namely, (1) the natural forces available, (2) the openings available, and the resistance to flow offered by these openings. The design problem includes, of course, a determination of the *desired air*

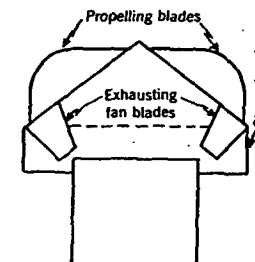
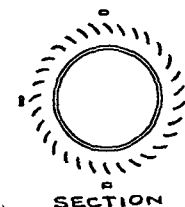
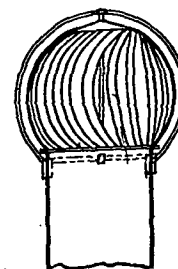


FIG. 10. ROTATING VENTILATORS

FIG. 11.

quantity and distribution in order that the openings may be properly placed.

The purpose of ventilation is to carry off either excess heat or air impurities, and the desired air quantities depend upon the amount of heat or of impurities present. The amount of heat can be determined, in the case of forge shops for example, from the amount of fuel burned, which in turn is based upon the production capacity for which the building is being designed. In the case of foundries, the heat given off by the metal in cooling from the molten state can be used. In some instances, not all of the heat may be dissipated to the air, but a fair estimate of the amount to be removed by the air can usually be made.

The next step is to select the temperature difference to be maintained. Knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain this temperature difference can be determined by means of the following equation:

$$H = \frac{c Q D}{V} \quad (3)$$

where

$c = 0.24$ = specific heat of air.

V = specific volume of the air, cubic feet per pound, about 13.5. (See Chapter 1.)

H = heat to be carried off, in Btu per minute.

Q = air flow in cubic feet per minute.

D = inlet-outlet temperature difference in degrees Fahrenheit.

For disposing of air impurities, the required air flow must be such that the outside air will dilute the impurities to a degree that they are no longer objectionable. For human occupancy, such as in auditoriums and classrooms, 10 cfm per person is usually taken as the minimum of outside air necessary for ventilation (see Chapter 3). For garage ventilation, sufficient air must be admitted to dilute the carbon monoxide content of the indoor air to 1 in 10,000 (see Garage Ventilation in this Chapter).

Air quantity and quality are not the only requirements. For human occupancy, air distribution is important. In ventilation the air distribution is almost entirely a matter of the number, the design, and the location of inlets and outlets. In locating openings, special precautions should be taken against the formation of dead air spaces or *pockets* within the zone of occupancy.

Suggested methods for estimating the air flow due to temperature difference alone and to wind alone 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 same openings have been assumed in both cases, and since the resistance to flow through the openings varies approximately with the square of the velocity³, this resistance becomes a limiting factor as the flow through the openings is increased.

Recent investigations^{1,2} show that the total flow is only 10 per cent above the flow caused by the greater force when the two forces are nearly equal, and this percentage decreases rapidly as one force increases above the other. Tests on roof ventilators indicate that this is too conservative in the direction of low total flow quantities, but there is in any case a large judgment factor involved. The wind velocity and direction, the outdoor temperature, or the indoor activities cannot be predicted with certainty, and great refinement in calculations is therefore not justified. When designing for winter conditions, an added variable is the heat lost by direct flow through walls and windows and by infiltration.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per lb is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Temperature differences are 10 F in summer and 30 F in winter, and the wind velocity is 5 mph in summer and 8 mph in winter. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution. The system must be designed for the summer conditions as these are the more severe. The heat to be removed per minute is:

$$H = \frac{15}{60} \times 7.75 \times 18,000 = 34,875 \text{ Btu.}$$

By Equation 3, the air flow required to remove this heat with a temperature difference of 10 deg is:

$$Q = \frac{VH}{cD} = \frac{13.5 \times 34,875}{0.24 \times 10} = 196,172 \text{ cfm.}$$

³This is true for *turbulent* flow only. It would be more correct to state that the resistance varies approximately with V^2 for high to moderate velocities, with $V^{1.8}$ for moderate to low velocities, and with the first power of the velocity for very low velocities through small openings.

This is equal to 19.6 air changes per hour. The assumption is made that the average temperature difference between indoors and outdoors is the same as the temperature rise of the air from the inlet opening to the outlet opening. Actually, the latter difference is larger and so the value of 19.6 air changes per hour is conservative as it allows for more cooling than is necessary for an *average* temperature difference of 10 deg.

If 196,172 cfm are to be circulated by the force of the temperature difference alone, the area of opening would be, by Equation 2:

$$A = \frac{Q}{9.4 \sqrt{H(t_1 - t_2)}} = \frac{196,172}{9.4 \sqrt{30 \times 10}} = 1,205 \text{ sq ft.}$$

If this area of openings were provided, a wind velocity of 5 mph, acting alone, would produce a flow according to Equation 1, of:

$$Q = EAV = 0.50 \times 1,205 \times 5 \times 88 = 265,100 \text{ cfm.}$$

If the inlet openings do not face the wind, but are at an angle with it, about half this amount may be considered to flow.

A factor of judgment must now be exercised in making the selection of the area of openings to be specified. Apparently 1205 sq ft are a very generous allowance because either a direct wind of 5 mph or an average temperature difference of 10 deg acting alone will more than suffice to carry away the heat, and when the two forces are acting together, the system may have an excess capacity of 25 per cent to 50 per cent, especially if the outlets are made up partially of roof ventilators which employ the force of the wind for producing a suction effect. On the other hand, the wind may at times come from an unfavorable direction, or its velocity may fall below 5 mph or the building construction may not permit a full 2400 sq ft of inlet window area and an equal amount of monitor or roof ventilator outlet area. In case the two sets of openings are not equal, their effectiveness is reduced.

From this example it must be apparent that while formulas may furnish a reliable guide, the final solution of a problem of natural ventilation requires a common sense analysis of local conditions to supplement and to modify the dictates of the formulas.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings 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 of occupancy.
2. 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.
3. Roof ventilators should be located 20 to 40 ft apart each way and preferably on the ridge of the roof. The closer spacings are used when ventilating rooms with low ceilings.
4. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.
5. 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.

6. 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.

7. 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.

8. 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.

9. 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.

10. 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. On the other hand, where the direction of the wind is quite variable, it may be stated as a general principle that windows should be arranged in sidewalls and monitors so that there will be approximately equal area on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force of the wind, and others opposed to a suction force, and effective movement through the building will be assured.

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. Special consideration should be given to the possibility of sidewall or monitor windows being closed on account of weather conditions. Such possibilities favor roof ventilators and specially designed stormproof inlets.

MEASUREMENT OF NATURAL AIR FLOW

The determination of the performance of any ventilating system involves measurements which are not easy to make. The difficulties are increased in the case of natural ventilation, since the motive forces and the air velocities are very small. The measurements necessary for giving the capacity of a system are (1) velocity of the wind, (2) velocity of the air through inlet and outlet openings, (3) outdoor air temperature, and (4) average indoor air temperature.

Measuring Wind Velocity. The cup-type of anemometer as used for Weather Bureau observations is sufficiently accurate for this measurement. Some more accurate instruments as well as direct-reading types have been developed for airport service, but for ventilation work it is the average wind velocity over a long period which determines the capacity of the system. Hence the use of the Weather Bureau instrument, with an observation period of one hour or more, is satisfactory. If observations of wind direction are required, these should be taken by observing a sensitive weather vane at frequent intervals (about every 5 minutes) during the same period.

Velocity of Air Through Openings. The vane type anemometer is the most practical instrument for this measurement.

Use a small (4 in.) low-speed anemometer, and correct all readings according to a recent calibration. Mount the anemometer in a strap iron clamp with a long handle for convenience. Divide each opening into 5 in. squares (by string or wire) and hold the anemometer in the center of each square for a definite period of from 15 to 30 seconds. Record the result of the traverse as soon as completed and start another one immediately. A series of traverses over a period of one hour, or the full period covered by the wind velocity observations with a fairly steady wind, may be considered a satisfactory test for that wind velocity. It is preferable to have an anemometer observer at each opening. If the opening is covered by a grille or register, use the proper correction factors (see Chapter 43).

Outdoor Temperature. It is easy to make an error of 1 to 5 deg in observing the outdoor air temperature. An accurate thermometer, calibrated in 1 deg divisions should be used. The thermometer should be mounted in the shade at about mid-height of the building and not too near the building wall or adjacent to an air outlet. The heat from a wall or roof which has been exposed to the sun is easily transmitted to a thermometer, with resulting high readings.

Average Indoor Temperature. It is important to note that the capacity of an opening (such as roof ventilator) does *not* depend on the difference in the temperatures measured adjacent to the opening. It depends rather on the difference between the *average* temperature of the column of air inside the building and that outside. Indoor temperatures should therefore be observed at various heights to secure a good average.

DAIRY BARN VENTILATION*

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 is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be

*For additional information on this subject refer to *Technical Bulletin*, U. S. Department of Agriculture (1630), by M. A. R. Kelley.

Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

Cow Barn Ventilation, by Alfred J. Offner (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).

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 cfm 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 per cu ft of barn space, and 0.197 to 0.305 Btu per hour per sq ft of barn exposure.

GARAGE VENTILATION^{6, 6}

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 over-emphasized. 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 for Heating and Ventilating Garages, adopted in 1929 and revised 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.

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

⁶Code for Heating and Ventilating Garages (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929). (A.S.H.V.E. Reprint, January, 1935).

⁶Airation Study of Garages, by W. C. Randall and L. W. Leonhard (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1930).

⁶Carbon Monoxide Concentration in Garages, by A. S. Langsdorf and R. R. Tucker (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1930).

⁶Carbon 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).

⁶Carbon 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).

⁶Carbon 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).

⁶Carbon 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).

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 below:

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 cfm, with an average rate of 35 cfm.
5. An air change of 350,000 cfm per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

PROBLEMS IN PRACTICE

1 ● What factors may make the adoption of a system of ventilation depending upon wind movement inadvisable in new construction?

- a. Variation in direction of wind.
- b. Variation in wind velocity.
- c. Inability to clean incoming air.
- d. Inability to control location, size and shape of buildings on adjacent property.
- e. Unsatisfactory warming of incoming air during cold weather.

2 ● a. What factors are important in the location and control of ventilating openings?

b. What types of ventilating openings are best suited to a proper distribution of the air supplied?

- a. The proper distribution of air as required by the occupants, and the best utilization of natural ventilating forces. The general rules on page 97 apply particularly to these factors.
- b. Windows with swinging sash and openings with deflectors may be used to direct air to the points desired.

3 ● a. What is the best location for ventilating openings?

b. How are the sizes of ventilating openings determined for proper air supply?

- a. Inlet openings should be low and facing the prevailing winds where possible. Outlet openings should be high and on the side opposite the prevailing winds.
- b. For simple openings use Formula 1:

$$Q = EAV$$

and for stacks use Formula 2:

$$Q = 9.4 A \sqrt{H(t_1 - t_2)}$$

The use of these formulae is illustrated in Example 1 of the text of this chapter. Inlet and outlet areas should be approximately the same for best results.

4 • a. What are the advantages of roof ventilators?

b. How are proper sizes determined for roof ventilators?

a. Roof ventilators offer the best utilization of the inductive force of the wind, and they may be very economically fitted with built-in fans to supply the necessary circulation when the force of the wind is not sufficient.

b. Because of the many factors affecting the flow through roof ventilators no accurate formula can be given. It is usual practice to make the combined throat area of all roof ventilators between one-half area and full area of the air inlets as determined by Formula 1.

5 • What methods of control are used in ventilating systems?

Hand control, control by a thermostat located in the ventilated space or in the ventilator, or wind velocity control designed to keep the air discharge constant regardless of wind velocity.

6 • How is the quantity of air required for a building determined?

Sufficient air must be supplied to carry away the heat and impurities generated within a building. The temperature rise and concentration of impurities in the exhaust air must be held within specified limits. (See Example 1.)

7 • What measurements are necessary to determine the capacity of a ventilating system?

Wind velocity and air velocities through openings, determined by suitable cup anemometers; outdoor air temperatures, measured by a shaded thermometer not near objects heated by the sun or near exhaust air openings; indoor air temperatures, measured at various heights to secure a good average.

8 • How much air must be supplied for dissipating the heat generated in a dairy barn housing 100 cows if the outside temperature is 20 F and the inside temperature is to be maintained at 45 F?

The total heat generated is $100 \times 3000 = 300,000$ Btu per hour or 5,000 Btu per minute. Then from Formula 3,

$$\begin{aligned} Q &= \frac{HV}{CD} \\ &= \frac{5000 \times 13.5}{0.24 \times (45 - 20)} \\ &= 11,250 \text{ cu ft per minute.} \end{aligned}$$

This amount of air should also keep down humidity and odors.

9 • a. What precaution is necessary in the ventilation of garages using natural ventilation?

b. How much window area is required for a garage with 50 x 100 sq ft floor area if natural ventilation is used?

a. The carbon monoxide content of the air should be kept below 1 part in 10,000 and windows should be kept open at all times.

b. The window area should aggregate 5 per cent of the floor area.

$$0.05 \times 50 \times 100 = 250 \text{ sq ft of window area.}$$

This area should be evenly distributed along two sides of the building.

Chapter 5

HEAT TRANSMISSION COEFFICIENTS AND TABLES

Heat Transfer, Calculations for Transmission Losses, Areas Where Transmission Losses Occur, Coefficients of Transmission, Table of Conductivities and Conductances, Tables of Over-all Coefficients of Heat Transfer for Typical Building Constructions

TO maintain specified inside temperature conditions and determine the type of plant required, it is essential to know the transmission losses of a structure and consider them in conjunction with the infiltration losses.

Whenever a difference in temperature exists between the two sides of any structural material, such as a wall or roof of a building, a transfer of heat takes place through that material. When the inside temperature is the higher, heat enters the inside surface of the wall by radiation, conduction and convection, because the air and objects within the building are always warmer than the inside surface of the wall when the inside air temperature t is greater than the outside air temperature t_o . This heat must then pass through the material of the wall from the inside to the outside surface, and is finally given off from the outside surface by radiation, conduction and convection, provided, of course, that equilibrium has been established and all four temperatures are constant. If the outside temperature is the higher, the reverse process takes place.

CALCULATIONS FOR TRANSMISSION LOSSES

The calculations for heat transmission losses are made by multiplying the area A in square feet of wall, glass, roof, floor, or material through which the loss takes place, by the proper coefficient U for such construction or material and by the temperature difference between the inside air temperature t at the proper level (in many cases not the *breathing-line*) and the outside air temperature t_o . Therefore,

$$H_t = AU(t - t_o) \quad (1)$$

where

H_t = Btu per hour transmitted through the material of the wall, glass, roof or floor.

A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured. (Use the net inside or heated surface dimensions in all cases.)

$t - t_o$ = temperature difference between inside and outside air, in which t must always be taken at the proper level. Note that t may not be the *breathing-line* temperature in all cases.

Heat is lost from a building by transmission through parts of the structure which separate heated spaces from the outside air or from unheated colder spaces within the building. In general, five parts are involved: (1) outside walls; (2) outside glass; (3) inside walls or partitions next to unheated spaces; (4) ceilings of upper floors, either below a cold attic space or as the underside of a roof slab; and (5) floors of heated rooms above an unheated space.

The net inside wall surface is usually determined by reference to the scale plans and elevations of the building concerned. In some cases, of course, the actual building may have to be measured. The total area of all outside openings which are occupied by windows and doors is accurately measured and listed as *glass*. The *glass* area is then deducted from the total outside wall area for each room and the difference is the net wall area. If there are no partitions, measure from the inside face of one wall to the inside face of the next wall. The areas of walls, ceilings and floors next to cold or unheated spaces are found, of course, by taking the inside dimensions of such areas, measured on the heated side.

COEFFICIENTS OF TRANSMISSION

The coefficients of transmission may be determined by means of the guarded hot box or the Nicholls heat meter described in Chapter 43, or they may be calculated from fundamental constants. Because of the unlimited number of combinations of building materials, it would be impractical to attempt to determine by test the heat transmission coefficient of every type of construction in use; consequently, in most cases it is advisable to calculate these coefficients.

Symbols

The following symbols are used in the heat transmission formulae in this chapter:

U = thermal transmittance or over-all coefficient of heat transmission; the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 deg F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

k = thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 deg F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C = thermal conductance; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a non-homogeneous material for the thickness or type under consideration for a difference in temperature of 1 deg F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plaster board and hollow clay tile.

f = film or surface conductance; the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = thermal conductance of an air space; the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of

an air space for a temperature difference of 1 deg F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, i.e.:

$$\frac{1}{U} = \text{over-all or air-to-air resistance.}$$

$$\frac{1}{k} = \text{internal resistivity.}$$

$$\frac{1}{C} = \text{internal resistance.}$$

$$\frac{1}{f} = \text{film or surface resistance.}$$

$$\frac{1}{a} = \text{air-space resistance.}$$

Fundamental Formulae

The formula of the over-all coefficient for a simple wall x inches thick is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{x}{k} + \frac{1}{f_o}} \quad (2)$$

and for a compound wall of several materials having thicknesses in inches of x_1, x_2, x_3 , etc., the coefficient is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o} + \text{etc.}} \quad (3)$$

In the case of air-space construction, an air-space coefficient for each air space must be inserted in either Equation 2 or 3. Thus for a simple wall with one air space,

$$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a} + \frac{x_2}{k_2} + \frac{1}{f_o}} \quad (4)$$

and for a wall of several air spaces having conductances of a_1, a_2, a_3 , etc., the coefficient is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{1}{a_1} + \frac{x_2}{k_2} + \frac{1}{a_2} + \frac{x_3}{k_3} + \frac{1}{a_3} + \frac{x_4}{k_4} + \frac{1}{f_o} + \text{etc.}} \quad (5)$$

With certain special forms of materials which have irregular air spaces (such as hollow tile) or are otherwise non-homogeneous, it is necessary to use the conductance (C) for the unit construction, in which case $\frac{x}{k}$ is replaced by $\frac{1}{C}$.

As in the case of the simple wall, f_i and f_o are always the inside and outside surface coefficients for the two materials in contact with air. If

the air is still (no wind), then for the same material f_i and f_o are the same, and $f_i = f_o$; but if the outside air is in motion, then f_o is always greater than f_i and will increase as the wind velocity increases. Values for f_i in still and moving air have been determined for various building materials at the University of Minnesota under a cooperative research agreement with the Society¹. The range of values for ordinary building materials is comparatively small and for practical purposes may be assumed constant for either still air or any given wind velocity, particularly in view of the fact that the surface resistances usually comprise only a small part of the total resistance of the construction, except in the case of thin, highly conductive walls. In determining basic heat transmission values for building construction, it is customary to use that value of f_o which will occur when a 15-mph wind blows parallel to the outer surfaces considered.

TABLE 1. CONDUCTANCES OF AIR SPACES^a AT VARIOUS MEAN TEMPERATURES

MEAN TEMP. DEG. FAHR.	CONDUCTANCES OF AIR SPACES FOR VARIOUS WIDTHS IN INCHES					
	0.128	0.250	0.364	0.493	0.713	1.00
20	2.300	1.370	1.180	1.100	1.040	1.030
30	2.385	1.425	1.234	1.148	1.080	1.070
40	2.470	1.480	1.288	1.193	1.125	1.112
50	2.560	1.535	1.340	1.242	1.168	1.152
60	2.650	1.590	1.390	1.295	1.210	1.195
70	2.730	1.648	1.440	1.340	1.250	1.240
80	2.819	1.702	1.492	1.390	1.295	1.280
90	2.908	1.757	1.547	1.433	1.340	1.320
100	2.990	1.813	1.600	1.486	1.380	1.362
110	3.078	1.870	1.650	1.534	1.425	1.402
120	3.167	1.928	1.700	1.580	1.467	1.445
130	3.250	1.980	1.750	1.630	1.510	1.485
140	3.340	2.035	1.800	1.680	1.550	1.530
150	3.425	2.090	1.852	1.728	1.592	1.569

^aThermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

The conductances of air spaces at various mean temperatures and widths, for ordinary building materials, are given in Table 1. These results were likewise obtained at the University of Minnesota under a cooperative research agreement with the Society.

Values for k and C , the conductivity and conductance of building materials and insulations, are given in Table 2 as taken from the published values of various investigators. It should be noted that values of k and C as well as of U are dependent on the mean temperature, and it is therefore desirable that the investigator determine heat-transmission values under conditions approximating those existing under actual conditions. Recommended values for calculating the coefficients of transmission of various types of construction are marked by an asterisk in Table 2.

¹Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS Vol. 36, 1930). See also references at end of chapter.

CHAPTER 5—HEAT TRANSMISSION COEFFICIENTS AND TABLES

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS^a

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	DENSITY (LB PER CU FT)	MEAN TEMP. (DEG. FAHR.)	CONDUCTIVITY (k) OR CONDUCTANCE (C)	RESISTIVITY OR RESISTANCE (R)	AUTHORITY
MASONRY MATERIALS						
Brick	Common	—	—	5.00*	0.20	—
	Face	—	—	9.20*	0.11	—
	Damp or wet	—	—	5.00*	0.20	(2)
Brickwork	Typical	110.0	75	12.00*	0.08	—
Cement Mortar	Typical	—	—	5.20*	0.19	(3)
Cinder Concrete	Typical (8 in.)	—	—	0.621*	1.61	—
Cinder Blocks	Typical (12 in.)	—	—	0.51*	1.96	—
	Typical	143.0	69	12.00*	0.08	(4)
Concrete	1-2-4 mix	—	—	9.46	0.11	—
	Various ages and mixes ^d	—	—	11.35 to 16.36	—	(5)
	Cellular	40.0	75	1.06	0.94	(3)
	"	50.0	75	1.44	0.69	(3)
	"	60.0	75	1.80	0.56	(3)
	"	70.0	75	2.18	0.46	(3)
	Typical gypsum fiber concrete, 87.5% gypsum and 12.5% wood chips	51.2	74	1.66*	0.60	(4)
	Special concrete made with an aggregate of hardened clay—1-2-3 mix	101.0	70	3.98	0.25	(3)
Concrete Blocks ^e	Typical (8 in.)	—	—	1.001*	1.00	—
	(12 in.)	—	—	0.801*	1.25	—
	Special concrete block made with an aggregate of hardened clay—4 x 8 x 16 in., 3 cores, 18% voids	74.0	—	0.66†	1.51	(3)
	Special concrete block made with an aggregate of hardened clay—8 x 8 x 16 in., 4 cores, 35% voids	74.5	—	0.30†	3.33	(3)
Stone	Typical	—	—	12.50*	0.08	—
Stucco	Typical	—	—	12.00*	0.08	—
Tile	Typical hollow clay (4 in.)	—	—	1.001*	1.00	—
	" " " (6 in.)	—	—	0.64†	1.57	—
	" " " (8 in.)	—	—	0.60†	1.67	—
	" " " (10 in.)	—	—	0.58†	1.72	—
	" " " (12 in.)	—	—	0.40†	2.50	—
	" " " (16 in.)	—	—	0.31†	3.23	—
	Hollow clay (2 in.) 1/2-in. plaster both sides	120.0	110	1.00†	1.00	(2)
	Hollow clay (4 in.) 1/2-in. plaster both sides	127.0	100	0.60†	1.67	(2)
	Hollow clay (6 in.) 1/2-in. plaster both sides	124.3	105	0.47†	2.13	(2)
	Hollow gypsum (4 in.)	51.8	70	0.461*	2.18	—
	Solid gypsum	75.6	76	1.66	0.60	(4)
	Solid gypsum	—	—	2.96	0.34	(4)
Tile or Terrazzo	Typical flooring	—	—	12.00*	0.08	—

AUTHORITIES:

¹U. S. Bureau of Standards, tests based on samples submitted by manufacturers.

²A. C. Willard, L. C. Lichty, and L. A. Harding, tests conducted at the University of Illinois.

³J. C. Peebles, tests conducted at Armour Institute of Technology, based on samples submitted by manufacturers.

⁴F. B. Rowley, tests conducted at the University of Minnesota.

⁵A.S.H.V.E. Research Laboratory.

⁶E. A. Allcutt, tests conducted at the University of Toronto.

⁷Loes and Chorlton.

*Recommended conductivities and conductances for computing heat transmission coefficients.

†For thickness stated or used on construction, not per 1-in. thickness.

*For additional conductivity data see Table 14, Page 63, 1934 A.S.H.V.E. Data Book.

*Recommended value. See Heating, Ventilating and Air Conditioning, by Harding and Willard, revised edition, 1932.

*One air cell in the direction of heat flow.

*See A.S.H.V.E. Research Paper, Conductivity of Concrete, by F. C. Houghton and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

*The 6-in., 8-in., and 10-in. hollow tile figures are based on two cells in the direction of heat flow. The 12-in. hollow tile is based on three cells in the direction of heat flow. The 16-in. hollow tile consists of one 10-in. and one 6-in. tile, each having two cells in the direction of heat flow.

†Not compressed.

*Roofing, 0.15-in. thick (1.34 lb per sq ft), covered with gravel (0.83 lb per sq ft), combined thickness assumed 0.25.

TABLE 2. CONDUCTIVITIES (*k*) AND CONDUCTANCES (*C*) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	Density (Lb per Cu Ft)	MEAN TEMP. (DEG FAHR)	CONDUCTIVITY (<i>k</i>) OR CONDUCTANCE (<i>C</i>)	RESISTIVITY ($\frac{1}{k}$) OR RESISTANCE (<i>R</i>)	AUTHORITY
INSULATION—BLANKET OR FLEXIBLE TYPES FIBER	Typical	—	—	0.27*	3.70	—
	Chemically treated wood fibers held between layers of strong paper/	3.62	70	0.25	4.00	(3)
	Reel grass between strong paper/	4.60	90	0.26	3.85	(1)
		3.40	90	0.25	4.00	(1)
	Fabric with non-metallic reflective surface (1/2 in. thick)	—	70	0.33†	3.03	(3)
	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)
	Jute fiber/	6.70	75	0.25	4.00	(3)
INSULATION—SEMI-RIGID TYPE FIBER	Felted cattle hair/	13.00	90	0.26	3.84	(1)
		11.00	90	0.26	3.84	(1)
	Flax/	12.10	70	0.30	3.33	(3)
	Flax and rye/	13.60	90	0.32	3.12	(1)
	Felted hair and asbestos/	7.80	90	0.28	3.57	(1)
	75% hair and 25% jute/	6.30	90	0.27	3.70	(1)
	50% hair and 50% jute/	6.10	90	0.26	3.85	(1)
	Jute/	6.70	75	0.25	4.00	(3)
	Felted jute and asbestos/	10.00	90	0.37	2.70	(1)
	Compressed peat moss	11.00	70	0.26	3.84	(3)
INSULATION—LOOSE FILL OR BAT TYPE FIBER	Made from celba fibers/	1.90	75	0.23	4.35	(3)
		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)
	Fibrous material 25 to 30 microns in diameter, made from virgin bottle glass	1.50	75	0.27	3.70	(3)
	Made from combined silicate of lime and alumina	4.20	72	0.24	4.17	(3)
	Made from expanded aluminum-magnesium silicate	6.20	42	0.32	3.12	(3)
	Cellular, dry	30.00	90	1.00	1.00	(1)
	"	24.00	90	0.77	1.30	(1)
	"	18.00	90	0.59	1.69	(1)
GLASS WOOL	"	12.00	90	0.44	2.27	(1)
	Flaked, dry and fluffy/	34.00	90	0.60	1.67	(1)
	"	26.00	90	0.52	1.92	(1)
	"	24.00	75	0.48*	2.08	(3)
	"	19.80	90	0.35	2.86	(1)
	"	18.00	75	0.34	2.94	(3)
	"	18.00	75	0.27*	3.70	(3)
	All forms, typical	8.10	90	0.31	3.22	(1)
	About 1/2-in. particles	21.00	90	0.30	3.33	(1)
	Fibrous material made from rock	18.00	90	0.29	3.45	(1)
GRANULAR	"	14.00	90	0.28	3.57	(1)
	"	10.00	90	0.27*	3.70	(1)
	Rock wool with a binding agent	14.50	77	0.33	3.03	(3)
	Rock wool with flax, straw pulp, and binder	14.50	75	0.38	2.63	(3)
	Rock wool with vegetable fibers	11.50	72	0.31	3.22	(3)
	Various	12.00	90	0.41	2.44	(1)
	Various from planer	8.80	90	0.41	2.44	(1)
	From maple, beech and birch (coarse)	13.20	90	0.36	2.78	(1)
	Redwood bark	3.00	90	0.31	3.22	(1)
		—	—	—	—	—
GYPSUM	Cellular, dry	30.00	90	1.00	1.00	(1)
	"	24.00	90	0.77	1.30	(1)
MINERAL WOOL	Various	12.00	90	0.41	2.44	(1)
	Various from planer	8.80	90	0.41	2.44	(1)
REGRANULATED CORK	From maple, beech and birch (coarse)	13.20	90	0.36	2.78	(1)
	Redwood bark	3.00	90	0.31	3.22	(1)
ROCK WOOL	Rock wool with a binding agent	14.50	77	0.33	3.03	(3)
	Rock wool with flax, straw pulp, and binder	14.50	75	0.38	2.63	(3)
SAWDUST	Various	12.00	90	0.41	2.44	(1)
	Various from planer	8.80	90	0.41	2.44	(1)
SHAVINGS	From maple, beech and birch (coarse)	13.20	90	0.36	2.78	(1)
	Redwood bark	3.00	90	0.31	3.22	(1)
INSULATION—RIGID CORKBOARD	Typical	—	—	0.30*	3.33	—

For notes see Page 107.

TABLE 2. CONDUCTIVITIES (*k*) AND CONDUCTANCES (*C*) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	DENSITY (Lb per Cu Ft)	MEAN TEMP. (DEG FAHR)	CONDUCTIVITY (k) OR CONDUCTANCE (C)	RESISTIVITY ($\frac{1}{k}$) OR RESISTANCE (R)	AUTHORITY	
INSULATION—RIGID —Continued CORKBOARD	No added binder.....	14.00	90	0.34	2.94	(1)	
	" " ".....	10.60	90	0.30	3.33	(1)	
	" " ".....	7.00	90	0.27	3.70	(1)	
	" " ".....	5.40	90	0.25	4.00	(1)	
	Asphaltic binder.....	14.50	90	0.32	3.12	(1)	
FIBER	Typical.....	—	—	0.33*	3.03	—	
	Core of fiber board coated two sides with non-metallic reflective surface ($\frac{1}{2}$ in. thick).....	23.4	70	0.27†	3.70	(3)	
	Fiber board coated one side with non-metallic reflective surface ($\frac{1}{2}$ in. thick).....	—	75	0.49†	2.04	(3)	
	Made from chemically treated wood fiber.....	20.00	70	0.36	2.78	(3)	
	Made from chemically treated wood and vegetable fibers.....	25.00	75	0.38	2.63	(3)	
	Chemically treated hog hair covered with film of asphalt.....	10.00	75	0.28	3.57	(3)	
	Made from corn stalks.....	15.00	71	0.33	3.03	(3)	
	" " exploded wood fiber.....	17.90	78	0.32	3.12	(4)	
	" " hard wood fibers.....	15.20	70	0.32	3.12	(3)	
	Insulating plaster 9/10-in. thick applied to $\frac{3}{4}$ -in. plaster board base.....	54.00	75	1.07†	0.93	(3)	
	Made from horioce roots.....	16.10	81	0.34	2.94	(3)	
	Made from 85% magnesia and 15% asbestos.....	19.30	86	0.51	1.96	(1)	
	Made from shredded wood and cement.....	24.20	72	0.46	2.17	(1)	
	" " sugar cane fiber.....	13.50	70	0.33	3.03	(3)	
	Sugar cane fiber insulation blocks encased in asphalt membrane.....	13.80	70	0.30	3.33	(3)	
	Made from wheat straw.....	17.00	68	0.33	3.03	(3)	
	" " wood fiber.....	15.90	72	0.33	3.03	(3)	
	" " " ".....	15.00	70	0.33	3.03	(3)	
	" " " ".....	—	52	0.33	3.03	(6)	
	" " " ".....	8.50	72	0.29	3.45	(3)	
	" " " ".....	15.20	—	0.33	3.03	(3)	
	" " " ".....	16.90	90	0.34	2.94	(1)	
	BUILDING BOARDS ASBESTOS	Compressed cement and asbestos sheets.....	123.00	86	2.70	0.37	(1)
		Corrugated asbestos board.....	20.40	110	0.48	2.08	(2)
		Pressed asbestos mill board.....	60.50	86	0.84	1.19	(1)
		Sheet asbestos.....	48.30	110	0.29	3.45	(2)
		Gypsum between layers of heavy paper.....	62.80	70	1.41	0.71	(3)
GYPSUM	Rigid, gypsum between layers of heavy paper ($\frac{1}{2}$ -in. thick).....	53.50	90	2.60†	0.38	(1)	
	Gypsum mixed with sawdust between layers of heavy paper (0.39-in. thick).....	60.70	90	3.60†	0.28	(1)	
	($\frac{1}{4}$ in.).....	—	—	3.73*	0.27	—	
	($\frac{1}{2}$ in.).....	—	—	2.82*	0.35	—	
	—.....	—	—	—	—	—	
PLASTER BOARD	Compressed cement and asbestos sheets.....	123.00	86	2.70	0.37	(1)	
	Corrugated asbestos board.....	20.40	110	0.48	2.08	(2)	
	Pressed asbestos mill board.....	60.50	86	0.84	1.19	(1)	
	Sheet asbestos.....	48.30	110	0.29	3.45	(2)	
	Gypsum between layers of heavy paper.....	62.80	70	1.41	0.71	(3)	
ROOFING CONSTRUCTION ROOFING	Asphalt, composition or prepared.....	70.00	75	6.50*	0.15	(3)	
	Built up— $\frac{1}{2}$ -in. thick.....	—	—	3.53†	0.28	—	
	Built up, bitumen and felt, gravel or slag surfaced.....	—	—	1.33†	0.75	(2)	
	Plaster board, gypsum fiber concrete and 3-ply roof covering.....	52.40	76	0.58†	1.72	(4)	
	Asbestos.....	65.00	75	6.00*	0.17	(3)	
SHINGLES	Asphalt.....	70.00	75	6.50*	0.15	(3)	
	Slate.....	201.00	—	10.37*	0.10	(7)	
	Wood.....	—	—	1.28*	0.78	—	
	—.....	—	—	—	—	—	
	—.....	—	—	—	—	—	
PLASTERING MATERIALS PLASTER	Cement.....	—	—	8.00	0.13	(2)	
	Gypsum, typical.....	—	—	3.30*	0.30	—	
	Thickness $\frac{1}{2}$ in.....	—	73	8.80†	0.11	(4)	
	Total thickness $\frac{1}{2}$ in.....	—	—	4.40†	0.23	—	
	$\frac{3}{8}$ -in. plaster, total thickness $\frac{1}{2}$ in.....	—	70	2.50*	0.40	(4)	
METAL LATH AND PLASTER WOOD LATH AND PLASTER	Cement.....	—	—	8.00	0.13	(2)	
	Gypsum, typical.....	—	—	3.30*	0.30	—	
	Thickness $\frac{1}{2}$ in.....	—	73	8.80†	0.11	(4)	
	Total thickness $\frac{1}{2}$ in.....	—	—	4.40†	0.23	—	
	$\frac{3}{8}$ -in. plaster, total thickness $\frac{1}{2}$ in.....	—	70	2.50*	0.40	(4)	

For notes see Page 107.

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TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	DENSITY (Lb per Cu Ft)	MEAN TEMP. (DEG FAHR)	CONDUCTIVITY (k) OR CONDUCTANCE (C)	RESISTIVITY ($\frac{1}{k}$) OR RESISTANCE (R)	AUTHORITY
BUILDING CONSTRUCTIONS	1-in. fir sheathing and building paper	---	30	0.711*	1.41	(4)
	1-in. fir sheathing, building paper, and yellow pine lap siding	---	20	0.501*	2.00	(4)
	1-in. fir sheathing, building paper and stucco	---	20	0.821*	1.22	(4)
	Pine lap siding and building paper—siding 4 in. wide	---	16	0.851*	1.18	(4)
FLOORING	Yellow pine lap siding	---	---	1.281*	0.78	(4)
	Maple—across grain	40.00	75	1.20	0.83	(3)
	Battleship linoleum (1/4-in.)	---	---	1.361*	0.74	(1)
AIR SPACE AND SURFACE COEFFICIENTS						
AIR SPACES	Over 3/4-in. faced with ordinary building materials	---	40	1.101*	0.91	(4)
SURFACES, ORDINARY	Still air (f_i)	---	---	1.651*	0.61	(4)
	15 mph—(f_o)	---	---	6.001*	0.17	(4)
SURFACE, BRIGHT ALUMINUM	Still air (f_i)	---	60	1.181*	0.85	(4)
AIR SPACES FACED WITH BRIGHT ALUMINUM FOIL	Air space, faced one side with bright aluminum foil, over 3/4-in. wide	---	50	0.461*	2.17	(4)
	Air space, faced one side with bright aluminum foil, 3/8-in. wide	---	50	0.621	1.61	(4)
	Air space, faced both sides with bright aluminum foil, over 3/4-in. wide	---	50	0.411*	2.44	(4)
	Air space, faced both sides with bright aluminum foil, 3/8-in. wide	---	50	0.571	1.75	(4)
	Air space divided in two with single curtain of bright aluminum foil (both sides bright)	---	50	0.231*	4.35	(4)
	Each space over 3/4-in. wide	---	50	0.311	3.23	(4)
	Air space with multiple curtains of bright aluminum foil, bright on both sides, curtains more than 3/4-in. apart, in standard construction:	---	---	---	---	---
	2 curtains, forming 3 spaces	---	50	0.151*	6.78	(4)
	3 curtains, forming 4 spaces	---	50	0.111*	9.22	(4)
	4 curtains, forming 5 spaces	---	50	0.091*	11.66	(4)
WOODS (Across Grain)						
BALSA		20.0	90	0.58	1.72	(1)
		8.8	90	0.38	2.63	(1)
		7.3	90	0.33	3.03	(1)
CALIFORNIA REDWOOD	0% moisture	22.0	75	0.66	1.53	(4)
	8% "	28.0	75	0.70	1.43	(4)
	8% "	22.0	75	0.70	1.43	(4)
	16% "	28.0	75	0.75	1.33	(4)
	16% "	22.0	75	0.74	1.35	(4)
CYPRESS		28.0	75	0.80	1.25	(4)
DOUGLAS FIR	0% moisture	28.7	86	0.67	1.49	(1)
	0% "	34.0	75	0.61	1.64	(4)
	8% "	26.0	75	0.66	1.52	(4)
	8% "	34.0	75	0.75	1.33	(4)
	16% "	26.0	75	0.76	1.32	(4)
	16% "	34.0	75	0.82	1.22	(4)
EASTERN HEMLOCK	0% moisture	22.0	75	0.60	1.67	(4)
	0% "	30.0	75	0.76	1.32	(4)
	8% "	22.0	75	0.63	1.59	(4)
	8% "	30.0	75	0.81	1.23	(4)
	16% "	22.0	75	0.67	1.49	(4)
	16% "	30.0	75	0.85	1.18	(4)

For notes see Page 107.

CHAPTER 5—HEAT TRANSMISSION COEFFICIENTS AND TABLES

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATORS—Continued

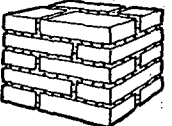
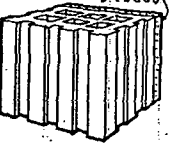
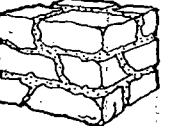
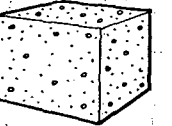
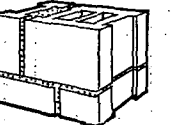
The coefficients are expressed in Btu per hour per square foot per degree Fahrenheit per 1 in. thickness, unless otherwise indicated.

Material	Description	DENSITY (Lb per Cu Ft)	MEAN TEMP. (DEG FAHR)	CONDUCTIVITY (k) OR CONDUCTANCE (C)	RESISTIVITY ($\frac{1}{k}$) OR RESISTANCE (R)	AUTHORITY
WOODS—Continued	0% moisture	40.0	75	1.01	0.99	(4)
	0% "	46.0	75	1.05	0.95	(4)
	8% "	40.0	75	1.08	0.93	(4)
	8% "	46.0	75	1.13	0.89	(4)
HARD MAPLE	0% moisture	40.0	75	1.15	0.87	(4)
	0% "	46.0	75	1.21	0.83	(4)
	8% "	30.0	75	0.76	1.32	(4)
	8% "	40.0	75	0.86	1.16	(4)
LONGLEAF YELLOW PINE	0% moisture	30.0	75	0.83	1.21	(4)
	0% "	40.0	75	0.95	1.05	(4)
	8% "	30.0	75	0.89	1.12	(4)
	8% "	40.0	75	1.03	0.97	(4)
MAHOGANY	0% moisture	34.3	86	0.90	1.11	(1)
	0% "	44.3	86	1.10	0.91	(1)
	8% "	---	---	1.15*	0.87	(4)
	8% "	---	---	1.15*	0.87	(4)
MAPLE OR OAK	0% moisture	22.0	75	0.62	1.61	(4)
	0% "	32.0	75	0.74	1.35	(4)
	8% "	22.0	75	0.68	1.47	(4)
	8% "	32.0	75	0.83	1.21	(4)
NORWAY PINE	0% moisture	22.0	75	0.74	1.35	(4)
	0% "	32.0	75	0.91	1.10	(4)
	8% "	22.0	75	0.67	1.49	(4)
	8% "	32.0	75	0.79	1.27	(4)
RED CYPRESS	0% moisture	32.0	75	0.71	1.41	(4)
	0% "	22.0	75	0.84	1.19	(4)
	8% "	32.0	75	0.84	1.19	(4)
	8% "	22.0	75	0.74	1.35	(4)
RED OAK	0% moisture	32.0	75	0.90	1.11	(4)
	0% "	38.0	75	0.98	1.02	(4)
	8% "	48.0	75	1.18	0.85	(4)
	8% "	38.0	75	1.03	0.97	(4)
SHORTLEAF YELLOW PINE	0% moisture	48.0	75	1.24	0.81	(4)
	0% "	38.0	75	1.07	0.94	(4)
	8% "	48.0	75	1.29	0.78	(4)
	8% "	26.0	75	0.74	1.35	(4)
SOFT ELM	0% moisture	36.0	75	0.91	1.10	(4)
	0% "	26.0	75	0.79	1.27	(4)
	8% "	36.0	75	0.97	1.03	(4)
	8% "	26.0	75	0.84	1.19	(4)
SOFT MAPLE	0% moisture	36.0	75	1.04	0.96	(4)
	0% "	28.0	75	0.73	1.37	(4)
	8% "	34.0	75	0.88	1.13	(4)
	8% "	28.0	75	0.77	1.30	(4)
SUGAR PINE	0% moisture	34.0	75	0.93	1.08	(4)
	0% "	28.0	75	0.81	1.24	(4)
	8% "	34.0	75	0.97	1.03	(4)
	8% "	36.0	75	0.89	1.12	(4)
WEST COAST HEMLOCK	0% moisture	42.0	75	0.95	1.05	(4)
	0% "	36.0	75	0.96	1.04	(4)
	8% "	42.0	75	1.02	0.98	(4)
	8% "	36.0	75	1.01	0.99	(4)
VIRGINIA PINE	0% moisture	42.0	75	1.09	0.92	(4)
	0% "	22.0	75	0.54	1.85	(4)
	8% "	28.0	75	0.64	1.56	(4)
	8% "	22.0	75	0.59	1.70	(4)
WHITE PINE	0% moisture	28.0	75	0.71	1.41	(4)
	0% "	22.0	75	0.65	1.54	(4)
	8% "	28.0	75	0.78	1.28	(4)
	8% "	34.3	86	0.96	1.04	(1)
YELLOW PINE OR FIR	0% moisture	22.0	75	0.68	1.47	(4)
	0% "	30.0	75	0.79	1.27	(4)
	8% "	22.0	75	0.73	1.37	(4)
	8% "	30.0	75	0.85	1.18	(4)
YELLOW PINE OR FIR	0% moisture	22.0	75	0.78	1.28	(4)
	0% "	30.0	75	0.91	1.10	(4)
	8% "	31.2	86	0.78	1.28	(1)
	8% "	---	---	1.00	1.00	(3)
YELLOW PINE OR FIR	0% moisture	---	---	0.80*	1.25	(4)
	0% "	---	---	0.80*	1.25	(4)

For notes see Page 107.

TABLE 3. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS^a

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL	THICKNESS OF MASONRY (INCHES)	WALL No.
	Solid Brick Based on 4-in. face brick and the remainder common brick.	8 12 16	1 2 3
	Hollow Tile Stucco Exterior Finish. The 8-in. and 10-in. tile figures are based on two cells in the direction of flow of heat. The 12-in. tile is based on three cells in the direction of flow of heat. The 16-in. tile consists of one 10-in. tile and one 6-in. tile each having two cells in the direction of heat flow.	8 10 12 16	4 5 6 7
	Limestone or Sandstone	8 12 16 24	8 9 10 11
	Concrete These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.	6 10 16 20	12 13 14 15
	Hollow Cinder Blocks Based on one air cell in direction of heat flow.	8 12	16 17
	Hollow Concrete Blocks Based on one air cell in direction of heat flow.	8 12	18 19

^aComputed from factors marked by * in Table 2.

^bBased on the actual thickness of 2-in. furring strips.

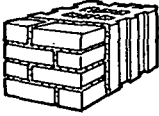
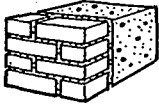
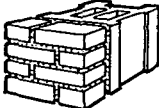
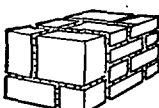
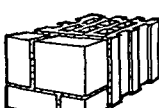
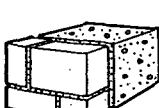
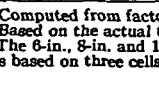
INTERIOR FINISH

UNINSULATED WALLS						INSULATED WALLS					
Plain walls—no interior finish	Plaster (½ in.) on walls	Plaster on wood lath—furred	Plaster (½ in.) on metal lath—furred	Plaster (½ in.) on plaster board (½ in.)—furred	Decorated building board—furred without plaster—furred (½ in.)	Plaster (½ in.) on rigid insulation (½ in.)—furred	Plaster (½ in.) on rigid insulation (1 in.)—furred	Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.)	Plaster (½ in.) on metal lath attached to furring strips—furred space (over ½ in. wide) faced one side with bright aluminum foil	Plaster on metal lath attached to furring strips (2 in.)—cork wool fill (1½ in.) ^b	Plaster (½ in.) on metal lath attached to furring strips (2 in.)—flexible insulation (½ in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.50 0.36 0.28	0.46 0.34 0.27	0.30 0.24 0.20	0.32 0.25 0.21	0.30 0.24 0.20	0.23 0.19 0.17	0.22 0.19 0.16	0.16 0.14 0.13	0.14 0.12 0.11	0.23 0.19 0.17	0.12 0.11 0.10	0.20 0.17 0.15
0.40 0.39 0.30 0.25	0.37 0.37 0.29 0.24	0.26 0.26 0.22 0.19	0.27 0.27 0.22 0.19	0.26 0.26 0.22 0.19	0.20 0.20 0.17 0.15	0.20 0.19 0.17 0.15	0.15 0.15 0.13 0.12	0.13 0.13 0.12 0.11	0.20 0.20 0.17 0.15	0.11 0.11 0.10 0.097	0.18 0.18 0.16 0.14
0.71 0.58 0.49 0.37	0.64 0.53 0.45 0.35	0.37 0.33 0.30 0.25	0.39 0.34 0.31 0.26	0.37 0.33 0.30 0.25	0.26 0.24 0.22 0.20	0.25 0.23 0.22 0.19	0.18 0.17 0.16 0.15	0.15 0.14 0.14 0.13	0.26 0.24 0.22 0.20	0.13 0.13 0.12 0.11	0.23 0.21 0.20 0.18
0.79 0.62 0.48 0.41	0.70 0.57 0.44 0.39	0.39 0.34 0.29 0.27	0.42 0.37 0.31 0.28	0.39 0.34 0.29 0.27	0.27 0.25 0.22 0.21	0.26 0.24 0.21 0.20	0.19 0.18 0.16 0.15	0.16 0.15 0.14 0.13	0.27 0.25 0.22 0.21	0.13 0.13 0.12 0.12	0.23 0.22 0.20 0.18
0.42 0.37	0.39 0.35	0.27 0.25	0.28 0.26	0.27 0.25	0.21 0.19	0.20 0.19	0.16 0.15	0.13 0.13	0.21 0.19	0.12 0.11	0.19 0.17
0.56 0.49	0.52 0.46	0.32 0.30	0.34 0.32	0.32 0.30	0.24 0.23	0.23 0.22	0.17 0.16	0.14 0.14	0.24 0.23	0.12 0.12	0.21 0.20

^aA waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 4. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS WITH VARIOUS TYPES OF VENEERS^a

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF WALL		WALL No.
	FACING	BACKING	
	4 in. Brick Veneer ^d	6 in. Hollow Tile ^a 8 in. 10 in. 12 in.	20 21 22 23
	4 in. Brick Veneer ^d	6 in. Concrete 10 in. 16 in.	24 25 26
	4 in. Brick Veneer ^d	8 in. Cinder Blocks ^a 12 in.	27 28
	4 in. Brick Veneer ^d	8 in. Concrete Blocks ^a 12 in.	29 30
	4 in. Cut-Stone Veneer ^d	8 in. Common Brick 12 in. 16 in.	31 32 33
	4 in. Cut-Stone Veneer ^d	6 in. Hollow Tile ^a 8 in. 10 in. 12 in.	34 35 36 37
	4 in. Cut-Stone Veneer ^d	6 in. Concrete 10 in. 16 in.	38 39 40

^aComputed from factors marked by * in Table 2.

^bBased on the actual thickness of 2-in. furring strips.

^cThe 6-in., 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.

INTERIOR FINISH

UNINSULATED WALLS					INSULATED WALLS						
Plain walls—no interior finish	Plaster (1/2 in.) on walls	Plaster on wood lath—furred	Plaster (1/2 in.) on metal lath—furred	Plaster (1/2 in.) on plaster board (1/2 in.)—furred	No plaster—decorated rigid or building board interior finish (1/2 in.)—furred	Plaster (1/2 in.) on rigid insulation (1/2 in.)—furred	Plaster (1/2 in.) on rigid insulation (1 in.)—furred	Plaster on corkboard (1 1/2 in.) set in cement mortar (1/2 in.)	Plaster on metal lath (1/2 in.) attached to furring strips (2 in.)—faced one side with bright aluminum foil	Plaster (1/2 in.) on metal lath attached to furring strips (2 in.)—rock wool fill (1 1/2 in.)	Plaster (1/2 in.) on metal lath attached to furring strips (2 in.)—flexible insulation (1/2 in.) between furring strips (one air space)
A	B	C	D	E	F	G	H	I	J	K	L
0.36 0.34 0.34 0.27	0.34 0.33 0.32 0.26	0.24 0.24 0.23 0.20	0.25 0.25 0.24 0.21	0.24 0.24 0.23 0.20	0.19 0.19 0.19 0.16	0.19 0.18 0.18 0.16	0.16 0.14 0.14 0.13	0.13 0.12 0.12 0.11	0.19 0.19 0.19 0.16	0.11 0.11 0.11 0.10	0.17 0.17 0.17 0.15
0.57 0.48 0.39	0.53 0.45 0.37	0.33 0.30 0.26	0.35 0.31 0.27	0.33 0.30 0.26	0.24 0.22 0.20	0.23 0.22 0.19	0.17 0.16 0.15	0.14 0.14 0.13	0.24 0.22 0.20	0.13 0.12 0.11	0.21 0.20 0.18
0.35 0.31	0.33 0.30	0.24 0.22	0.25 0.23	0.24 0.22	0.19 0.18	0.18 0.17	0.14 0.14	0.12 0.12	0.19 0.18	0.11 0.11	0.17 0.16
0.44 0.40	0.42 0.38	0.28 0.26	0.30 0.28	0.28 0.26	0.21 0.20	0.21 0.20	0.16 0.15	0.13 0.13	0.21 0.20	0.12 0.11	0.19 0.18
0.37 0.28 0.23	0.35 0.27 0.22	0.25 0.21 0.18	0.26 0.21 0.18	0.25 0.21 0.18	0.19 0.17 0.15	0.19 0.16 0.14	0.15 0.13 0.12	0.13 0.12 0.11	0.19 0.17 0.15	0.11 0.10 0.095	0.17 0.15 0.14
0.37 0.36 0.35 0.28	0.35 0.34 0.33 0.26	0.25 0.24 0.24 0.20	0.26 0.25 0.25 0.21	0.25 0.24 0.24 0.20	0.20 0.19 0.19 0.17	0.19 0.19 0.18 0.16	0.15 0.15 0.14 0.13	0.13 0.13 0.12 0.11	0.20 0.19 0.19 0.17	0.11 0.11 0.11 0.10	0.18 0.17 0.17 0.15
0.61 0.51 0.41	0.56 0.47 0.38	0.34 0.31 0.26	0.36 0.32 0.28	0.34 0.31 0.26	0.25 0.23 0.20	0.24 0.22 0.20	0.18 0.17 0.15	0.15 0.14 0.13	0.25 0.23 0.21	0.13 0.12 0.11	0.22 0.20 0.18

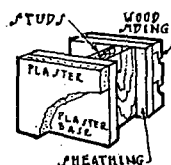
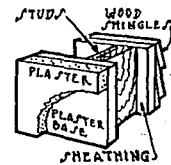
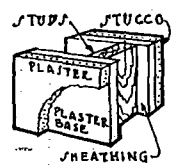
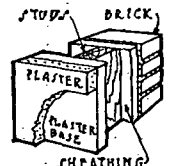
^aCalculations include cement mortar (1/2 in.) between veneer or facing and backing.

^bBased on one air cell in direction of heat flow.

^cA waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 5. COEFFICIENTS OF TRANSMISSION (U) OF VARIOUS TYPES OF FRAME CONSTRUCTION*

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

TYPICAL CONSTRUCTION	EXTERIOR FINISH	TYPE OF SHEATHING	Wall No.
	Wood Siding or Clapboard	1 in. Wood ^d	41
		$\frac{1}{2}$ in. Rigid Insulation	42
		$\frac{1}{2}$ in. Plaster Board	43
	Wood Shingles	1 in. Wood ^d	44
		$\frac{1}{2}$ in. Rigid Insulation ^e	45
		$\frac{1}{2}$ in. Plaster Board ^e	46
	Stucco	1 in. Wood ^d	47
		$\frac{1}{2}$ in. Rigid Insulation	48
		$\frac{1}{2}$ in. Plaster Board	49
	Brick/Veneer	1 in. Wood ^d	50
		$\frac{1}{2}$ in. Rigid Insulation	51
		$\frac{1}{2}$ in. Plaster Board	52

*Computed from factors marked by * in Table 2.

^bThese coefficients may also be used with sufficient accuracy for plaster on wood lath or plaster on plaster board.

^cBased on the actual width of 2 by 4 studding, namely, $3\frac{3}{4}$ in.

INTERIOR FINISH

NO INSULATION BETWEEN STUDDING							INSULATION BETWEEN STUDDING		
Plaster on wood lath on studding	Plaster ($\frac{1}{2}$ in.) on metal lath on studding	Plaster ($\frac{1}{2}$ in.) on plaster board ($\frac{1}{2}$ in.) on studding	Plaster ($\frac{1}{2}$ in.) on rigid insulation ($\frac{1}{2}$ in.) on studding	Plaster ($\frac{1}{2}$ in.) on rigid insulation (1 in.) on studding	Plaster ($\frac{1}{2}$ in.) on corkboard ($1\frac{1}{2}$ in.) on studding	No plaster—decorated rigid or building board interior finish ($\frac{1}{2}$ in.)	Plaster ($\frac{1}{2}$ in.) on metal lath—stud traces faced one side with bright aluminum foil	Plaster ($\frac{1}{2}$ in.) on metal lath ^b on studding—rock wool fill ($3\frac{3}{4}$ in.) ^c between studding ^d	Plaster ($\frac{1}{2}$ in.) on metal lath ^b on studding—flexible insulation ($\frac{1}{2}$ in.) between studding and in contact with sheathing
A	B	C	D	E	F	G	H	I	J
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.19	0.061	0.17
0.23	0.24	0.23	0.18	0.14	0.11	0.18	0.18	0.060	0.17
0.31	0.33	0.31	0.22	0.17	0.13	0.23	0.23	0.064	0.20
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.19	0.061	0.17
0.19	0.20	0.19	0.15	0.12	0.10	0.16	0.16	0.057	0.14
0.24	0.25	0.24	0.19	0.15	0.11	0.19	0.19	0.061	0.17
0.30	0.31	0.30	0.22	0.16	0.12	0.22	0.22	0.064	0.20
0.27	0.29	0.27	0.20	0.16	0.12	0.21	0.21	0.062	0.19
0.40	0.43	0.40	0.26	0.19	0.14	0.28	0.28	0.067	0.24
0.27	0.28	0.27	0.20	0.15	0.12	0.21	0.21	0.062	0.18
0.25	0.26	0.25	0.19	0.15	0.11	0.19	0.20	0.061	0.18
0.35	0.37	0.35	0.24	0.18	0.13	0.25	0.25	0.066	0.22

^dYellow pine or fir—actual thickness about $\frac{3}{4}$ in.

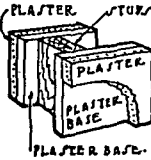
^eFurring strips between wood shingles and sheathing.

^fSmall air space and mortar between building paper and brick veneer neglected.

^gA waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 6. COEFFICIENTS OF TRANSMISSION (*U*) OF FRAME INTERIOR WALLS AND PARTITIONS^a

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION 	WALL No.	SINGLE PARTITION (FINISH ON ONE SIDE OF STUDDING)	DOUBLE PARTITION (FINISHED ON BOTH SIDES OF STUDDING)				
			Air Space Between Studding	Flaked Gypsum Fill ^b Between Studding	Rock Wool Fill ^b Between Studding	1/2-in. Flexible Insulation Between Studding (One Air Space)	Stud Space Faced One Side with Bright Aluminum Foil
TYPE OF WALL		A	B	C	D	E	F
Wood Lath and Plaster On Studding	53	0.62	0.34	0.11	0.065	0.21	0.24
Metal Lath and Plaster ^c On Studding	54	0.69	0.39	0.11	0.066	0.23	0.26
Plaster Board (3/4 in.) and Plaster ^d On Studding	55	0.61	0.34	0.10	0.065	0.21	0.24
1/2 in. Rigid Insulation and Plaster ^d On Studding	56	0.35	0.18	0.083	0.056	0.14	0.15
1 in. Rigid Insulation and Plaster ^d On Studding	57	0.23	0.12	0.066	0.048	0.097	0.10
1 1/2 in. Corkboard and Plaster ^d On Studding	58	0.16	0.081	0.052	0.040	0.070	0.073
2 in. Corkboard and Plaster ^d On Studding	59	0.12	0.063	0.045	0.035	0.057	0.059

^aComputed from factors marked by * in Table 2.

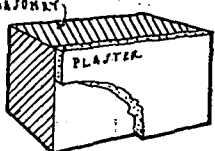
^bThickness assumed 3/4 in.

^cPlaster on metal lath assumed 3/4-in. thick.

^dPlaster assumed 1/2-in. thick.

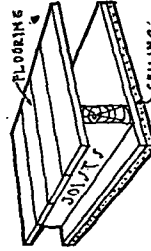
TABLE 7. COEFFICIENTS OF TRANSMISSION (*U*) OF MASONRY PARTITIONS^a

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION 	No.	PLAIN WALLS (NO PLASTER)	WALLS PLASTERED ON ONE SIDE	WALLS PLASTERED ON BOTH SIDES
TYPE OF WALL		A	B	C
4-in. Hollow Clay Tile	60	0.45	0.42	0.40
4-in. Common Brick	61	0.50	0.46	0.43
4-in. Hollow Gypsum Tile	62	0.30	0.28	0.27
2-in. Solid Plaster	63	-----	-----	0.53

^aComputed from factors marked by * in Table 2.

TABLE 8. COEFFICIENTS OF TRANSMISSION (*U*) OF FRAME CONSTRUCTION FLOORS AND CEILINGS^a
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION 	No.	INSULATION BETWEEN JOISTS	TYPE OF FLOORING				
			No Flooring	Yellow Pine Flooring ^b on Joists	Yellow Pine Flooring on Rigid Insulation (1/2 in.) on Joists	Maple or Oak Flooring on Yellow Pine Sub-Flooring ^b on Joists	1/2-in. Battiship Insulation on Yellow Pine Sub-Flooring ^b
TYPE OF CEILING			A	B	C	D	E
No Ceiling	1	None	-----	0.46	0.27	0.34	0.34
Metal Lath and Plaster (3/4 in.)	2	None	0.89	0.30	0.21	0.25	0.25
Wood Lath and Plaster	3	None	0.62	0.28	0.20	0.24	0.24
Plaster Board (3/4 in.) and Plaster (1/2 in.)	4	None	0.61	0.28	0.20	0.24	0.23
Rigid Insulation (1/2 in.) and Plaster (1/2 in.)	5	None	0.35	0.21	0.16	0.18	0.18
Metal Lath and Plaster	6	Flexible ^d Insulation (1/2 in.)	0.24	0.16	0.13	0.15	0.15
Metal Lath and Plaster	7	Rigid Insulation ^d (1/2 in.)	0.26	0.17	0.14	0.15	0.15
Metal Lath and Plaster	8	Bright Aluminum Foil ^e	0.59	0.22	0.16	0.19	0.19
Metal Lath and Plaster	9	Rock Wool Fill (3/4 in.)	0.067	0.063	0.058	0.060	0.060
Corkboard (1 1/2 in.) and Plaster (1/2 in.)	10	None	0.16	0.12	0.10	0.11	0.11
Corkboard (2 in.) and Plaster (1/2 in.)	11	None	0.12	0.10	0.087	0.094	0.094

^aComputed from factors marked by * in Table 2.

^bThickness assumed to be 3/4 in.

^cThickness assumed to be 1/2 in.

^dBased on one air space with no flooring, and two air spaces with flooring. The value of *U* will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1-in. furring strips.

^eAir space faced on one side with bright aluminum foil.

TABLE 9. COEFFICIENTS OF TRANSMISSION (*U*) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS^a
Coefficients are expressed in *Btu per square foot per degree Fahrenheit difference in temperature between the air on the two sides,*
and are based on still air (no wind) conditions on both sides.

TYPICAL CONSTRUCTION	THICKNESS OF CONCRETE (INCHES)	No.	TYPE OF FLOORING							
			No Flooring (Concrete Bare) ^a	Yellow Pine Flooring on Wood Sleepers Embedded in Concrete ^b	Maple or Oak Flooring on Yellow Pine Sub-Flooring ^c on Wood Sleepers Embedded in Concrete	Tile or Terrazzo Flooring on Concrete	½-In. Battlement Linoleum Directly on Concrete			
TYPE OF CEILING						A	B	C	D	E
No Ceiling	4	1	0.65	0.40	0.31	0.61	0.44			
	6	2	0.59	0.37	0.30	0.56	0.41			
	8	3	0.53	0.35	0.28	0.51	0.38			
	10	4	0.49	0.33	0.27	0.47	0.36			
½ in. Plaster Applied Directly to Under Side of Concrete	4	5	0.59	0.38	0.30	0.56	0.41			
	6	6	0.54	0.35	0.28	0.52	0.38			
	8	7	0.50	0.33	0.27	0.47	0.36			
	10	8	0.45	0.32	0.26	0.44	0.34			
Suspended or Furred Metal Lath and Plaster (¾ in.) Ceiling	4	9	0.37	0.28	0.23	0.36	0.29			
	6	10	0.35	0.26	0.22	0.34	0.28			
	8	11	0.33	0.25	0.21	0.32	0.27			
	10	12	0.32	0.24	0.21	0.31	0.25			
Suspended or Furred Ceiling of Plaster Board (¾ in.) and Plaster (½ in.)	4	13	0.35	0.26	0.22	0.34	0.28			
	6	14	0.33	0.25	0.21	0.32	0.26			
	8	15	0.31	0.24	0.21	0.30	0.25			
	10	16	0.30	0.23	0.20	0.29	0.24			
Suspended or Furred Ceiling of Rigid Insulation (½ in.) and Plaster (½ in.)	4	17	0.24	0.20	0.17	0.24	0.21			
	6	18	0.23	0.19	0.17	0.23	0.20			
	8	19	0.22	0.18	0.16	0.22	0.19			
	10	20	0.22	0.18	0.16	0.21	0.19			
Plaster (½ in.) on Corkboard (1½ in.) Set in Cement Mortar (½ in.) on Concrete	4	21	0.15	0.13	0.12	0.14	0.14			
	6	22	0.14	0.13	0.12	0.14	0.13			
	8	23	0.14	0.12	0.11	0.14	0.13			
	10	24	0.14	0.12	0.11	0.14	0.13			

^aComputed from factors marked by * in Table 2.

^bThe figures in Column A may be used with sufficient accuracy for concrete floors covered with carpet.

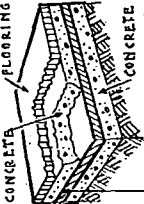
^cThe figures in Column B may be used with sufficient accuracy for concrete floors covered with carpet.

^dThe figures in Column C may be used with sufficient accuracy for maple or oak flooring^e applied directly over the concrete on wood sleepers.

^eThickness of maple or oak flooring assumed to be ¾ in.

^fThickness of tile or terrazzo assumed 1 in.

TABLE 10. COEFFICIENTS OF TRANSMISSION (*U*) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORINGS.^a
Coefficients are expressed in *Btu per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor,*
and are based on still air (no wind) conditions.

TYPICAL CONSTRUCTION	THICKNESS OF CONCRETE (INCHES)	No.	TYPE OF FINISH FLOORING				
			No Flooring (Concrete Bare)	Yellow Pine Floorings ^a on Wood Sleepers Resting on Concrete	Maple or Oak Floorings ^a on Yellow Pine Sub-Flooring on Wood Sleepers Resting on Concrete	Tile or Terrazo ^d on Concrete	¼-in. Battiship Linoleum Directly on Concrete
 CONCRETE FLOORING CONCRETE INSULATION BETWEEN TWO MEMBRANE WATERPROOFING COURSES	4	1	1.07	0.35	0.28	0.98	0.60
	6	2	0.90	0.33	0.27	0.84	0.54
	8	3	0.79	0.32	0.26	0.74	0.50
	10	4	0.70	0.30	0.25	0.66	0.46
None	4	5	0.66	0.29	0.24	0.63	0.44
	8	6	0.54	0.27	0.23	0.52	0.39
None ^e	4	7	0.22	0.16	0.14	0.22	0.19
	8	8	0.21	0.15	0.13	0.20	0.18
	4	9	0.12	0.099	0.093	0.12	0.11
	8	10	0.12	0.096	0.090	0.12	0.11
1 in. Rigid Insulation ^e	4	7	0.22	0.16	0.14	0.22	0.19
	8	8	0.21	0.15	0.13	0.20	0.18
1 in. Rigid Insulation ^e	4	9	0.12	0.099	0.093	0.12	0.11
	8	10	0.12	0.096	0.090	0.12	0.11
2 in. Corkboard ^e	4	9	0.12	0.099	0.093	0.12	0.11
	8	10	0.12	0.096	0.090	0.12	0.11
2 in. Corkboard ^e	4	9	0.12	0.099	0.093	0.12	0.11
	8	10	0.12	0.096	0.090	0.12	0.11

^aComputed from factors marked by * in Table 2.

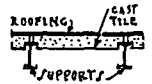
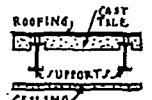
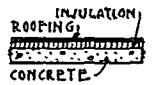
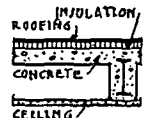
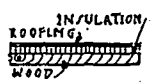
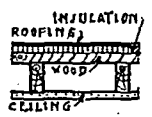
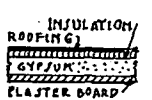
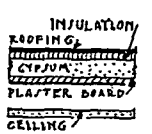
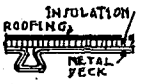
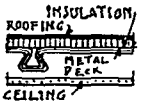
^bAssumed ¾ in. thick.

^cAssumed ¾ in. thick.

^dAssumed 1 in. thick.

^eThe figures for Nos. 5 to 10, inclusive, include 3-in. cinder concrete placed directly on the ground. The insulation is applied between the cinder concrete and the stone concrete. Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations.

TABLE 11. COEFFICIENTS OF TRANSMISSION (*U*) OF VARIOUS TYPES OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING*

TYPICAL CONSTRUCTION		TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	No.
WITHOUT CEILINGS	WITH METAL LATH AND PLASTER CEILINGS ^d			
		Precast Cement Tile	1½	1
		Concrete Concrete Concrete	2 4 6	2 3 4
		Wood Wood Wood Wood	1½ 1½ ^b 2½ 4 ^b	5 6 7 8
		Gypsum Fiber Concrete ^c (2 in.) on Plaster Board (¾ in.)	2¾	9
		Gypsum Fiber Concrete ^c (3 in.) on Plaster Board (¾ in.)	3¾	10
		Gypsum Fiber Concrete ^c (2 in.) on Rigid Insulation Board (¼ in.)	2¼	11
		Gypsum Fiber Concrete ^c (2 in.) on Rigid Insulation Board (1 in.)	3	12
		Flat Metal Roofs Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph.		13

*Computed from factors marked by * in Table 2.

^bNominal thicknesses specified—actual thicknesses used in calculations.

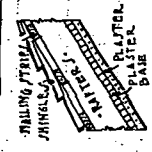
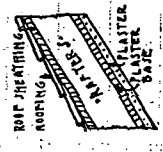
^cGypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber.

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

WITHOUT CEILING—UNDER SIDE OF ROOF EXPOSED								WITH METAL LATH AND PLASTER CEILINGS ^d							
No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)	No Insulation	Rigid Insulation (½ in.)	Rigid Insulation (1 in.)	Rigid Insulation (1½ in.)	Rigid Insulation (2 in.)	Corkboard (1 in.)	Corkboard (1½ in.)	Corkboard (2 in.)
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
0.84	0.37	0.24	0.18	0.14	0.22	0.16	0.13	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11
0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	0.42	0.26	0.19	0.15	0.12	0.18*	0.14	0.11
0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11
0.64	0.33	0.22	0.16	0.13	0.21	0.15	0.12	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11
0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095
0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092
0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082
0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097
0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091
0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.10	0.20	0.16	0.13	0.11	0.09	0.12	0.10	0.087
0.19	0.15	0.12	0.10	0.09	0.12	0.10	0.08	0.16	0.13	0.11	0.09	0.08	0.10	0.09	0.077
0.95	0.39	0.25	0.18	0.14	0.23	0.17	0.13	0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11

^dThese coefficients may be used with sufficient accuracy for wood lath and plaster, or plaster board 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.

TABLE 12. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS^a
Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

TYPICAL CONSTRUCTION	TYPE OF ROOFING AND ROOF SHEATHING	INSULATION BETWEEN ROOF RAFTERS	No.	TYPE OF CEILING (Applied Directly to Roof Rafters)									
				A	B	C	D	E	F	G	H	I	
	Wood Shingles on Wood Strips ^b	None	1	0.46	0.30	0.29	0.29	0.22	0.21	0.18	0.12	0.10	Corkboard (2 in.) and Plaster (1/2 in.)
		1/2 in. Flexible ^c	2		0.17	0.16	0.16	0.14	0.13	0.11	0.091	0.079	Corkboard (1 1/2 in.) and Plaster (1/2 in.)
		1 in. Flexible ^c	3		0.13	0.12	0.12	0.11	0.11	0.092	0.078	0.069	Rigid Insulation (1 in.) and Plaster (1/2 in.)
		Bright Aluminum Foil ^d	4		0.22	0.21	0.21	0.17	0.17	0.13	0.10	0.085	Rigid Insulation (1/2 in.)
		3/8 in. Rock Wool ^e	5		0.093	0.092	0.092	0.082	0.068	0.053	0.048	0.044	
	Asphalt Shingles, Rigid Asbestos Shingles, Composition Roofing, or Slate or Tile Roofing ^f on Wood Sheathing ^g	None	6	0.56	0.34	0.32	0.32	0.24	0.23	0.17	0.13	0.11	
		1/2 in. Flexible ^c	7		0.18	0.17	0.17	0.14	0.14	0.12	0.094	0.089	
		1 in. Flexible ^c	8		0.13	0.13	0.13	0.11	0.11	0.095	0.080	0.071	
		Bright Aluminum Foil ^d	9		0.24	0.23	0.23	0.18	0.18	0.14	0.11	0.093	
		3/8 in. Rock Wool ^e	10		0.085	0.084	0.084	0.080	0.069	0.054	0.049	0.045	

^aComputed from factors marked by * in Table 2. Nos. 6 to 10, inclusive, based on 1/2-in. thick slate.

^bBased on 1 in. by 4 in. strips spaced 2 in.

^cFigured based on two air spaces.

^dRoofing felt between roof sheathing and slate or tile neglected in calculations.

^eAssumed 3/8 in. thick based on the actual width of 2 in. by 4 in. rafters.

^fSheathing assumed 1/2 in. thick.

^gAir space faced on one side with bright aluminum foil.

CHAPTER 5—HEAT TRANSMISSION COEFFICIENTS AND TABLES

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS AND SKYLIGHTS
Coefficients are based on a wind velocity of 15 mph, and are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air inside and outside of the door, window or skylight

degree Fahrenheit - 32

A. Windows and Skylights		
	U	
Single.....	1.13 ^{a, c}	
Double.....	0.45 ^a	
Triple.....	0.281 ^a	
B. Solid Wood Doors ^{b, c}		
NOMINAL THICKNESS INCHES	ACTUAL THICKNESS INCHES	U
1	2 5/8	0.69
1 1/4	1 1/8	0.59
1 1/2	1 3/8	0.52
1 3/4	1 5/8	0.51
2	1 7/8	0.46
2 1/2	2 1/8	0.38
3	2 5/8	0.33

^aSee *Heating, Ventilating and Air Conditioning*, by Harding and Willard, revised edition, 1932.

^bComputed using $C = 1.15$ for wood; $f_1 = 1.65$ and $f_2 = 6.0$.

^cIt 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 per square foot per degree difference between inside and outside air temperatures.

While most building materials have surfaces which show similar characteristics as far as the transmission of heat is concerned, it is a well-known fact that certain surfaces such as aluminum bronze, gold bronze, aluminum foil, or in fact any metallic, highly polished surface presents a greater resistance to heat transmission than the surface of the average building material.

The greater heat resistance of such metallic surfaces is due primarily to their higher reflectivity and consequent lower emissivity of radiant heat. The use of multiple layers of metallic surfaces, combined with air spaces of low resistance, provides a definite insulating effect. Factors² for air spaces bounded by aluminum foil are given in Table 2.

Coefficients of transmission of various types of wall, ceiling, floor and roof construction with aluminum insulation can be readily calculated. The present installation practice indicates that air spaces of 1/2 in. to 1 1/2 in. are preferred but manufacturers' recommendations should be closely followed in the application of aluminum foil insulation.

The majority of the conductivities and conductances of the building materials and insulations given in Table 2 were determined by the hot-plate method of testing³. Attention is called to the fact that conductivities per inch of thickness of materials or insulations do not afford a true basis for comparison, although they are frequently used for that purpose.

¹Insulating Value of Bright Metallic Surfaces, by F. B. Rowley (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1934, p. 263).

²Standard Test Code for Heat Transmission through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928). See also Chapter 43.

Correct comparisons should take into consideration many different factors, including conductivities or conductances, thicknesses installed and manner of installation, while the selection of an insulation should also give consideration to structural qualities, as well as to material and application costs. Fire, vermin, and rot resistance are other important factors to be considered when comparing materials. At present there is no universally recognized method of rating insulations. Conductivities and conductances of building materials and insulations are useful to the heating engineer in determining over-all coefficients of heat transmission of walls, floors, roofs and ceilings.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 3 to 13, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and $\frac{1}{2}$ in. of plaster is 0.46, and the number assigned to a wall of this construction is 1-B, Table 3.

Example 1. Calculate the coefficient of transmission (U) of an 8-in. brick wall with $\frac{1}{2}$ in. of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph. It is assumed that the outside course is of face brick having a conductivity of 9.20, and that the inside course is of common brick having a conductivity of 5.0, the thicknesses each being 4 in. The conductivity of the plaster is assumed to be 3.3, and the inside and outside surface coefficients are assumed to average 1.65 and 6.00, respectively, for still air and a 15 mph wind velocity.

Solution. k (face brick) = 9.20; x = 4.0 in.; k (common brick) = 5.0; x = 4.0 in.; k (plaster) = 3.3; x = $\frac{1}{2}$ in.; f_i = 1.65; f_o = 6.0. Therefore,

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}}$$

$$= \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

= 0.46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

The coefficients in the tables were determined by calculations similar to those shown in Example 1, using Fundamental Formulae 2, 3, 4 and 5 and the values of k (or C), f_i , f_o and a indicated in Table 2 by asterisks. In computing heat transmission coefficients of floors laid directly on the ground (Table 10), only one surface coefficient (f_i) is used. For example, the value of U for a 1-in. yellow pine floor (actual thickness, 25/32 in.) placed directly on 6-in. concrete on the ground, is determined as follows:

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.781}{0.80} + \frac{6.0}{12.0}} = 0.48 \text{ Btu per hour per square foot per degree difference in temperature between the ground and the air immediately above the floor.}$$

The thicknesses upon which the coefficients in Tables 3 to 13, inclusive, are based are as follows:

Brick veneer.....	4 in.
Plaster and metal lath.....	$\frac{3}{4}$ in.

Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard).....	$\frac{1}{2}$ in.
Slate (roofing).....	$\frac{1}{2}$ in.
Stucco on wire mesh reinforcing.....	1 in.
Tar and gravel or slag-surfaced built-up roofing.....	$\frac{3}{8}$ in.
1-in. lumber (S-2-S).....	$2\frac{25}{32}$ in.
$1\frac{1}{2}$ -in. lumber (S-2-S).....	$1\frac{15}{16}$ in.
2-in. lumber (S-2-S).....	$1\frac{5}{8}$ in.
$2\frac{1}{2}$ -in. lumber (S-2-S).....	$2\frac{3}{8}$ in.
3-in. lumber (S-2-S).....	$2\frac{5}{8}$ in.
4-in. lumber (S-2-S).....	$3\frac{3}{8}$ in.
Finish flooring (maple or oak).....	$1\frac{3}{16}$ in.

Solid brick walls are based on 4-in. face brick and the remainder common brick. Stucco is assumed to be 1-in. thick on masonry walls. Where metal lath and plaster are specified, the metal lath is neglected.

Rigid insulation refers to the so-called board form which may be used structurally, such as for sheathing. Flexible insulation refers to the blankets, quilts or semi-rigid types of insulation.

Actual thicknesses of lumber are used in the computations rather than nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as it is probable that the values of U for these two types of roofs will compare favorably.

The coefficients of transmission of the pitched roofs in Table 12 apply where the roof is over a heated attic or top floor so the heat passes directly through the roof structure including whatever finish is applied to the underside of the roof rafters.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined coefficient of transmission of a top-floor ceiling, unheated attic space, and pitched roof, per square foot of roof area, is as follows:

$$U = \frac{U_r \times U_{ce}}{n \times U_r + U_{ce}} \quad (6)$$

where

U_r = coefficient of transmission of the roof.
 U_{ce} = coefficient of transmission of the ceiling.
 n = the ratio of the area of the roof to the area of the ceiling.

In using this formula, a correction factor must be applied. As the amount of heat transferred through an air space is proportional to the difference of the fourth powers of the absolute temperatures of the surfaces enclosing the air space, a greater amount of heat is absorbed or emitted by radiation by the surfaces enclosing an unheated attic than by the surfaces of a wall or ceiling in a room under still-air conditions, where the surrounding objects are only slightly higher in temperature than the interior surfaces of the walls and ceiling. For example, the average

coefficient of a surface in still air is 1.65 Btu per hour per square foot per degree Fahrenheit, whereas the average coefficient of an air space in an outside wall is about 1.10 Btu per hour per square foot per degree Fahrenheit difference between the two surfaces, at a mean temperature of 40 F. An air space coefficient of 1.10 is equivalent to a surface coefficient of 2.20 for each of the two surfaces enclosing the air space, where the over-all transmission is computed by using the coefficients of the two surfaces enclosing the air space instead of the coefficient of the air space itself. Hence, in determining the values of U_r and U_{ce} to be used in the formula, the coefficients for the surfaces of the roof and ceiling enclosing the attic should be increased to allow for the additional amount of heat transferred by radiation, and a coefficient of 2.20 may be used with sufficient accuracy for each of these surfaces, although in very precise work a correction should be made to allow for the fact that the area of a pitched roof over an unheated attic is greater than the area of the ceiling, and hence, the amount of heat absorbed by radiation by each square foot of roof surface is less than is given off by radiation by each square foot of ceiling surface.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall surfaces, the combined coefficients may be used for determining the heat loss through the roof construction between the attic and top-floor ceiling, but it should be noted that these coefficients should be multiplied by the roof area and not by the ceiling area. If the unheated attic contains windows, ventilators or vertical wall surfaces, which would tend to reduce temperature in the attic to a temperature approaching or equaling the outside temperature, the roof should be neglected and only the top-floor ceiling construction and the corresponding ceiling area taken into consideration, using the coefficients given in Tables 8 or 9. Where there are no dormers, doors, or windows, and when the transmission coefficients of the roof and the ceiling are approximately the same, the value of the attic temperature may be taken as an average between the inside and the outside temperature.

Basements and Unheated Rooms

The heat loss through floors into basements and into unheated rooms kept closed may be computed by assuming a temperature for these rooms of 32 F.

Additional information on the inside and outside temperatures to be used in heat loss calculations is given in Chapter 7.

REFERENCES

- A.S.H.V.E. research paper entitled 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).
 A.S.H.V.E. research paper entitled Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).
 A.S.H.V.E. research paper entitled 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).
 A.S.H.V.E. research paper entitled Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).
 A.S.H.V.E. research paper entitled Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).
 A.S.H.V.E. research paper entitled Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

- A.S.H.V.E. research paper entitled The Heat Conductivity of Wood at Climatic Temperature Differences, by F. B. Rowley (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1933).
 A.S.H.V.E. research paper entitled Insulating Value of Bright Metallic Surfaces, by F. B. Rowley (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1934).
 Heat Transmission through Building Materials, by F. B. Rowley and A. B. Algren, *University of Minnesota Engineering Experiment Station Bulletin No. 8*.
 Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).
 Importance of Radiation in Heat Transfer through Air Spaces, by E. R. Queer (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).
 Properties of Metal Foil as an Insulating Material, by J. L. Gregg (*Refrigerating Engineering*, May, 1932).
 Thermal Insulation with Aluminum Foil, by R. B. Mason (*Industrial and Engineering Chemistry*, March, 1933).
 Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932.
 Thermal Insulation of Buildings, Technical Paper No. 11 (*American Architect*, May, 1934).
 House Insulation, Its Economics and Application, by Russell E. Backstrom (Report of the National Committee on Wood Utilization, *United States Government Printing Office*, 1931).
 Heat Insulation as Applied to Buildings and Structures, by E. A. Allcut, University of Toronto, 1934.

PROBLEMS IN PRACTICE

1 • What is the coefficient U and how is it applied?

The coefficient U is the heat loss through walls, ceilings, and floors and the value depends upon the construction and material, expressed in Btu per hour per square foot per degree difference in temperature between the inside and outside. To determine the total heat loss, multiply U for each material by the square feet of surface and the temperature difference.

2 • What is the conductivity of face brick?

9.20 (Table 2).

3 • What is the conductance of wood shingles?

1.28 (Table 2).

4 • What is the over-all coefficient of transmission U for a solid brick wall 12-in. thick with plaster on wood lath, furred?

0.24 (Table 3, Wall 2C).

5 • Find the value of U for a 6-in. concrete wall with plaster on metal lath attached to 2-in. furring strips with flanged $\frac{1}{2}$ -in. blanket insulation.

0.23 (Table 3, Wall 12L).

6 • Find the value of U for a wood siding wall with an interior finish of $\frac{3}{4}$ -in. plaster on metal lath; sheathing thickness, $2\frac{5}{32}$ in.

0.26 (Table 5, Wall 41B).

7 • What value of U should be used for a brick veneer wall with $\frac{1}{2}$ -in. rigid insulation sheathing finished on the interior with plaster on $\frac{1}{2}$ -in. rigid insulation?

0.19 (Table 5, Wall 51D).

8 • What value of U should be used in computing the heat loss from an attic through a floor of yellow pine on joists with a ceiling of metal lath and plaster?

0.30 (Table 8, Floor 2B).

9 • What is the over-all heat transfer coefficient for a 6-in. concrete floor with no insulation and with yellow pine flooring on sleepers resting on concrete?

0.33 (Table 10, Floor 2B).

10 • What is the coefficient U for a built-up flat roof of 4-in. concrete with a metal lath and plaster ceiling insulated with 1-in. cork board?

0.17 (Table 11, Roof 3N).

11 • A solid 12-in. common brick wall is finished on the inside with $\frac{1}{2}$ -in. insulation plaster base, and $\frac{1}{2}$ -in. plaster; the plaster base is furred 1 in. from the brick; k for insulating material = 0.34. Calculate the over-all coefficient U .

$$f_i = 1.65; f_o = 6.00; k \text{ for brick} = 5.00; a \text{ for 1-in. air space} = 1.1$$

$$\text{Over-all heat resistance} = R = \frac{1}{1.65} + \frac{0.5}{0.34} + \frac{0.5}{3.3} + \frac{1}{1.1} + \frac{12}{5} + \frac{1}{6} = 5.703$$

$$U = \frac{1}{R} = 0.175$$

12 • A wall is built with two layers of $\frac{1}{2}$ -in. insulating material spaced 1 in. apart; the air space is lined on one side with bright aluminum foil; mean temperature is 40 F; still air on both sides of wall; k for insulating material is 0.34. Calculate the value of U .

$$f_i = 1.65; f_o = 1.65; a = 0.46$$

$$R = \frac{1}{1.65} + \frac{0.5}{0.34} + \frac{1}{0.46} + \frac{0.5}{0.34} + \frac{1}{1.65} = 6.327$$

$$U = \frac{1}{R} = 0.158$$

13 • What is the inside surface temperature of a 6-in. solid concrete wall? Inside air, 70 F; outside air, -20 F with 15 mph wind.

The temperature drop from point to point through a wall is directly proportional to the heat resistance.

$$f_i = 1.65; k \text{ for concrete} = 12; f_o = 6.0$$

$$\text{Over-all resistance } R = \frac{1}{1.65} + \frac{6}{12} + \frac{1}{6.0} = 1.27$$

$$\frac{\text{Temperature drop, inside air to surface}}{\text{Temperature drop, air to air}} = \frac{1.65}{1.27}$$

$$\text{Temperature drop, inside air to surface} = \frac{90}{1.27 \times 1.65} = 43$$

$$70 - 43 = 27 \text{ F, inside surface temperature of wall.}$$

14 • How many inches of insulating material having a conductivity of 0.30 would be required, for the wall of Question 3, to raise the inside surface temperature to 60 F?

Temperature drop, air to inside surface = 10 F; temperature drop, inside surface to outside air = 80 F. Therefore, the heat resistance from inside wall surface to outside air must be eight times that from inside air to inside wall surface, or $8 \times \frac{1}{1.65} = 4.85$. The resistance for added material is, therefore,

$$4.85 - \left(\frac{6}{12} + \frac{1}{6} \right) = 4.19$$

$$4.19 \times 0.30 = 1.25 \text{ in. of insulation.}$$

Chapter 6

AIR LEAKAGE

Nature of Air Infiltration, Air Leakage Through Walls, Window Leakage, Wind Velocity to be Selected, Crack used for Computations, Multi-Story Buildings, Heat Equivalent of Air Entering by Infiltration

AIR leakage losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building, such as cracks around doors and windows, fireplaces and chimneys. This leakage of air must be considered in heating and cooling calculations. (See Chapters 7 and 8.)

THE NATURE OF AIR FILTRATION

The natural movement of air through building construction is due to two causes. One is the pressure exerted by the wind; the other is the difference in density of outside and inside air because of differences in temperature.

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. Tests on mechanically ventilated classrooms of average construction have shown that air infiltration acts quite independently of the planned air supply. Accordingly, the heating or cooling load owing to air infiltration from natural causes should be considered in addition to the ventilating load.

The air exchange owing to temperature difference, inside to outside, is not appreciable in low buildings. In tall, single story buildings with openings near the ground level and near the ceiling, this loss must be considered. Also in multi-storied buildings it is a large item unless the sealing between various floors and rooms is quite perfect. This temperature effect is a *chimney action*, causing air to enter through openings at lower levels and to leave at higher levels.

A complete study of all of the factors involved in air movement through building constructions would be very complex. Some of the complicating factors are: the variations in wind velocity and direction; the exposure of the building with respect to air leakage openings and with respect to adjoining buildings; the variations in outside temperatures as influencing the chimney effect; the relative area and resistance of openings on the windward and leeward sides and on the lower floors and on the upper floors; the influence of a planned air supply and the related outlet vents; and the variation from the average of individual building units. A study of infiltration points to the need for care in the obtaining of good building construction, or unnecessarily large heat losses will result.

AIR LEAKAGE THROUGH WALLS

Table 1 gives data on infiltration through brick and frame walls. 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.

TABLE 1. INFILTRATION THROUGH WALLS

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..... Plastered.....	1.75 0.017	4.20 0.037	7.85 0.066	12.2 0.107	18.6 0.161	22.9 0.236
13 in. Brick Wall..... Plain..... Plastered.....	1.44 0.005	3.92 0.013	7.48 0.025	11.6 0.043	16.3 0.067	21.2 0.097
Frame Wall, with lath and plaster ^b	0.03	0.07	0.13	0.18	0.23	0.26

*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 p. 140.

^bWall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster.

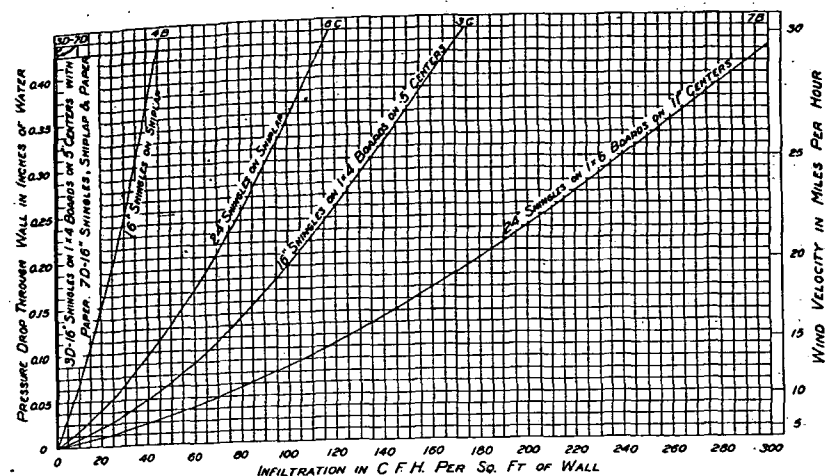


FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

The value of building paper when applied between sheathing and 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.

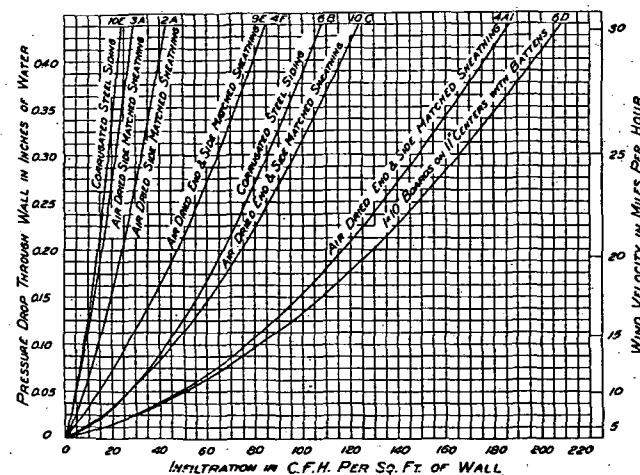


FIG. 2. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

The amount of infiltration that may be expected through simple walls used in farm and other shelter buildings, is shown in Fig. 2. The infiltration indicated in Figs. 1 and 2 is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

WINDOW LEAKAGE

The amount of infiltration for various types of windows is given in Table 2. The fit of double-hung wood windows is determined by crack and clearance as illustrated in Fig. 3. 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. Values of leakage shown in Table 2 for the average double-hung wood window were determined by setting the average measured crack and clearance found in a field survey of a large number of windows on nine windows tested in the laboratory. In addition, the table gives figures for a poorly fitted window. All of the figures for double-hung wood windows are for the *unlocked* condition. Just how a window is closed, or fits when it is closed, has considerable influence on the leakage. The leakage will be high if the sash are short, if the meeting rail members are warped, or if the frame and sash are not fitted squarely to each other. It is possible to have a window with approximately the average crack and clearance that will have a leakage at least double that of the figures shown. Values for the average double-hung wood window in Table 2 are considered to be easily obtainable figures provided the workmanship on the window is good. Should it be known that the windows under consideration are poorly fitted, the larger leakage values should be used. Locking a window generally decreases its leakage, but in some cases may push the meeting rail members apart and increase the leakage. On windows with large clearances, locking will usually reduce the leakage.

Wood casement windows may be assumed to have the same unit leakage as for the average double-hung wood window when properly fitted. Locking, a normal operation in the closing of this type of window, maintains the crack at a low value.

For 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.

Leakage values for storm sash are given in Figs. 4 and 5. 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. When storm sash are applied to poorly fitted windows, a reduction in leakage of 50 per cent may be secured.

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

TABLE 2. INFILTRATION THROUGH WINDOWS
Expressed in Cubic Feet per Foot of Crack per Hour

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	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked ^b	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction ^b	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weather-stripped, $\frac{1}{8}$ -in. crack and $\frac{1}{4}$ -in. clearance. Includes wood frame leakage	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped ^d	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weather-stripped, $\frac{1}{8}$ -in. crack and $\frac{1}{4}$ -in. clearance. Includes wood frame leakage	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped ^d	5.9	18.9	34.1	51.4	70.5	91.5
	Double-Hung Metal Windows ^f						
Rolled Section Steel Sash Windows ^k	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
	Industrial pivoted, $\frac{1}{8}$ -in. crack	52	108	176	244	304	372
	Architectural projected, $\frac{1}{8}$ -in. crack ^b	15	36	62	86	112	139
	Architectural projected, $\frac{1}{4}$ -in. crack ^b	20	52	88	116	152	182
	Residential casement, $\frac{1}{8}$ -in. crack ⁱ	6	18	33	47	60	74
	Residential casement, $\frac{1}{4}$ -in. crack ⁱ	14	32	52	76	100	128
	Heavy casement section, projected, $\frac{1}{8}$ -in. crack ^j	3	10	18	26	36	48
	Heavy casement section, projected $\frac{1}{4}$ -in. crack ^j	8	24	38	54	72	92
Hollow Metal, vertically pivoted window ^f		30	88	145	186	221	242

^aThe 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 at the end of this chapter.

^bThe 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 not-calked tests.

^cThe fit of the average double-hung wood window was determined as $\frac{1}{8}$ -in. crack and $\frac{1}{4}$ -in. clearance by measurements on approximately 600 windows under heating season conditions.

^dThe 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.

^eA $\frac{1}{8}$ -in. crack and clearance represents a poorly fitted window, much poorer than average.

^fWindows tested in place in building.

^gIndustrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

^hArchitectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms. $\frac{1}{8}$ in. crack is obtainable in the best practice of manufacture and installation, $\frac{1}{4}$ in. crack considered to represent average practice.

ⁱOf same design and section shapes as so-called *heavy section casement* but of lighter weight. $\frac{1}{8}$ in. crack is obtainable in the best practice of manufacture and installation, $\frac{1}{4}$ in. crack considered to represent average practice.

^jMade of heavy sections. Ventilators swing in or out and stay set at any degree of opening. $\frac{1}{8}$ in. crack is obtainable in the best practice of manufacture and installation, $\frac{1}{4}$ in. crack considered to represent average practice.

^kWith reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With $\frac{1}{8}$ -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.

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.

CHOOSING WIND VELOCITY

Although all authorities do not agree upon the value of the wind velocity that should be chosen for any given locality, it is common engineering practice to use the average wind velocity during the three coldest months of the year. Until this point is definitely established the practice of using average values will be followed. Average wind velocities for the months of December, January and February for various cities in the United States and Canada are given in Table 2, Chapter 7.

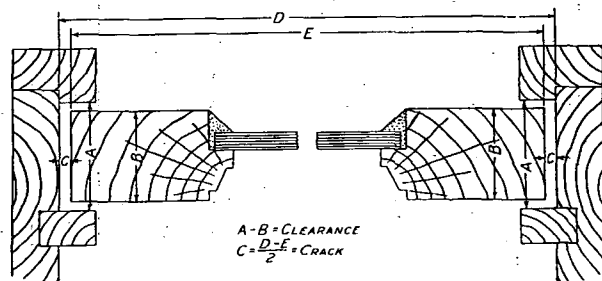


FIG. 3. DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

In considering both the transmission and infiltration losses, the more exact procedure would be to select the outside temperature and the wind velocity corresponding thereto, based on Weather Bureau records, which would result in the maximum heat demand. Since the proportion of transmission and infiltration losses varies with the construction and is different for every building, the proper combination of temperature and wind velocity to be selected would be different for every type of building, even in the same locality. Furthermore, such a procedure would necessitate a laborious cut-and-try process in every case in order to determine the worst combination of conditions for the building under consideration. It would also be necessary to consider heat lag due to heat capacity in the case of heavy masonry walls, and other factors, to arrive at the most accurate solution of the problem. Although heat capacity should be considered wherever possible, it is seldom possible to accurately determine the worst combination of outside temperature and wind velocity for a given building and locality. The usual procedure, as already explained, is to select an outside temperature based on the lowest on record and the average wind velocity during the months of December, January and February.

The direction of prevailing winds may usually be included within an angle of about 90 deg. The windows that are to be figured for prevailing

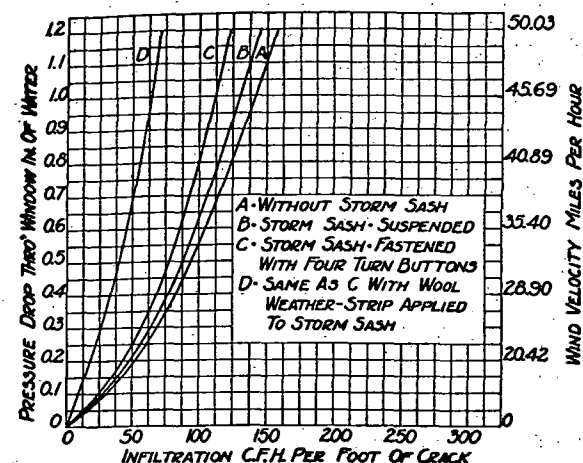


FIG. 4. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH— $\frac{5}{64}$ -IN. CRACK AND $\frac{1}{32}$ -IN. CLEARANCE

and non-prevailing winds will ordinarily each occupy about one-half the perimeter of the structure, the proportion varying to a considerable extent with the plan of the structure. (See discussion of wind movement in Chapter 4.)

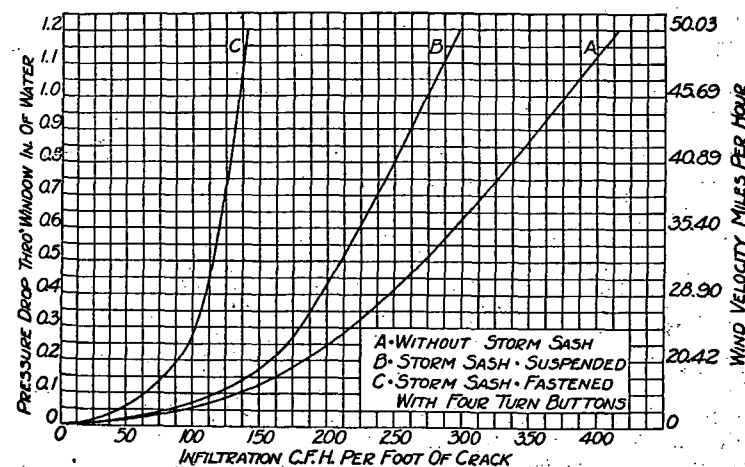


FIG. 5. INFILTRATION THROUGH SASH PERIMETER OF WINDOW WITH AND WITHOUT STORM SASH— $\frac{1}{8}$ -IN. CRACK AND $\frac{1}{8}$ -IN. CLEARANCE

CRACK USED FOR COMPUTATIONS

In no case should the amount of crack used for computation be less than half of the total crack in the outside walls of the room. Thus, 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. 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.

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 3. This method may be used to advantage as a check on the calculations made in the more exact manner.

TABLE 3. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION

KIND OF ROOM OR BUILDING	NUMBER OF AIR CHANGES TAKING PLACE PER HOUR
Rooms, 1 side exposed.....	1
Rooms, 2 sides exposed.....	1½
Rooms, 3 sides exposed.....	2
Rooms, 4 sides exposed.....	2
Rooms with no windows or outside doors.....	½ to ¾
Entrance Halls.....	2 to 3
Reception Halls.....	2
Living Rooms.....	1 to 2
Dining Rooms.....	1 to 2
Bath Rooms.....	2
Drug Stores.....	2 to 3
Clothing Stores.....	1
Churches, Factories, Lofts, etc.....	½ to 3

MULTI-STORY BUILDINGS

In tall buildings, infiltration may be considerably influenced by temperature difference or chimney effect which will operate to 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 levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors 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* is located at mid-height of a building, and that the temperature difference is 70 F, the following formulae 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:

$$M_e = \sqrt{M^2 - 1.75 a} \quad (1)$$

$$M_e = \sqrt{M^2 + 1.75 b} \quad (2)$$

where

M_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.

M = 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.

b = distance if *below* mid-height.

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, although probably no greater wind velocities should be figured at such extremely high levels.

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 high altitudes 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.

Heating Surface for Stair-Wells

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 even to the point of omitting all heating surface on the top several floor 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.

HEAT EQUIVALENT OF AIR ENTERING BY INFILTRATION

The heat required to warm cold, outside air, which enters a room by infiltration, to the temperature of the room is given by the following equation:

$$H_1 = 0.24 Q d (t - t_0) \quad (3)$$

where

H_1 = Btu per hour required for heating air leaking into building from outside temperature t_0 to inside temperature t .

Q = cubic feet of air entering per hour at inside temperature t .

d = density (pounds per cubic foot) of air at inside temperature t .

t = inside temperature at the proper level.

t_0 = outside air temperature for which heating system is designed.

0.24 = specific heat of air.

It is sufficiently accurate to take $d = 0.075$ lb, in which case the equation reduces to

$$H_1 = 0.018 Q (t - t_0) \quad (4)$$

While a heating reserve must be provided to warm inleaking air on the windward side of a building, this does not necessarily mean that the heating plant must be provided with a reserve capacity, since the inleaking air, warmed at once by adequate heating surface in exposed rooms, will move transversely and upwardly through the building, thus relieving other radiators of a part of their load. The actual loss of heat of a building caused by infiltration is not to be confused with the necessity for providing additional heating capacity for a given space. Infiltration is a disturbing factor in the heating of a building, and its maximum effect (maximum in the sense of an average of wind velocity peaks during the heating season above some reasonably chosen minimum) must be met by a properly distributed reserve of heating capacity, which reserve, however, is not in use at all places at the same time, nor in any one place at all times.

REFERENCES

- Air Leakage, by Houghten and Schrader (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924).
 Air Infiltration Through Various Types of Brick Wall Construction, by Larson, Nelson and Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).
 Infiltration through Plastered and Unplastered Brick Walls, by F. C. Houghten and Margaret Ingels (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927).
 Air Leakage around Window Openings, by C. C. Schrader (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924).
 Effect of Frame Calking and Storm Sash on Infiltration around and through Windows, by Richtmann and Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).
 Air Leakage on Metal Windows in a Modern Office Building, by Houghten and O'Connell (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).
 The Weathertightness of Rolled Section Steel Windows, by Emswiler and Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).
 Air Leakage through a Pivoted Metal Window, by Houghten and O'Connell (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).
 Pressure Difference across Windows in Relation to Wind Velocity, by Emswiler and Randall (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).
 Air Infiltration Through Various Types of Wood Frame Construction, by Larson, Nelson and Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).
 Neutral Zone in Ventilating, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926).
 Air Infiltration Through Double-Hung Wood Windows, by Larson, Nelson and Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

- Flue Action in Tall Buildings, by H. L. Alt (*Heating, Piping and Air Conditioning*, May, 1932).
 Air Infiltration Through Steel Framed Windows, by D. O. Rusk, V. H. Cherry and L. Boelter (*Heating, Piping and Air Conditioning*, October, 1932).
 Investigation of Air Outlets in Class Room Ventilation, by Larson, Nelson, and Kubasta (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).
 Influence of Stack Effect on the Heat Loss in Tall Buildings, by Axel Marin (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934).
 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).

PROBLEMS IN PRACTICE

1 ● Why is it important in exterior walls of air space construction to place a wind stop in the outer surface?

During the heating season, air within the space is at a higher temperature than outdoor air. If cracks are present in the outer surface, a chimney action takes place which causes a continuous change of air in the space. This causes a greater heat loss through the inner wall surface for two reasons: (1) the temperature difference becomes greater and (2) the surface coefficient on the air space side is increased because of the higher air velocity. Another undesirable condition resulting from this leakage is a lowered temperature of the inside surface. The increased radiation from occupants to walls caused by this condition must be offset by higher air temperatures.

The wind stop in the outer wall surface is therefore important because:

- It reduces the loss of heat through the wall.
- It helps to maintain a high inner wall surface temperature.

2 ● Why is it essential to consider this in heating calculations?

The inflowing air displaces inside heated air and must be heated up to the internal temperature.

3 ● Where is it necessary to consider infiltration created by temperature difference?

In tall, single-story buildings and in multi-story buildings where the floors are not adequately isolated.

4 ● Why is the infiltration in a building less than that determined in laboratory tests?

In laboratory tests, the indicated wind velocity is measured by the difference in pressure on the two sides of a single wall, window, or object tested. In a building, an internal back pressure is built up between its walls to a point where outflow on the lee side is equal to inflow on the windward side and this back pressure reduces the actual inflow below that determined in the laboratory for a comparable wind.

5 ● Is heat loss by infiltration through walls of importance?

Only in the case of simple walls or poorly constructed compound walls.

6 ● What measurements are required to calculate the heat loss through double-hung wood windows?

Sash crack (equal to the sash perimeter plus the meeting rail) and frame crack (equal to the frame perimeter).

7 ● What is the basis for selecting the wind velocity and outside temperature to be used in making infiltration calculations?

Weather Bureau records. The wind velocity taken is the average during the three coldest months and the temperature used is the lowest on record for the given locality.

8 ● How does the temperature difference influence the heat loss in a tall building?

The chimney effect caused by the temperature difference operates to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels.

9 ● For a wind velocity of 15 mph and a building 180 ft high, calculate the effective wind velocity at the ground floor and at a height of 150 ft.

a. At the ground floor the effective wind velocity would be

$$M_e = \sqrt{15^2 + 1.75 \times 90} = 19.6 \text{ mph}$$

b. At a floor 150 ft above the ground

$$M_e = \sqrt{15^2 - 1.75 \times 60} = 11.0 \text{ mph}$$

10 ● A room contains three 2 ft-8 in. by 5 ft-6 in. plain double-hung wood windows with $\frac{1}{8}$ -in. crack and $\frac{3}{4}$ -in. clearance. Assume a wind velocity of 20 mph and a temperature difference of 75 F. Neglecting chimney effect, what is the maximum heat loss due to infiltration?

From Table 2, heat loss per foot of crack per degree temperature difference is 1.067 Btu per hour. Length of crack for the three windows is 57 ft. The maximum heat loss, due to infiltration, is equal to $1.067 \times 57 \times 75$ or 4561 Btu per hour.

11 ● Find the infiltration through a wall with 16-in. shingles on 1 in. by 4 in. boards with 20 mph wind velocity. Give the pressure drop through the wall.

Referring to Curve 3C, Fig. 1, the value on the horizontal scale corresponding to 20 mph is approximately 102 cfh per square foot of wall.

The pressure drop through the wall is 0.193 in. of water (see left hand vertical scale).

12 ● What will be the infiltration through air-dried end and side-matched sheathing for 15 mph wind velocity?

Referring to Curve 10C, Fig. 2, the value on the horizontal scale corresponding to 15 mph is 50 cfh per square foot of wall.

13 ● From Table 2, find the infiltration (cubic feet per hour per foot of crack) for an average double-hung window, not weather stripped, with a 20 mph wind velocity.

59.3 cu ft per foot of crack per hour.

14 ● Using the value found in Question 11, what will be the heat requirement in a building with a total crack (all windows and doors) of 180 ft if the wind velocity is 15 mph, the outside temperature is 0 F, and the inside temperature is 70 F?

Using one half of the total crack, the volume of air is:

$$90 \times 59.3 = 5337 \text{ cu ft}$$

$$H = 0.018 \times 5337 \times (70 - 0) = 6724.6 \text{ Btu. (See Equation 4.)}$$

Chapter 7

HEATING LOAD

Factors Governing Heat Demand, Procedure, Temperatures, Wind Movement, Heat Sources Other Than Heating Plant, Example, Condensation

To design any system of heating, the maximum probable heat demand must be accurately estimated in order that the apparatus installed shall be capable of maintaining the desired temperature at all times. The factors which govern this maximum heat demand—most of which are seldom, if ever, in equilibrium—include the following:

- | | |
|---|---------------------------------------|
| 1. Outside temperature. | } Outside Conditions
(The Weather) |
| 2. Rain or snow. | |
| 3. Sunshine or cloudiness. | |
| 4. Wind velocity. | |
| 5. Heat transmission of exposed parts of building. | } Building
Construction |
| 6. Infiltration of air through cracks, crevices and open doors and windows. | |
| 7. Heat capacity of materials. | |
| 8. Rate of absorption of solar radiation by exposed materials. | |
| 9. Inside temperatures. | } Inside
Conditions |
| 10. Stratification of air. | |
| 11. Type of heating system. | |
| 12. Ventilation requirements. | |
| 13. Period and nature of occupancy. | |
| 14. Temperature regulation. | |

The *inside conditions* vary from time to time, the physical properties of the *building construction* may change with age, and the *outside conditions* are changing constantly. Just what the worst combination of all of these variable factors is likely to be in any particular case is therefore conjectural. Because of the nature of the problem, extreme precision in estimating heat losses at any time, while desirable, is hard of attainment.

The procedure to be followed in determining the heat loss from any building can be divided into seven consecutive steps, as follows:

1. Determine on the inside air temperature, at the breathing line or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 1.)
2. Determine on an outside air temperature for design purposes, based on the minimum temperatures recorded in the locality in question, which will provide for all but the most severe weather conditions. Such conditions as may exist for only a few consecutive hours are readily taken care of by the heat capacity of the building itself. (See Table 2.)

3. 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 5.)

4. Measure up 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.

5. 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 and 2.)

6. 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 6.)

7. The sum of the heat losses by transmission (Item 5) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (Item 6) of the cold air entering by infiltration represents the total heat loss equivalent for any building.

Item 7 represents the heat losses after the building is heated and under stable operating conditions in coldest weather. Additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to the specified standard inside temperature.

The rate at which this additional heat is required depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated.

This additional heat may be figured and allowed for as conditions re-

TABLE 1. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED^a

TYPE OF BUILDING	DEG FAHR	TYPE OF BUILDING	DEG FAHR
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—		HOMES.....	70-72
Private rooms.....	70-72	STORES.....	65-68
Private rooms (surgical).....	70-80	PUBLIC BUILDINGS.....	68-72
Operating rooms.....	70-95	WARM AIR BATHS.....	120
Wards.....	68	STEAM BATHS.....	110
Kitchens and laundries.....	66	FACTORIES AND MACHINE SHOPS.....	60-65
Toilets.....	68	FOUNDRIES AND BOILER SHOPS.....	50-60
Bathrooms.....	70-80	PAINT SHOPS.....	80

^aThe 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 3.

quire, 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 made except in the size of boilers or furnaces.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building and which should always be stated in the heating specifications 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 1 presents values which conform with good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 3. 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.¹

According to Fig. 5, Chapter 3, 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 given in Table 1, 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 1 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. In the case of heated spaces adjacent to unheated spaces, it will usually be sufficient to assume the temperature in such spaces as the mean between the temperature of the inside heated spaces and the outside air temperature, excepting where the combined heat transmission coefficient of the roof and ceiling can be used, in which case the usual inside and outside temperatures should be applied. (See discussion regarding the use of combined coefficients of pitched roofs, unheated attics and top-floor ceilings Chapter 5.)

¹See Chapter 3, p. 69.

High Ceilings: Research data concerning stratification of air in buildings are lacking, but in general it may be said that where the increase in temperature is due to the natural tendency of the warmer or less dense air to rise, as where a direct radiation system is installed, the temperature of the air at the ceiling increases with the ceiling height. The relation, however, is not a straight-line function, as the amount of increase per foot of height apparently decreases as the height of the ceiling increases, according to present available information².

Where ceiling heights are under 20 ft, it is common engineering practice to consider that the Fahrenheit temperature increases 2 per cent for each foot of height above the breathing line. This rule, sufficiently accurate for most cases, will give the probable air temperature at any given level for a room heated by direct radiation. Thus, the probable temperature in a room at a point three feet above the breathing line, if the breathing line temperature is 70 F, will be

$$(1.00 + 3 \times .02) 70 = 74.2 \text{ F.}$$

With certain types of heating and ventilating systems, which tend to oppose the natural tendency of warm air to rise, the temperature differential between floor and ceiling can be greatly reduced. These include unit heaters, fan-furnace heaters, and the various types of mechanical ventilating systems. The amount of reduction is problematical in certain instances, as it depends upon many factors such as location of heaters, air temperature, and direction and velocity of air discharge. In some cases it has been possible to reduce the temperature between the floor and ceiling by a few degrees, whereas, in other cases, the temperature at the ceiling has actually been increased because of improper design, installation or operation of equipment. So much depends upon the factors enumerated that it is not advisable to allow less than 1 per cent per foot (and usually more) above the breathing line in arriving at the air temperature at any given level for any of these types of heating and ventilating systems, unless the manufacturers are willing to guarantee that the particular type of equipment under consideration will maintain a smaller temperature differential for the specific conditions involved.

Temperature at Floor Level: In determining mean air temperatures just above floors which are next to ground or unheated spaces, a temperature 5 deg lower than the breathing-line temperature may be used, provided the breathing-line temperature is not less than 55 F.

OUTSIDE TEMPERATURES

The outside temperature used in computing the heat loss from a building is seldom taken as the lowest temperature ever recorded in a given locality. Such temperatures are usually of short duration and are rarely repeated in successive years. It is therefore evident that a temperature somewhat higher than the lowest on record may be properly assumed in making the heat-loss computations.

¹Temperature 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).

²Tests 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. Journal Section, *Heating, Piping and Air Conditioning*, November, 1934).

The outside temperature to be assumed in the design of any heating system is ordinarily not more than 15 deg above the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In the case of massive and well insulated buildings in localities where the minimum does not prevail for more than a few hours, or where the lowest recorded temperature is extremely unusual, more than 15 deg above the minimum may be allowed, due primarily to the *fly-wheel* effect of the heat capacity of the structure. The outside temperature assumed and used in the design should always be stated in the heating specifications. Table 2 lists the coldest dry-bulb temperatures ever recorded by the Weather Bureau at the places listed.

If Weather Bureau reports are not available for the locality in question, then the reports for the station nearest to this locality are to be used, unless some other temperature is specifically stated in the specifications. In computing the average heat transmission losses for the heating season in the United States the average outside temperature from October 1 to May 1 should be used.

WIND MOVEMENT

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. Therefore, pending further studies of actual buildings, it is recommended that the average wind movement in any locality during December, January and February be provided for in computing (1) the heat transmission of a building, and (2) the heat required to take care of the infiltration of outside air.

The first condition is readily taken care of, as explained in Chapter 5, by using a surface coefficient f_o for the outside wall surface which is based on the proper wind velocity. In case specific data are lacking for any given locality, it is sufficiently accurate to use an average wind velocity of approximately 15 mph which is the velocity upon which the heat transmission coefficient tables in Chapter 5 are based.

In a similar manner, the heat allowance for infiltration through cracks

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TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS^a

Col. A	Col. B	Col. C	Col. D	Col. E	Col. F
State	City	Average Temp., Oct. 1st-May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
Ala.	Mobile	58.9	-1	10.4	N
	Birmingham	53.8	-10	8.5	N
Ariz.	Phoenix	59.5	12	6.4	E
	Flagstaff	35.8	-25	7.8	SW
Ark.	Fort Smith	50.4	-15	8.1	E
	Little Rock	51.6	-12	8.7	NW
Calif.	San Francisco	54.2	27	7.6	N
	Los Angeles	58.5	28	6.3	NE
Colo.	Denver	38.9	-29	7.5	S
	Grand Junction	38.9	-21	5.3	NW
Conn.	New Haven	38.4	-15	9.7	N
D. C.	Washington	43.4	-15	7.1	NW
Fla.	Jacksonville	62.0	10	9.2	NE
Ga.	Atlanta	51.5	-8	12.1	NW
	Savannah	58.5	8	9.5	NW
Idaho	Lewiston	42.3	-23	5.3	E
	Pocatello	35.7	-28	9.6	SE
Ill.	Chicago	36.4	-23	12.5	W
	Springfield	39.8	-24	10.1	NW
Ind.	Indianapolis	40.3	-25	11.5	SW
	Evansville	45.1	-16	9.8	S
Iowa	Dubuque	33.9	-32	7.1	NW
	Sioux City	32.6	-35	11.6	NW
Kans.	Concordia	39.8	-25	8.1	S
	Dodge City	41.4	-26	9.8	NW
Ky.	Louisville	45.3	-20	9.9	SW
La.	New Orleans	61.6	7	8.8	N
	Shreveport	56.2	-5	8.9	SE
Me.	Eastport	31.5	-23	12.0	W
	Portland	33.8	-21	9.2	NW
Md.	Baltimore	43.8	-7	7.8	NW
Mass.	Boston	38.1	-18	11.2	W
Mich.	Alpena	29.6	-28	12.4	W
	Detroit	35.8	-24	12.7	SW
	Marquette	28.3	-27	11.1	NW
Minn.	Duluth	24.3	-41	12.6	SW
	Minneapolis	29.4	-33	11.3	NW
Miss.	Vicksburg	56.8	-1	8.3	SE
Mo.	St. Joseph	40.7	-24	9.3	NW
	St. Louis	43.6	-22	11.6	S
	Springfield	44.3	-29	10.8	SE
Mont.	Billings	34.0	-49	10.5	W
	Havre	27.6	-57	9.5	SW
Nebr.	Lincoln	37.0	-29	10.5	S
	North Platte	35.4	-35	8.5	W
Nev.	Tonopah	39.4	-10	10.0	SE
	Winnemucca	37.9	-28	8.7	NE
N. H.	Concord	33.3	-35	6.6	NW
N. J.	Atlantic City	41.6	-9	15.9	NW
N. Y.	Albany	35.2	-24	8.1	S
	Buffalo	34.8	-20	17.2	W
	New York	40.7	-14	17.1	NW

^aU. S. data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

CHAPTER 7—HEATING LOAD

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS^a—(Continued)

Col. A	Col. B	Col. C	Col. D	Col. E	Col. F
State or Province	City	Average Temp., Oct. 1st-May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hour	Direction of Prevailing Wind, Dec., Jan., Feb.
N. M.	Santa Fe	38.3	-13	7.8	NE
N. C.	Raleigh	50.0	-2	8.2	SW
	Wilmington	54.2	5	8.5	SW
N. Dak.	Bismarck	24.6	-45	9.1	NW
	Devils Lake	20.3	-44	10.6	W
Ohio	Cleveland	37.2	-17	13.0	SW
	Columbus	39.9	-20	12.0	SW
Okla.	Oklahoma City	47.9	-17	12.0	N
Oreg.	Baker	35.2	-24	6.9	SE
	Portland	46.1	-2	7.5	S
Pa.	Philadelphia	42.7	-6	11.0	NW
	Pittsburgh	41.0	-20	11.7	W
R. I.	Providence	37.2	-17	12.8	NW
S. C.	Charleston	57.4	7	10.6	SW
	Columbia	54.0	-2	8.1	NE
S. Dak.	Huron	28.2	-43	10.6	NW
	Rapid City	33.4	-34	8.2	W
Tenn.	Knoxville	47.9	-16	7.8	SW
	Memphis	51.1	-9	9.7	S
Texas	El Paso	53.5	-5	10.4	NW
	Fort Worth	55.2	-8	10.4	NW
	San Antonio	60.6	4	8.0	NE
Utah	Modena	36.3	-24	8.8	W
	Salt Lake City	40.0	-20	6.7	SE
Vt.	Burlington	31.5	-29	11.8	S
Va.	Norfolk	49.3	2	12.5	N
	Lynchburg	46.8	-7	7.1	NW
	Richmond	47.0	-3	7.9	SW
Wash.	Seattle	44.8	3	11.3	SE
	Spokane	37.7	-30	7.1	SW
W. Va.	Elkins	39.4	-28	6.6	W
	Parkersburg	42.6	-27	7.5	SW
Wis.	Green Bay	30.0	-36	10.4	SW
	La Crosse	31.7	-43	7.3	S
	Milwaukee	33.4	-25	11.5	W
Wyo.	Sheridan	30.7	-41	6.0	NW
	Lander	30.0	-40	5.0	SW
Alta.	Edmonton	23.0	-57	6.5	SW
B. C.	Victoria	43.9	-1.5	12.5	N
	Vancouver	42.0	2	4.5	E
Man.	Winnipeg	17.5	-47	10.0	NW
N. B.	Fredericton	27.0	-35	9.6	NW
N. S.	Yarmouth	35.0	-12	14.2	NW
Ont.	London	32.6	-27	10.3	SW
	Ottawa	26.5	-34	8.4	NW
	Port Arthur	22.4	-37	7.8	NW
	Toronto	32.9	-26.5	13.0	SW
P. E. I.	Charlottetown	29.0	-27	9.4	SW
Que.	Montreal	27.8	-29	14.3	SW
	Quebec	24.2	-34	13.6	SW
Sask.	Prince Albert	15.8	-70	5.1	W
Yukon	Dawson	2.1	-68	3.7	S

^aU. S. data from U. S. Weather Bureau.
Canadian data from Meteorological Service of Canada.

and walls (Tables 1 and 2, Chapter 6) must be based on the proper wind velocity for a given locality. In the case of *tall buildings* special attention must be given to infiltration factors. (See Chapter 6).

In the past many designers have used empirical *exposure factors* which were arbitrarily chosen to increase the calculated heat loss on the side or sides of the building exposed to the prevailing winds. It is also possible 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. Recent investigations indicate, however, that the wind direction indicated by Weather Bureau instruments does not always correspond with the direction of actual impact on the building walls, due to deflection by surrounding buildings.

The exposure factor, which is still in use by many engineers, is usually taken as 15 per cent, and is added to the calculated heat loss on the side or sides exposed to what is considered the prevailing winter wind. There is a need for actual test data on this point, and pending the time when it can be secured, the question must be left to the judgment of the designing engineer. It should be remembered that the values of *U* in the tables in Chapter 5 are based on a wind velocity of 15 mph and that the infiltration figures are supposed to be selected from the tables in Chapter 6 to correspond to the wind velocities given in Table 2 of the present chapter.

The *Heating, Piping and Air Conditioning Contractors National Association* has devised a method³ for calculating the square feet of equivalent direct radiation required in a building. This method makes use of exposure factors which vary according to the geographical location and the angular situation of the construction in question in reference to prevailing winds and the velocity of them.

HEAT FROM SOURCES OTHER THAN HEATING PLANT

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.

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

³See Standards of Heating, Piping and Air Conditioning Contractors National Association.

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 = $\frac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2,546$, and in the second case Btu per hour = $\text{bhp} \times 2,546$, in which 2,546 is the Btu equivalent of 1 hp-hour. In high-powered 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 3.415. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of illuminating gas gives off about 535 Btu per hour; and one cubic foot of natural gas gives off about 1000 Btu per hour. 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, see Chapter 3.

In intermittently heated buildings, besides the capacity necessary to care for the normal heat loss which may be calculated according to customary rules, additional capacity should be provided to supply the heat necessary to warm up the cold material of the interior walls, floors, and furnishings. Tests have shown that when a cold building has had its temperature raised to about 60 F from an initial condition of about 0 F, the heat absorbed from the air by the material in the structure may vary from 50 per cent to 150 per cent of the normal heat loss of the building. It is therefore necessary, in order to heat up a cold building within a reasonable length of time, to provide such additional capacity. If the interior material is cold when people enter a building, the radiation of heat from the occupants to the cold material will be greater than is normal and discomfort will result. (See Chapter 3.)

CONDENSATION ON BUILDING SURFACES⁴

Condensation on the interior surfaces of buildings is often a serious problem. Water dripping from a ceiling may cause irreparable damage to manufactured articles and machinery. It often results in short-circuiting of electric power and lighting systems, necessitating shut-downs and incurring costly repairs. It also causes rotting of wood roof structures, corrosion of metal roofs, and spalling and disintegration of gypsum and other types of roof decks not properly protected.

Condensation is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature, and can be remedied in two ways, (1) by increasing the temperature of such surfaces above the dew-point temperature, or (2) by lowering the humidity.

Dehumidification, of course, is not advisable where a high relative humidity is necessary for manufacturing processes. Hence, the only alternative is to increase the surface temperature by decreasing the inside

⁴See Preventing Condensation on Interior Building Surfaces, by Paul D. Close (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

surface resistance. This can be accomplished by increasing the velocity of air passing over the surface, or by increasing the over-all resistance of the wall or roof by installing a sufficient thickness of insulation.

The latter method is generally used, and the thickness of insulation is determined by ascertaining the amount of resistance to be added to increase the temperature of the interior surface above the dew-point temperature for the maximum conditions involved. This in turn is based on the fundamental principle that the drop in temperature is proportional to the resistance. See Question 1st at the end of this chapter.

EXAMPLES OF HEAT LOSS COMPUTATIONS

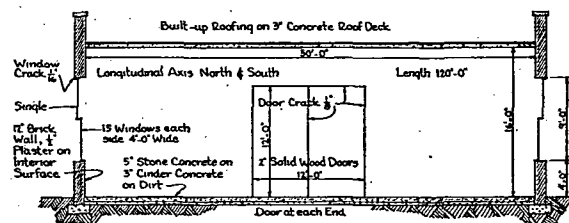


FIG. 1. ELEVATION OF FACTORY BUILDING

1. LOCATION.....Philadelphia, Pa.
2. LOWEST OUTSIDE TEMPERATURE. (Table 2).....- 6 F
3. BASE TEMPERATURE: In this example a design temperature 10 F above lowest on record instead of 15 F is used. Hence the base temperature =
 $(- 6 + 10) = + 4 \text{ F.}$
4. DIRECTION OF PREVAILING WIND (during Dec., Jan., Feb.).....Northwest
5. BREATHING-LINE TEMPERATURE (5 ft from floor).....60 F
6. INSIDE AIR TEMPERATURE AT ROOF:
 The air temperature just below roof is higher than at the breathing line. Height of roof is 16 ft, or it is $16 - 5 = 11$ ft above breathing line. Allowing 2 per cent per foot above 5 ft, or $2 \times 11 = 22$ per cent, makes the temperature of the air under the roof = $1.22 \times 60 = 73.2 \text{ F.}$
7. INSIDE TEMPERATURE AT WALLS:
 The air temperature at the mean height of the walls is greater than at the breathing line. The mean height of the walls is 8 ft and allowing 2 per cent per foot above 5 ft, the average mean temperature of the walls is $1.06 \times 60 = 63.6 \text{ F.}$ By similar assumptions and calculations, the mean temperature of the glass will be found to be 64.2 F and that of the doors 61.2 F.
8. AVERAGE WIND VELOCITY (Table 2).....11.0 mph
9. OVER-ALL DIMENSIONS (See Fig. 1).....120 x 50 x 16 ft
10. CONSTRUCTION:

Walls—12-in. brick, with $\frac{1}{2}$ -in. plaster applied directly to inside surface.
 Roof—3-in. stone concrete and built-up roofing.

Floor—5-in. stone concrete on 3-in. cinder concrete on dirt.
 Doors—One 12 ft x 12 ft wood door (2 in. thick) at each end.
 Windows—Fifteen, 9 ft x 4 ft single glass double-hung windows on each side.

11. TRANSMISSION COEFFICIENTS:

Walls—(Table 3, Chapter 5, Wall 2B)..... $U = 0.34$
 Roof—(Table 11, Chapter 5, Roofs 2A and 3A)..... $U = 0.77$
 Floor—(Table 10, Chapter 5, Floors 5A and 6A)..... $U = 0.63$
 Doors—(Table 13B, Chapter 5)..... $U = 0.46$
 Windows—(Table 13A, Chapter 5)..... $U = 1.13$

12. INFILTRATION COEFFICIENTS:

Windows—Average windows, non-weatherstripped, $\frac{1}{16}$ -in. crack and $\frac{3}{16}$ -in. clearance. The leakage per foot of crack for an 11-mile wind velocity is 25.0 cfh. (Determined by interpolation of Table 2, Chapter 6.) The heat equivalent per hour per degree per foot of crack is taken from Chapter 6.

$$25.0 \times 0.018 = 0.45 \text{ Btu per deg Fahr per foot of crack.}$$

Doors—Assume infiltration loss through door crack twice that of windows or $2 \times 0.45 = 0.90 \text{ Btu per deg Fahr per foot of crack.}$

Walls—As shown by Table 1, Chapter 6, a plastered wall allows so little infiltration that in this problem it may be neglected.

13. CALCULATIONS: See calculation sheet, Table 3.

TABLE 3. CALCULATION SHEET SHOWING METHOD OF ESTIMATING HEAT LOSSES OF BUILDING SHOWN IN FIG. 1

PART OF BUILDING	WIDTH IN FEET	HEIGHT IN FEET	NET SURFACE AREA OR CRACK LENGTH	COEFFICIENT	TEMP. DIFF.	TOTAL BTU
North Wall:						
Brick, $\frac{1}{2}$ -in. plaster.....	50	16	656	0.34	59.6	13,293
Doors (2-in. wood).....	12	12	144	0.46	57.2	3,789
$\frac{1}{16}$ in. Crack.....		1 pair doors	60	0.90	57.2	1,544*
West Wall:						
Brick, $\frac{1}{2}$ -in. plaster.....	120	16	1380	0.34	59.6	27,964
Glass (Single).....	15 x 4	9	540	1.13	60.2	36,734
$\frac{1}{16}$ in. Crack.....		Double Hung Windows (15)	450	0.45	60.2	6,095*
South Wall.....	Same as North Wall					18,626
East Wall.....	Same as West Wall					70,793
Roof, 3-in. concrete and slag-surfaced built-up roofing.....	50	120	6000	0.77	69.2	319,704
Floor, 5-in. stone concrete on 3-in. cinder concrete.....	50	120	6000	0.63	5 ^b	18,900
GRAND TOTAL of heat required for building in Btu per hour.....						517,442

*This building has no partitions and whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, only one-half of the total crack will be used in computing infiltration for each side and each end of building.

^bA 5 F temperature differential is commonly assumed to exist between the air on one side of a large floor laid on the ground and the ground.

PROBLEMS IN PRACTICE

1 • What is the relation between the sensible heat loss from a building and the heat required for humidification?

A house with a volume of 14,000 cu ft has a heat loss 120 Mbh for standard uninsulated frame construction and a 70 F temperature difference. Assuming a leakage rate of $1\frac{1}{2}$ air changes per hour it would require about 10 Mbh to maintain a relative humidity of 45 per cent when the outside air is 0 F and 50 per cent relative humidity. By using an insulation such as rock wool, the sensible heat loss of this house may be reduced to approximately 77 Mbh. The insulation does not affect the humidification load, which now assumes greater importance.

2 • What inside dry-bulb temperatures are usually assumed for: (a) homes, (b) schools, (c) public buildings?

Referring to Table 1:

a. 70 to 72 F.

b. Temperature varies from 55 to 75 F, depending on the room. Classrooms, for instance, are usually specified as 70 to 72 F.

c. 68 to 72 F.

3 • How is the outside temperature selected for use in computing heat losses?

The outside temperature used in computing heat losses is generally taken from 10 to 15 F higher than the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In some cases where the lowest recorded temperature is extremely unusual, the design temperature is taken even higher than 15 F above the lowest recorded temperature.

4 • What are the effects of wind movement on the heating load?

a. Wind movement increases the heat transmission of walls, glass, and roof; it affects poor walls to a much greater extent than good walls.

b. 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.

5 • Calculate the heat given off by eighteen 200-watt lamps.

$$200 \times 18 \times 3.415 = 12,294 \text{ Btu per hour.}$$

6 • A two-story, six room, frame house, 28-ft by 30-ft foundation, has the following proportions:

Area of outside walls, 1992 sq ft.

Area of glass, 333 sq ft.

Area of outside floors, 54 sq ft.

Cracks around windows, 440 ft.

Cracks around doors, 54 ft.

Area of second floor ceiling, 783 sq ft.

Volume, first and second floors, 13,010 cu ft.

Ceilings, 9 ft high.

The minimum temperature for the heating season is -34 F, and the required inside temperature at the 30-in. level is 70 F. The average number of degree days for a heating season is 7851, and the average wind velocity is 10 mph, northwest.

The walls are constructed of 2-in. by 4-in. studs with wood sheathing, building paper, and wood siding on the outside, and wood lath and plaster on the inside. Windows are single glass, double-hung, wood, without weatherstrips. The second floor ceiling is metal lath and plaster, without an attic floor. The roof is of wood shingles on wood strips with rafters exposed. The area of the roof is 20 per cent greater than the area of the ceiling. Select values for the following: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for roof;

CHAPTER 7—HEATING LOAD

(e) U for ceiling and roof combined; (f) air leakage, cubic feet per hour per foot of window crack; (g) air leakage, cubic feet per hour per foot of door crack.

a. 0.25 (Table 5, Chapter 5).

b. 1.13 (Table 13, Chapter 5).

c. 0.69 (Table 8, Chapter 5).

d. 0.46 (Table 12, Chapter 5).

e. 0.236 (Equation 6, Chapter 5).

f. 21.4 (Table 2, Chapter 6).

g. 42.8, which is double the window leakage.

7 • Using the data of Question 6, calculate the maximum Btu loss per hour for the various constructions, and show the percentage of the total heat which is lost through each construction described.

Assume 2 per cent rise in temperature for each foot in height. The average temperature will be 72.8 F for walls, doors, and windows, and 79.1 F for the second floor ceiling.

a. Outside walls	46,200 Btu loss	37.2 per cent of total
b. Glass	34,950 Btu loss	28.1 per cent of total
c. Doors	5,670 Btu loss	4.6 per cent of total
d. Second floor ceiling	17,840 Btu loss	14.3 per cent of total
e. Air leakage, windows	15,750 Btu loss	12.7 per cent of total
f. Air leakage, doors	3,865 Btu loss	3.1 per cent of total

Total 124,275 Btu loss 100.0 per cent of total

8 • For the house in Question 6, place 1-in. insulation in the outside walls and second floor ceiling; k for insulation = 0.34. Use weatherstrip on doors and windows, and double glass on the windows; $C_a = 0.55$. Calculate or select the following values: (a) U for walls; (b) U for glass; (c) U for second floor ceiling; (d) U for combination of ceiling and roof; (e) air leakage, cubic feet per hour per foot of door crack; (f) air leakage, cubic feet per hour per foot of window crack.

a. 0.144.

b. 0.55.

c. 0.23.

d. 0.13.

e. 15.5.

f. 31.0.

9 • Calculate the maximum Btu loss per hour and show the percentage loss by each channel for the house as insulated in Question 8.

a. Outside walls	26,650 Btu loss	36.2 per cent of total
b. Glass	17,000 Btu loss	23.1 per cent of total
c. Doors	5,670 Btu loss	7.7 per cent of total
d. Ceiling	10,070 Btu loss	13.7 per cent of total
e. Air leakage, windows	11,400 Btu loss	15.5 per cent of total
f. Air leakage, doors	2,795 Btu loss	3.8 per cent of total

Total 73,585 Btu loss 100.0 per cent of total

10 • From the results of Questions 7 and 9, calculate the Btu saved and the percentage saved by each change in construction.

	UNINSULATED	INSULATED	BTU SAVED	PER CENT SAVED
a. Outside walls.....	46,200	26,650	19,550	42.3
b. Glass.....	34,950	17,000	17,950	51.4
c. Doors.....	5,670	5,670	0	0
d. Ceiling.....	17,840	10,070	7,770	43.5
e. Air leakage, windows.....	15,750	11,400	4,350	27.6
f. Air leakage, doors.....	3,865	2,795	1,070	27.7

11 • From the results of Questions 7 and 9, calculate the heat loads per heating season in Btu and note the savings by better construction.

The 7851 degree days for the heating season multiplied by 24 hours, times the Btu loss per hour for 1 F drop in temperature gives the Btu load per heating season.

Saving = 250,800,000 - 148,000,000 = 102,800,000 Btu.

Chapter 8

COOLING LOAD

Conditions to be Maintained, Cooling Load, Transmission for Surfaces not Exposed to the Sun, Outside Temperatures, Solar Radiation, Time Lag, Transmission of Solar Radiation Through Glass, Heat and Moisture Leakage, Heat and Moisture Sources

THE method of calculating the cooling load is similar to that used in calculating the heating load. The direction of the flow of heat is reversed, however, and in most cases additional factors must be considered, such as solar radiation and the heat from occupants, lights, motors, and other sources. The character of the load depends on the type of building to be cooled as, for example, in auditoriums and other places of assemblage where the maximum load usually is that due to the heat and moisture given off by the occupants, or in office buildings and residences where solar radiation and the transmission and infiltration of heat through the building shell are most important.

While cooling is generally identified with the summer season, it is often necessary to cool in winter as well as in summer. In a crowded place of assemblage the heat given off by the occupants, together with that given off by the lighting and power equipment, may be more than the normal heat loss through the structure even in winter under cold climatic conditions.

Much of the basic information for the design of comfort conditioning installations has resulted from research conducted at the A.S.H.V.E. Research Laboratory and at institutions with which coöperative research investigations have been carried on. These data include the effective temperature index, and heat and moisture loss data given in Chapter 3.

COMFORT CONDITIONS

The conditions to be maintained in an enclosure are variable and depend on many factors, especially the season of the year and (during the summer) the outside dry-bulb temperature and the duration of the period of occupancy. Information concerning the proper effective temperatures to be maintained for various seasons is given in Chapter 3, where are also tabulated the most desirable indoor air conditions to be maintained in summer for exposures less than three hours. (See Table 2, Chapter 3.)

In installations for restaurants and theaters the requirements are different from those in offices, since there must be a considerable volume of air circulated in order to provide ventilation and cooling.

TABLE 1. 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	93	77	5.2	S
	Mobile	94	78	8.6	SW
Ariz.	Phoenix	110	77	6.0	W
Ark.	Little Rock	95	77	7.0	NE
Calif.	Los Angeles	88	70	6.0	SW
	San Francisco	85	68	11.0	SW
Colo.	Denver	90	64	6.8	S
Conn.	New Haven	88	74	7.3	S
D. C.	Washington	95	78	6.2	S
Fla.	Jacksonville	94	78	8.7	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	91	75	7.3	NW
	Savannah	95	79	7.8	SW
Idaho.	Boise	95	65	5.8	NW
Ill.	Chicago	95	75	10.2	NE
	Peoria	91	75	8.2	S
Ind.	Indianapolis	90	73	9.0	SW
Iowa.	Des Moines	92	74	6.6	SW
Ky.	Louisville	94	75	8.0	SW
La.	New Orleans	94	79	7.0	SW
Maine	Portland	85	71	7.3	S
Md.	Baltimore	93	76	6.9	SW
Mass.	Boston	88	73	9.2	SW
Mich.	Detroit	93	73	10.3	SW
Minn.	Minneapolis	93	72	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	92	75	9.5	S
	St. Louis	95	78	9.4	SW
Mont.	Helena	87	63	7.3	SW
Nebr.	Lincoln	93	74	9.3	S
Nev.	Reno	93	64	7.4	W
N. J.	Trenton	95	76	10.0	SW
N. Y.	Albany	90	74	7.1	S
	Buffalo	83	72	12.2	SW
	New York	95	75	12.9	SW
N. M.	Santa Fe	87	63	6.5	SE
N. C.	Asheville	87	72	5.6	SE
	Wilmington	93	79	7.8	SW
N. Dak.	Bismarck	88	69	8.8	NW
Ohio.	Cleveland	95	75	9.9	S
	Cincinnati	95	78	6.6	SW
Okla.	Oklahoma City	96	76	10.1	S
Oreg.	Portland	83	65	6.6	NW
Pa.	Philadelphia	95	78	9.7	SW
	Pittsburgh	91	73	9.0	NW
R. I.	Providence	85	73	10.0	NW
S. C.	Charleston	94	80	9.9	SW
	Greenville	93	76	6.8	NE
Tenn.	Chattanooga	94	76	6.5	SW
	Memphis	93	77	7.5	SW
Texas	Dallas	99	76	9.4	S
	Galveston	93	79	9.7	S
	San Antonio	100	78	7.4	SE
	Houston	93	79	7.7	S
	El Paso	98	69	6.9	E

TABLE 1. DESIGN DRY- AND WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Continued)

STATE	CITY	DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Utah	Salt Lake City	95	67	8.2	SE
Vt.	Burlington	85	71	8.9	S
Va.	Norfolk	91	76	10.9	S
	Richmond	95	78	6.2	SW
Wash.	Seattle	83	61	7.9	S
	Spokane	89	63	6.5	SW
W. Va.	Parkersburg	90	74	5.3	SE
Wis.	Madison	89	73	8.1	SW
	Milwaukee	93	74	10.4	S
Wyo.	Cheyenne	85	62	9.2	S

COOLING LOAD

The cooling load may be divided into the following parts:

1. Transmission of heat through walls, roof, and glass with allowances for sun-exposed surfaces and heat capacity.
2. Transmission of solar radiation through glass and absorption by interior furnishings.
3. Heat and moisture from infiltration and from outside air introduced.
4. Heat and moisture from occupants and heat from lights, machinery and other sources.

Transmission for Surfaces Not Exposed to the Sun

The transmission load for surfaces not exposed to the sun is calculated in a manner similar to that described in Chapter 7, by means of the following formula:

$$H_t = AU(t_o - t) \quad (1)$$

where

H_t = heat transmitted through the material of the wall, glass, roof, or floor, Btu per hour.

A = net inside area of wall, glass, roof, or floor, square feet.

t = inside temperature, degrees Fahrenheit.

t_o = outside temperature, degrees Fahrenheit.

U = coefficient of transmission of wall, floor, roof or glass, Btu per hour per square foot per degree Fahrenheit difference in temperature. (Tables 3 to 13, Chapter 5.)

Outside Temperatures

Summer dry-bulb and wet-bulb temperatures for various cities are given in Table 1. It will be noted that 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 per cent to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system covering this range. The temperatures shown in Table 1 are based on a study of hourly temperatures in New York City from which factors were derived and applied to

the average maximum dry- and wet-bulb temperatures for other cities. This study covered a twenty-year record of Weather Bureau temperatures. The design temperatures given are not exceeded more than 5 to 8 per cent of the time during a cooling season of 1200 hours in June, July, August, and September for an average year.

Solar Radiation

Fig. 1 shows the total amount of solar energy in Btu per square foot per hour received during the day by a surface normal to the rays of the sun, by a horizontal surface, and by east, west, and south walls. The curves are drawn from A.S.H.V.E. Laboratory data obtained by pyrheliometer, are based on sun time, and are for a perfectly clear day on August 1 at a north latitude of 40 deg. Data from these curves may be used with little error for most United States latitudes and for all of the hotter months of the year.

The absorption of solar radiation by a surface depends upon the character of the surface and the angle of the surface with respect to the direction of the radiation. The heat absorption by a black oilcloth surface perpendicular to the sun's rays was found to be as high as 273 Btu per square foot per hour, based on tests conducted by the A.S.H.V.E. Research Laboratory in Pittsburgh¹. Lamp black, red brick dust, and aluminum bronze painted surfaces perpendicular to the sun's rays showed, respectively, 94.0, 63.4, and 28.2 per cent as high a rate of absorption as the black oilcloth.

TABLE 2. ALLOWANCE FOR SOLAR RADIATION ON ROOFS AND WALLS
APPROXIMATE NUMBER OF DEGREES TO ADD TO DRY-BULB TEMPERATURE
FOR DIFFERENT TYPES OF SURFACES

TYPE OF SURFACE	BLACK	RED BRICK OR TILE	ALUMINUM PAINT
Roof, horizontal.....	45	30	15
East or west wall.....	30	20	10
South wall.....	15	10	5

Solar radiation is an important factor in the mechanism of heat flow into buildings. Research conducted at the A.S.H.V.E. Research Laboratory² has shown that a large error may be introduced into the calculations by failure to consider the periodical character of heat flow resulting from the diurnal movement of the sun and the heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall into a building on a hot, sunny day.

Unfortunately, the calculations for the transmission of heat from solar radiation through building walls are too complicated to be of much

¹Absorption of Solar Radiation in Relation to the Temperature, Color, Angle, and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

²For further information on this subject see following A.S.H.V.E. research papers: Coefficients of Heat Transfer as Measured under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928); Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930); 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).

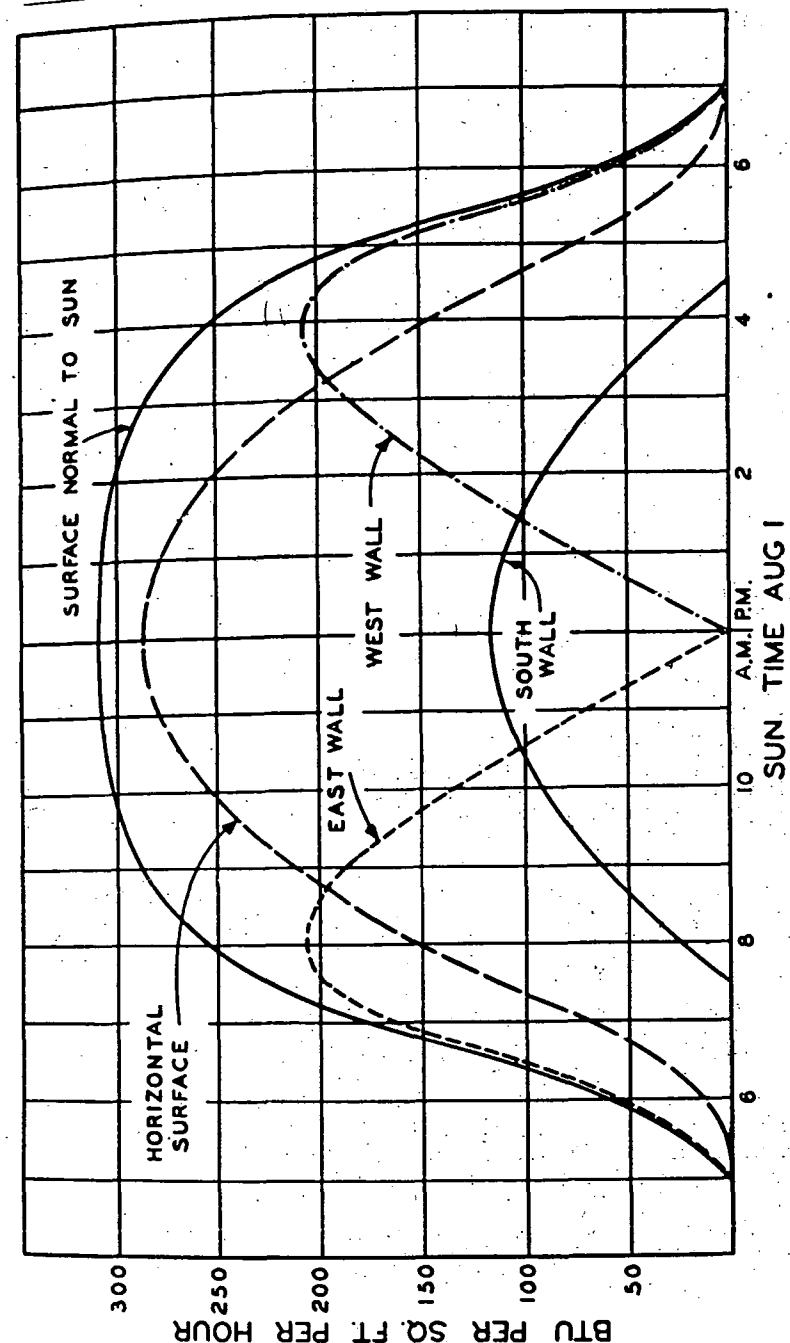


FIG. 1. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1

practical value to the heating and ventilating engineer. Approximate results may be obtained by adding the number of degrees given in Table 2 to the outside design dry-bulb temperature in calculating the heat transmission through a wall or roof which may be exposed to the sun for an appreciable length of time. Table 2 was obtained from a study of the data in A.S.H.V.E. research papers on solar radiation^{1, 2}. Black and aluminum painted surfaces represent the extremes which are likely to occur. For other types of surfaces, values intermediate between those given in the table can be used.

Time Lag

The calculation of heat transmitted through walls and roofs does not take into consideration the heat capacity of the structure and the consequent time lag in the transmission of heat. In the thick walls used in modern office buildings the time lag may amount to 10 hours or more³. Thus in many cases the wall transmission cannot be added directly to the cooling load from other sources because the peak of the wall transmission load may not coincide with the peak of the total cooling load and may even occur after the cooling system has been shut down for the day. The data in Table 3 were taken from A.S.H.V.E. research papers^{3, 4} and while they result principally from a study of experimental slabs, they give an idea of the time lag to be expected in various structures.

TABLE 3. TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

TYPE AND THICKNESS OF WALL OR ROOF	TIME LAG, HOURS
2-in. pine.....	1½
6-in. concrete.....	3
4-in. gypsum.....	2½
3-in. concrete and 1-in. cork.....	2
2-in. iron and cork (equivalent to ¾-in. concrete and 2.15-in. cork).....	2½
4-in. iron and cork (equivalent to 5½-in. concrete and 1.94-in. cork).....	7½
8-in. iron and cork (equivalent to 16-in. concrete and 1.53-in. cork).....	19
22-in. brick and tile wall.....	10

In intermittently cooled buildings the cooling capacity must be sufficient to care for the load imposed by the necessity to cool down the furnishings and the material of the interior construction to the point of maintained temperatures.

Transmission of Solar Radiation Through Glass

In considering the transmission through glass several factors must be considered. As the sun's rays impinge against a pane of glass, most of the radiation passes through to the other side, a small amount is reflected, and the balance is absorbed by the glass. The amount absorbed depends upon the character and thickness of the glass and the angle between the sun's rays and the glass. The temperature of the glass is raised by the absorbed

¹Heat 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).

⁴Field Studies of Office Building Cooling, by J. H. Walker, S. S. Sanford, and E. P. Wells (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

heat and this heat is then delivered to the air on the two sides of the glass in proportion to the difference between glass and air temperatures.

The A.S.H.V.E. tests indicated that a single pane of double strength glass 0.127 in. thick absorbs approximately 11 per cent of the solar radiation passing through it when the impingement is normal. For smaller angles of impingement, the glass retards percentages of the total radiant energy approximately in proportion to the sine of the angle. Other experiments⁴ indicate a glass absorption of 16.7 per cent for one pane of glass and 37.5 per cent for two ¼-in. panes separated by a 1¼-in. air space.

The amount of solar radiation delivered to an unshaded glass surface may be obtained from the curves in Fig. 1. For surfaces other than those given, the solar radiation incident to the glass must be calculated. Hendrickson and Walker⁵ have shown how this may be done if the wall faces some direction other than east, west, or south. They have also shown how to calculate the net glass area on which the solar radiation impinges when the glass is partly shaded by the frame or wall. The values from Fig. 1 must be used only for the net glass area on which the sun shines. Tests at the A.S.H.V.E. Research Laboratory⁶ have determined the percentages of heat from solar radiation actually delivered to a room with bare windows and with various types of outdoor and indoor shading. The data in Table 4 are taken from these tests.

TABLE 4. SOLAR RADIATION TRANSMITTED THROUGH BARE AND SHADED WINDOWS

TYPE OF APERTURANCE	PER CENT DELIVERED TO ROOM
Bare window glass.....	97
Canvas awning.....	28
Inside shade, fully drawn.....	45
Inside shade, one-half drawn.....	68
Inside Venetian blind, fully covering window.....	58
Outside Venetian blind, fully covering window.....	22

The percentage figures in this table were obtained by dividing the total amount of heat actually entering through the shaded window by the total amount of heat calculated to enter through a bare window (solar radiation plus glass transmission based on observed outside glass temperature). For bare windows on which the sun shines, the transmission of heat from outside air to glass is small or negative as the glass temperature is raised by the solar radiation absorbed. Therefore, in calculating the total heat gain through windows on the sunny sides of buildings, it is sufficiently accurate to figure the total cooling load due to the window, as the solar radiation times the proper factor from Table 4, and to neglect the heat transmission through the glass caused by the difference between the temperatures of the inside and outside air. Another reason for neglecting this glass transmission load is that the curves in Fig. 1 were

⁵Summer Cooling for Comfort as Affected by Solar Radiation, by G. A. Hendrickson and J. H. Walker, *Heating and Ventilating*, November, 1932, and The Determination of Sun Effect on Summer Cooling Loads, by G. A. Hendrickson and J. H. Walker, *Heating and Ventilating*, June, 1933.

⁶Studies 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).

based on the maximum intensity of solar radiation observed at the A.S.H.V.E. Laboratory during a three-year study, so results based on these curves will be amply high. It will be noted that Table 4 gives the amount of heat delivered through the window as 97 per cent of the solar radiation, which is greater than is indicated by the figures for absorption in the preceding paragraph. The explanation is that much of the radiation absorbed by the glass is delivered to the room.

Although 97 per cent of the heat from solar radiation is delivered to a room through bare window glass, more recent tests⁷ have indicated that in the case of buildings having floors of high heat capacity such as concrete floors on which the solar radiation falls, approximately one-half of the heat entering a bare window is absorbed by the floor and does not immediately become a part of the cooling load but is delivered back to the air in the building at a slow rate over a period of 24 hours or longer.

Fig. 1 shows that the maximum solar intensity on any surface is of limited duration. 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. The cooling load due to solar radiation therefore does not have to be figured as a steady load. Another point which should be noted is that the maximum solar radiation load on an east wall occurs early in the morning when the outside temperature is low.

In a paper by the A.S.H.V.E. Research Laboratory⁸ it was shown that ordinary double strength window glass transmits no measurable amount of energy radiated from a source at 500 F or lower; that it transmits only 6.0 and 12.3 per cent of the total radiation from surfaces at 700 F and 1000 F, respectively; and that it transmits 65.7 per cent of the radiation from an arc lamp, 76.3 per cent of the radiation from an incandescent tungsten lamp, and 89.9 per cent of the radiation from the sun. Thus, glass windows in a room constitute heat traps, which allow rather free transmission of radiant energy into the room from the sun to warm objects in it, but do not allow the transmission of re-radiated heat from these same objects.

Tests⁴ have been made which indicated that sunshine through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sunshine load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the amount of cooling required will be reduced and there will also be less difference in the cooling requirements of different sides of the building. The total cooling load for a building exposed to the sun on more than one side is of course less than the sum of the maximum cooling loads in the individual rooms since the maximum solar radiation load on the different sides occurs at different times. In determining the total cooling load for a building if the time

⁷Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutterlet, and Albert J. Wahl (A.S.H.V.E. JOURNAL SECTION, Heating, Piping and Air Conditioning, April, 1935).

⁸Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934).

when the maximum load occurs is not obvious, the load should be calculated for various times of day to determine the time at which the sum of the loads on the different sides of the building is a maximum.

Heat and Moisture Leakage

An allowance must be made for the heat and moisture in the outside air introduced for ventilating purposes or entering the building through cracks, crevices, doors, and other places where infiltration might occur.

The volume of air entering due to infiltration may be estimated from data given in Chapter 6 using wind velocities from Table 1, Chapter 8. Information on the amount of outside air required for ventilation will be found in Chapter 3.

The heat gain resulting from the outside air introduced may be estimated from the following formula:

$$H_i = 60 Q d_o (\Theta_o - \Theta) \quad (2)$$

where

H_i = heat to be removed from outside air entering the building, Btu per hour.

Q = volume of outside air entering the building, cubic feet per minute.

d_o = density of outside air, pounds of dry air per cubic foot of outside air, at the temperature t_o .

Θ_o = heat content of mixture of outside dry air (at temperature t_o) and water vapor, Btu per pound of dry air.

Θ = heat content of mixture of inside dry air (at temperature t) and water vapor, Btu per pound of dry air.

Heat and Moisture Sources

Figs. 8 to 11, Chapter 3, show the heat and moisture given off by human beings under various conditions of activity. For average conditions where a person is normally at rest, as in a theater, or doing very light work, as in a restaurant or residence, the total amount of heat given off will average

TABLE 5. HEAT GAIN DUE TO VARIOUS DEVICES, BTU PER HOUR

Lights and electrical appliances.....	3,413 per kw
Motors with connected load in same room ^a	
Nameplate rating, 1/8 to 1/2 hp.....	4,250 per hp
Nameplate rating, 1/2 to 3 hp.....	3,700 per hp
Nameplate rating, 3 to 20 hp.....	2,950 per hp
Restaurant coffee urns, 10-gal capacity.....	16,000
Dish warmers per 10 sq ft of shelf.....	6,000
Restaurant range—4 burners and oven.....	100,000
Residence gas range.....	
Giant burner.....	12,000
Medium burner.....	9,000
Oven.....	1,000 per cu ft of space
Pilot.....	250
Electric range.....	
Small burner, 1000 to 1350 watts.....	3,413 per kw
Large burner, 1700 to 2200 watts.....	3,413 per kw
Oven, 2000 to 3000 watts.....	3,413 per kw
Appliance connection, 660 watts.....	2,250
Warming compartment, 300 watts.....	1,025

^aDeduct 2545 Btu per hp if connected load is outside of room.

about 400 Btu per hour. Part of this is latent heat due to the evaporation of 700 to 1200 grains of moisture per hour. Examples illustrating heat and moisture loss calculations for human beings are given in Chapter 3.

All sources of heat must of course be considered in designing the conditioning system. The heat gain due to various devices is given in Table 5.

Moisture evaporated by appliances must be included in the total latent heat load. In some cases only a small part of the heat from lights immediately affects the cooling load. Tests show that with lights placed near the ceiling under some conditions the electricity used for illumination has little effect on the cooling load in an office cooled during the usual short period. The air heated by the lights stratifies closely to the ceiling and the temperature of the lower layers of air is raised only a small amount after a considerable lapse of time. An example of cooling load calculation is given in Chapter 9.

PROBLEMS IN PRACTICE

1 ● In buildings such as an office building which is cooled intermittently, will the maximum cooling load occur coincidently with the maximum dry-bulb temperature?

Not necessarily. Tests indicate that particularly for east and south exposures the maximum rate of cooling occurs shortly after the equipment is started in the morning. This extra heat is that which was absorbed by the building during the preceding day and also during the off period. The rapid lowering of the room temperature after the cooling equipment is started causes this stored up heat to flow quickly from wall and floor surfaces to the room air.

2 ● The outdoor and indoor temperatures are 90 F and 78 F, respectively. What is the amount of heat transmitted per hour through a 7 ft by 4 ft north window?

$$H_t = 28 \times 1.13 (90 - 78) = 380 \text{ Btu per hour.}$$

(Equation 1, Chapter 8 and Table 13, Chapter 5.)

3 ● What are the proper design temperatures for a Detroit store?

Outdoor dry-bulb, 93 F; wet-bulb, 73 F. (Table 1, Chapter 8.)

Indoor dry-bulb, 79.2 F; wet-bulb, 64.8 F. (Table 2, Chapter 3.)

4 ● a. What is the maximum heat transmission for a flat roof exposed to the sun with the outdoor and indoor temperature 95 F and 80 F, respectively? The roof is of uninsulated 6-in. concrete, with its underside exposed, and with a black upper surface.

b. If the temperatures specified were the maximum for the day and occurred at 12 o'clock, at what time would the maximum cooling load due to the roof exist?

$$a. H_t = 1 \times 0.64 (95 + 45 - 80) = 38.4 \text{ Btu per hour per square foot.}$$

(Equation 1 and Table 2, Chapter 8, and Table 11, Chapter 5.)

b. At 3 p.m. (Table 3.)

5 ● For south windows equipped with canvas awnings, what is the maximum amount of heat delivered to a room when the outdoor temperature is 90 F and the indoor temperature is 78 F?

$115 \times 0.28 = 32.2$ Btu per square foot of glass (Fig. 1 and Table 4; note that glass transmission can be neglected).

6 ● What is the heat gain per cubic foot of outside air introduced, under the following conditions:

Outdoor temperatures, 90 F dry-bulb and 75 F wet-bulb.

Inside temperatures, 78 F dry-bulb and 65 F wet-bulb.

$H_i = 60 Q d_o (\Theta_o - \Theta_i)$. (From Equation 2.)

The relative humidity of the outdoor air is 50 per cent (Psychrometric Chart), and the volume of 1 lb of air is $\frac{13.84 + 14.53}{2} = 14.19$ cu ft (Table 5, Chapter 1).

$$d_o = \frac{1}{14.21} = 0.0704.$$

$\Theta_o = 38.46$ and $\Theta_i = 29.96$ (Table 5, Chapter 1). The total heat of any air-vapor mixture may be obtained from the last column in Table 5, Chapter 1, by considering the temperatures to be wet-bulb readings, since the total heat of a mixture is constant for a given wet-bulb temperature.

$$H_i = 0.0704 (38.46 - 29.96) = 0.598 \text{ Btu per cu ft.}$$

7 ● If there are twenty 200-watt lights in use in a room, what is the cooling load due to lights?

$$200 \times 20 = 4000 \text{ watts} = 4 \text{ kw.}$$

$$3413 \times 4 = 13,652 \text{ Btu per hour (Table 5, Chapter 8).}$$

8 ● a. If a restaurant has two 10-gal coffee urns, what is the cooling load due to them?

b. What is the cooling load due to four 1350-watt burners on an electric range?

$$a. 16,000 \times 2 = 32,000 \text{ Btu per hour (Table 5, Chapter 8).}$$

$$b. 1.35 \times 4 \times 3413 = 18,430 \text{ Btu per hour (Table 5, Chapter 8).}$$

9 ● Why may the heat gain by transmission through glass on the sunny exposures be neglected?

Solar radiation on a glass surface during peak hours will cause the outer surface of the glass to rise up to or slightly above the simultaneous air temperature, hence no heat can flow from outside air to the glass at this time.

10 ● What should be the dry- and wet-bulb temperatures in a restaurant when the outdoor dry-bulb temperature is 95 F?

Dry-bulb, 80 F; wet-bulb, 65 F (Table 2, Chapter 3).

11 ● In an office building in which the occupants remain indoors most of the day, what is the most desirable indoor dry-bulb temperature and relative humidity in summer?

76.5 F and 50 per cent relative humidity (Fig. 5, Chapter 3).

12 ● An office with western exposure in a building having concrete floors is cooled only during office hours. If the windows are bare and unshaded, what is the maximum cooling load in the office due to solar radiation through the windows?

$215 \times 0.97 \times \frac{1}{2} = 104$ Btu per square foot of glass per hour (Fig. 1 and Table 4; note that only one-half of the heat need be considered because of absorption by the floor).

13 ● A 7 × 4 ft west window is equipped with an inside Venetian blind which is adjusted to fully cover the window when the sun shines. The net glass area is 75 per cent of the total area of the window. What is the cooling load due to the window at 10 a.m. and 4 p.m. assuming solar radiation as shown in Fig. 1 and

dry-bulb temperatures as follows: 10 a.m., outside 85 F and inside 77 F; 4 p.m., outside 95 F and inside 80 F?

10 a.m.: $H_t = 28 \times 1.13 (85 - 77) = 253$ Btu per hour (Equation 1, Chapter 8 and Table 13, Chapter 5).

4 p.m.: $28 \times 0.75 = 21$ sq ft net glass area.

$21 \times 215 \times 0.58 = 2619$ Btu per hour (Fig. 1 and Table 4).

14 ● What is the cooling load in a restaurant due to a $\frac{1}{4}$ hp refrigerating machine in an ice cream cabinet?

$4250 \times \frac{1}{4} = 1062.5$ Btu per hour (Table 5).

Chapter 9

CENTRAL AIR CONDITIONING SYSTEMS

Types of Systems, Dehumidifiers, Designing the System, Zoning, Location of Apparatus, Temperature of the Air Leaving Inlets, Air Quantity Required, Heat to be Removed by Cooling and Dehumidifying Apparatus, Size of Reheaters, Surface Cooling Problems, Auxiliary Equipment

CENTRAL systems, equipped for cooling and dehumidifying, are used principally in the air conditioning of theaters, restaurants, office buildings, or other places where people gather, and in manufacturing establishments where air conditions have an important influence on the quality of product or rate of production. A central cooling and dehumidifying plant is one in which the fans, dehumidifiers, and other related apparatus are assembled in suitable apparatus rooms from which supply and return ducts lead to the conditioned spaces. The design of such systems is considered in this chapter, while in Chapter 22 central systems for heating and humidifying are described. Air conditioning for industrial processes is considered in Chapter 40.

TYPES OF SYSTEMS

Dehumidification or cooling of air may be accomplished by several methods and by use of many heat transfer mediums. Most comfort-conditioning, central station, air-conditioning systems employ cold water or the direct expansion of a refrigerant in either spray type or surface type equipment to accomplish the required cooling and dehumidification. Among the several other methods that may be employed are: passing the air through or over a dehydrating agent and then lowering the dry-bulb temperature to the proper level, and evaporative cooling. With the former method the excess sensible heat may be removed with cold water and where this is not available, mechanical refrigeration must be used after the air has been dehydrated. The latter method is applicable to comfort conditioning only in regions where the summer wet-bulb temperature is low.

If the system is intended solely for summer conditioning, the apparatus will consist essentially of a dehumidifier of the surface type or spray type; filters; fan and motor; reheater; outside air, return air, and supply air duct work; air inlets and outlets; spray pump for spray dehumidifier; refrigeration equipment; and suitable controls. Generally, however, a central station air conditioning system is designed for year-round service. This means that properly sized heaters and dehumidifiers, with their respective

controls, must be added. With few exceptions, systems designed to meet summer capacity requirements will have ample capacity for winter and intermediate season conditioning.

A common arrangement of a central station spray type system for cooling and dehumidifying is illustrated in Fig. 1. The plant may be designed to condition 100 per cent outside air, 100 per cent return air, or a mixture of outside and return air. Further, part of the air returned from the conditioned space may be by-passed¹ around the conditioner as illustrated in Fig. 2. The reheater may be installed in the fan inlet chamber as shown, in the by-pass air duct, or in the fan discharge duct, depending upon apparatus space and other design conditions. Still another arrangement of equipment will result if the dehumidified air fan delivers the conditioned air to several other fans rather than to the con-

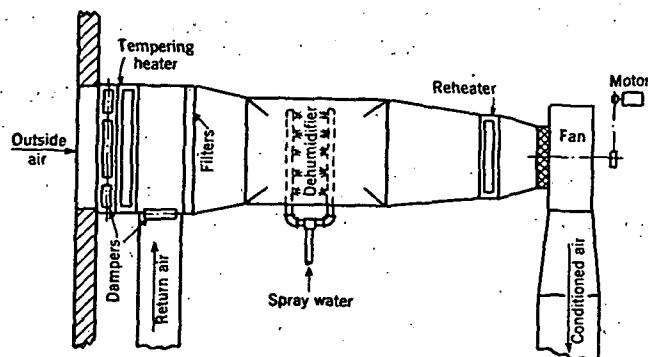


FIG. 1. SPRAY TYPE AIR CONDITIONING APPARATUS

ditioned space directly. These booster fan equipments may use part by-pass air as illustrated in Fig. 3 or 100 per cent dehumidified air and reheaters. The main apparatus, in either case, may or may not have a by-pass connection, depending on load conditions and other design factors.

The systems illustrated in Figs. 1 and 2 may be converted into the surface cooling type by merely replacing the dehumidifiers with surface cooling coils which use cold water or direct expansion of refrigerant to accomplish the required cooling and dehumidifying. The coils may also be installed within the spray chamber, either in series with the sprays or below them.

DEHUMIDIFIERS

Information on spray type dehumidifiers is given in Chapter 11.

Surface cooling type dehumidifiers generally consist of extended-surface coils within which the water or refrigerant is circulated or the refrigerant is expanded. The air to be cooled and dehumidified is drawn or blown over the coils. This system is generally comparatively low in initial cost and has low operating costs. Some localities have refrigeration codes

¹Patents exist covering the use of the by-pass for cooling and dehumidifying systems.

which restrict the use, in comfort conditioning applications, of refrigerants acting by direct expansion in coils exposed to the air stream. Therefore, local codes should be consulted by the designer before he plans a system employing direct-expansion methods. Close humidity control cannot be maintained during the cooling season by the surface cooling type of equipment. Winter humidification may be accomplished by use of evaporating pans or spray nozzles. The cooling coils serve no purpose during the intermediate or heating seasons, so in this respect the spray type equipment is often preferred, in that during certain seasons evaporative cooling will be sufficient to produce the cooling desired. Effective cooling and dehumidification accomplished by surface units are dependent upon many variable factors. The air velocity through the unit, air

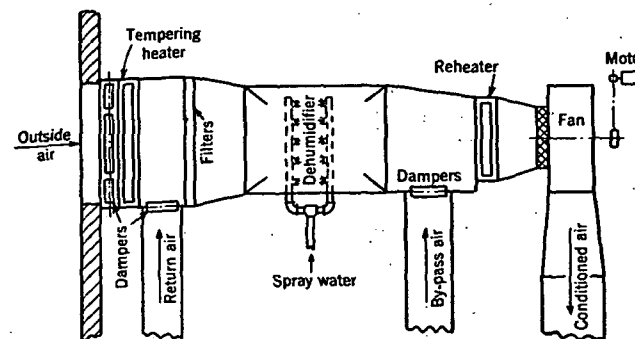


FIG. 2. SPRAY TYPE AIR CONDITIONING APPARATUS WITH BY-PASS

temperature, moisture content of the air, water or refrigerant temperature, and velocity of the water or refrigerant through the tubes must be considered in selecting the proper unit for a given design load. If any of these factors vary without a corresponding variation of the other factors, the effective work of the coil will increase or decrease, as the case may be.

DESIGNING THE SYSTEM

The general procedure for the design of a central cooling and dehumidifying system is as follows:

1. Calculate the heat gain for each room or space to be conditioned. (See Chapters 5 and 8.)
2. Establish the temperature of air leaving the supply inlets.
3. Calculate the quantity of air to be circulated.
4. Estimate the temperature loss in the duct system.
5. Determine the volume of outside air to be introduced. (See Chapter 3.)
6. Calculate the heat to be removed by the cooling and dehumidifying apparatus.
7. Calculate the size of the reheating equipment.
8. Select cooling equipment and heating equipment from manufacturers' data and performance curves.
9. Design the air distribution system and the air outlets and inlets. (See Chapters 19 and 20.)

10. Calculate the total static pressure of the system.
11. Select the fan, motor, and drive. (See Chapters 17 and 42.)
12. Select the pump and motor.
13. Design the control system. (See Chapter 14.)

ZONING

The above general outline of procedure will prove satisfactory for the smaller and less complex installations. However, when dealing with air-conditioning systems for large buildings, after a proper analysis has been made of the conditions to be maintained and the heat loads encountered, it is generally considered best practice to divide the complete job into a number of suitably sized units. In some cases a unit per floor or group of

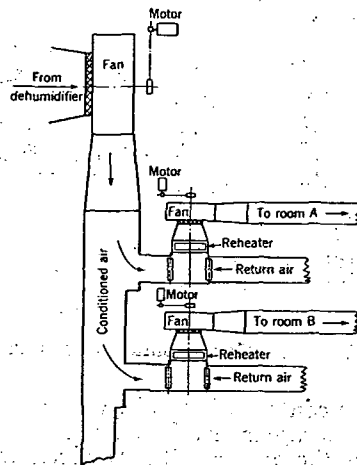


FIG. 3. CENTRAL DEHUMIDIFYING PLANT AND LOCAL RECIRCULATING FANS

floors may complete the design satisfactorily, whereas in others it may be advantageous to have separate units for each of the various outside exposures of the building. Where the floor area is large in relation to the outside wall exposure, it is obvious that provision must be made for the variable load to which the outside exposures are subjected. The heat loads on inside rooms are apt to be less variable since the fluctuations of the outside weather conditions are not directly involved. Such conditions often result in the natural zoning or segregation of rooms having similar exposures and internal heat loads. Variation in the hours of occupancy in different portions of a building also frequently require careful zoning for successful operation.

LOCATION OF APPARATUS

Availability of space for apparatus and duct work is of primary importance when selecting the type of system for a given design. In general,

for large installations, the refrigeration equipment, because of its size, weight, and operating characteristics, is located in the basement along with the boilers, fire pumps, and other equipment. The air conditioning apparatus is generally located where clean outdoor air is readily available, the designer bearing in mind that supply and return air ducts, steam connections, water and drain connections, and electrical connections must be made to the equipment proper.

TEMPERATURE OF AIR LEAVING INLETS

In comfort conditioning applications, air has been distributed from properly designed inlets without producing drafts at temperatures varying from approximately 5 to 30 deg below the required room temperature. Factors influencing the design and selection of air inlets are: ceiling height, contour of ceiling, length of blow, and temperature and quantity of air to be distributed. Most summer conditioning installations are designed to supply the air to the conditioned space at from 8 to 18 deg below room temperature. Recently the use of specially designed nozzles has indicated the possibility of reducing the air quantity necessary to dissipate a given heat load by introducing the air into the room as much as 30 deg below room temperature. Directional flow inlets which spread the air fanwise permit lower inlet temperatures than single direction inlets. Comfort conditioning systems employing differentials greater than 18 deg require special consideration and design experience because high pressure inlets or nozzles are usually used. Further, care must be taken to allow a sufficient air quantity under all load conditions to insure good distribution. If winter heating, as well as summer conditioning, is to be accomplished by the same distributing system, the design of the inlets will be influenced as discussed in Chapter 22. Industrial systems in which drafts are not objectionable usually employ a temperature differential equal to the dew-point depression.

AIR QUANTITY REQUIRED

For calculating the quantity of air required to absorb a given heat gain, the following approximate formulæ may be used:

$$M = \frac{H_s}{60 \times 0.24 \times (t - t_y)} = dQ \quad (1)$$

or, assuming a constant value of 0.075 lb for d ,

$$Q = \frac{H_s \times 55.2}{60 \times (t - t_y)} \quad (2)$$

where

Q = volume of air required, cubic feet per minute.

H_s = total sensible heat gain, Btu per hour.

t = room temperature, degrees Fahrenheit.

t_y = inlet temperature, degrees Fahrenheit.

M = weight of air required, pounds per minute.

d = density of air at the temperature and relative humidity of the room, pounds per cubic foot.

Example 1. The total sensible heat gain in a restaurant when held at 80 F is 199,736 Btu per hour. Assuming a 12 deg F temperature differential between the entering air and

the room temperatures, which is the same as assuming the dry-bulb temperature of the entering air to be 68 F, calculate the required air capacity of the system.

Solution.

$$Q = \frac{199,736 \times 55.2}{60 \times 12} = 15,313 \text{ cfm} = 1146 \text{ lb per minute.}$$

If a system similar to the one shown in Fig. 1 is used, 1146 lb per minute will be the capacity of the dehumidifier as well as of the fan equipment.

Example 2. If in addition to the 199,736 Btu per hour sensible heat load, the conditioned space has a moisture gain of 384,000 grains per hour, calculate the apparatus dew point required to give maintained conditions of 80 F dry-bulb and 65 F wet-bulb, with a corresponding 56½ F dew point.

Solution. With 384,000 grains of moisture per hour to be picked up, the entering dew-point temperature should be low enough so that the addition of this moisture will not increase the dew point above 56½ F.

Grains per pound of air saturated at 56½ F
(Table 5, Chapter 1)

68.0

Less: Grains per pound to be picked up,

$$\frac{384,000}{1146 \times 60}$$

5.6

Grains per pound allowable in entering air

62.4

This corresponds to an apparatus dew-point temperature of 54.17 F.

Example 3. Illustration of the by-pass system. (See Fig. 2.)

Assume the same data as for Example 2. Instead of passing all of the air through the dehumidifier for cooling and dehumidifying, a portion may be passed through and the balance be mixed with the conditioned air at the leaving end of the dehumidifier, the mixture being proportioned so that the resultant conditions will be those required to give proper conditions in the area considered.

Solution. The quantity of air to be dehumidified, the quantity to be by-passed, and the apparatus dew-point temperature may be approximately calculated as follows:

Let

X = percentage of air to be by-passed.

Y = percentage of air to be passed through the dehumidifier.

t_d = apparatus dew-point temperature, degrees Fahrenheit.

The quantity X of 80-F air must mix with the quantity Y of dehumidified air to produce air with a resultant 68 F dry-bulb temperature. Also, X quantity of air at 56½ F dew point must be mixed with Y quantity of dehumidified air to give a resultant dew-point temperature of the mixture of 54.17 F. It is assumed that the air passing through the dehumidifier is saturated.

Solving simultaneous equations,

$$80.0X + Yt_d = 68.00 \quad (3)$$

$$56.5X + Yt_d = 54.17 \quad (4)$$

$$23.5X + 0 = 13.83$$

$$X = \frac{13.83 \times 100}{23.5} = 59 \text{ per cent, air by-passed.}$$

$$Y = 100 - X = 41 \text{ per cent, air passed through washer.}$$

The second step is to determine the apparatus dew-point temperature. Substitute X in either Equation 3 or Equation 4, and solve for t_d :

$$80 \times 0.59 + t_d \times 0.41 = 68$$

$$t_d = \frac{68 - 47}{0.41} = 51.2 \text{ F, the apparatus dew point.}$$

HEAT TO BE REMOVED BY COOLING AND DEHUMIDIFYING APPARATUS

Example 4. Assume the same data as for Example 3. If the amount of outside air, at 95 F dry-bulb and 75 F wet-bulb, required for ventilation has been found to be 169 lb per minute, determine the refrigeration capacity required.

Solution. As the total weight of the air introduced per minute is 1146 lb, and 41 per cent of it goes through the dehumidifier, the total work to be done may be computed as follows:

Air passing through dehumidifier, 1146×0.41	470 lb
Less: Outside air for ventilation.....	169 lb
Return air.....	301 lb

The refrigeration required for the return air is:

Total heat per pound at 65 F.....	29.96 Btu
Less: Total heat per pound at 51.2 F.....	20.92 Btu
Requirement for cooling 1 lb of return air.....	9.04 Btu

301 lb $\times$ 9.04 Btu = 2721 Btu per minute required to cool the return air.

The refrigeration required for the outside air is:

Total heat per pound of outside air.....	38.46 Btu
Less: Total heat per pound at 51.2 F.....	20.92 Btu
Requirement to cool 1 lb of outside air.....	17.54 Btu

169 lb $\times$ 17.54 Btu = 2964 Btu per minute required to cool the outside air.

Thus, the total refrigeration required is:

2721 Btu + 2964 Btu = 5685 Btu per minute, which is equivalent to a load of 28.4 tons of refrigeration.

SIZE OF REHEATERS

A properly designed air-conditioning system will have reheaters of sufficient capacity to heat the conditioned air from the apparatus dew-point temperature to the inlet delivery temperature. If winter heating is to be accomplished, consult Chapter 22.

The following general formula may be used to determine the amount of heat necessary to reheat a given quantity of air:

$$H_1 = 0.24 (t_y - t_d) M \quad (5)$$

where

H_1 = heat to be supplied to reheater coil, Btu per hour.

Example 5. Assume the same data as for Example 1, and find the amount of reheating required.

Solution.

$$H_1 = 0.24 (68 - 54.17) 1146 \times 60 = 228,200 \text{ Btu per hour.}$$

SURFACE COOLING PROBLEM

The amount of coil surface required for a given amount of work is dependent upon factors previously listed. Obviously, the various types of

surfaces made available by different manufacturers will have different transmission values. It is recommended that the designer consult the latest manufacturers' catalogs because more accurate ratings are being issued from time to time.

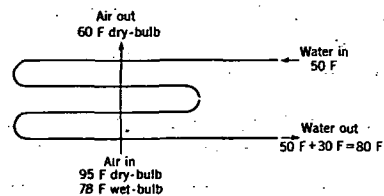


FIG. 4. COUNTER-FLOW SURFACE COOLING DIAGRAM.

Example 6. It is desired to cool and dehumidify 30,000 cfm of air at 95 F dry-bulb, 78 F wet-bulb, and 72 F dew point, to a 60 F dew point. Cooling water is available at 50 F in a quantity which will allow a 30 F rise in temperature to be used. The counter-flow surface cooling used is sketched in Fig. 4.

Solution. The pounds of partially saturated air cooled and dehumidified per hour equal 60 times the cubic feet of air at 95 F dry-bulb and 78 F wet-bulb brought past the coil surface per minute, multiplied by the pounds per cubic foot of the air as determined from Table 3, Chapter 1.

$$30,000 \times 60 \times 0.0708 = 127,440 \text{ lb per hour.}$$

The total heat H_t to be removed per hour by the surface coil is found to be equal to the pounds of partially saturated air passed over the coil per hour times the difference between the total heat of air at 78 F wet-bulb and at 60 F wet-bulb.

$$H_t = 127,440 (41.42 - 26.37) = 1,918,000 \text{ Btu per hour.}$$

The latent heat L to be removed per hour will be found by multiplying the pounds of partially saturated air passed over the coils per hour by the difference in the latent heat of the air per pound at the initial and final dew points.

$$L = 127,440 (1091.6 - 1086.2) = 688,000 \text{ Btu per hour.}$$

The sensible heat H_s to be removed per hour is equal to the total heat of the air less its latent heat.

$$H_s = H_t - L = 1,918,000 - 688,000 = 1,230,000 \text{ Btu per hour.}$$

Manufacturers' standard ratings for surface coolers are usually based on the cubic feet of air passed through their equipment per minute, reduced to the conditions of saturated air measured at a temperature of 70 F. In the present example, to convert the 127,440 lb of air cooled per hour to a basis which will permit the use of such standard ratings, it is necessary to multiply the pounds of air cooled per hour by the specific volume of the air, and to divide by 60.

$$\frac{127,440 \times 13.68}{60} = 29,100 \text{ cfm of 70 F saturated air.}$$

The amount of cooling water necessary when a 30 deg rise in its temperature is to be used is:

$$\frac{1,918,000}{30 \times 8.34 \times 60} = 128 \text{ gpm.}$$

With counter flow of air and water, it is necessary to determine the mean temperature difference between the air and the water in order to properly use the transmission coefficients given in apparatus rating tables.

$$\text{Mean temperature difference} = \frac{D_1 - D_2}{\log_e \frac{D_1}{D_2}} \quad (6)$$

where

D_1 = the difference between the temperatures of inlet air and outlet water, degrees Fahrenheit.

D_2 = the difference between the temperatures of outlet air and inlet water, degrees Fahrenheit.

$$\frac{(95 - 80) - (60 - 50)}{\log_e \frac{(95 - 80)}{(60 - 50)}} = 12.33 \text{ F.}$$

If from apparatus rating tables based on air velocities over the coils and water velocities through the coils, it has been found that the transmission coefficient is equal to 8.0 Btu per square foot per degree difference in mean temperature between the air and the water, the area of cooling coil surface necessary will be equal to the sensible heat divided by the transmission coefficient and also by the mean temperature difference.

$$\frac{1,230,000}{8.0 \times 12.33} = 12,450 \text{ square feet of cooling coil surface necessary.}$$

The latent heat is taken out at the same time the sensible heat is extracted, but no extra surface is required unless the latent heat exceeds approximately 40 per cent of the total heat. This is because the wetted surface has a much higher coefficient of transmission. Approximately 10 per cent more surface should be added if the latent heat exceeds 40 per cent of the total heat.

AUXILIARY EQUIPMENT

Consult Chapters 14, 17, 19, 20, 22, and 42 for information on the air distribution system; air outlets and inlets; static pressure on fan; fans; motor and drive; and the control system.

PROBLEMS IN PRACTICE

1 • What is meant by the term evaporative cooling?

Evaporative cooling, or adiabatic saturation of the air, is only effective when the air to be cooled is very dry. It is accomplished by passing the air in an unsaturated condition through a water spray which evaporates a part of the water at the expense of the sensible heat. In this adiabatic transfer the total heat content of the air remains constant while the dew point rises and the dry-bulb falls until the air is saturated.

2 • In summer air conditioning what factors control the difference between the dry-bulb temperature of the conditioned space and the dry-bulb temperature of the entering air?

1. The duct and supply grille arrangement permitted by architectural and structural requirements for the particular space, e.g., ceiling height and obstructions on ceilings, such as beams.
2. The state of activity of the occupants.
3. The velocity at the inlet grille, as limited by noise level requirements.

4. The direction of the jet relative to the occupants.
5. In some cases, the temperature of the available water supply, which may have some bearing on the air delivery temperature.

3 • What factors determine the volume of conditioned air which must be delivered to the space?

The sensible heat to be removed, and the allowable temperature differential.

4 • What factors determine the dew point of the air entering the space?

The maximum dew point desired in the conditioned space, and the moisture gain in the space per unit weight of air supplied.

5 • Why must the air leaving a dehumidifying type air washer be reheated before delivery?

The air leaves the dehumidifying air washer saturated at a relatively low temperature which in most cases is lower than the allowable delivery dry-bulb temperature. Also, the air may possibly be carrying a small amount of entrained water which might settle out in the ducts near the washer and cause corrosion difficulties.

6 • What methods are used for reheating air?

1. Passing it over reheating coils.
2. Mixing it with by-passed air at a higher temperature.

7 • What determines the final temperature of the spray water in a dehumidifier?

Because of the effectiveness of the heat transfer between air and finely divided spray water in a well designed dehumidifier, the air will be cooled to within 1 or 2 F of the final water temperature, provided the air velocity through the washer does not exceed 600 fpm. This final temperature should then be taken as 1 or 2 F lower than the required dew point of the air leaving the washer.

8 • What are the advantages of using counter flow of air and water in surface coolers?

Counter flow results in a higher mean temperature difference than does parallel flow for the same range of air and water temperatures, which means that less cooling surface is required. Counter flow permits higher initial water temperatures and also allows a greater temperature rise for the water. These factors combine to reduce the cost of circulating and refrigerating the cooling water.

9 • What factors other than cost should be considered in determining whether to use a central system or another type?

- a. *Appearance:* The equipment must be designed to harmonize with the architecture of the building.
- b. *Distribution:* The system must maintain adequate and uniform air motion over the entire conditioned space.
- c. *Control:* The control system must be designed to give effective partial load operation.

10 • Can the central cooling and dehumidifying system be used as an all-year-round conditioner?

By modifying the control system and adding blast coils or a water heater to the spray type system, the cooling system will function as one for heating and humidifying. The surface cooling type may be transformed by modifying the control, and adding another set of coils and a humidifier.

11 • Will the tons of refrigeration-effect per day be the value calculated in Example 4 of this chapter times the hours of operation?

No. The tons of refrigeration-effect are functions of the load. The components of the load vary, that is, the number of people occupying the space, the outdoor conditions, and the solar radiation will change from hour to hour and from day to day. The calculated load represents the maximum required for design peak conditions.

Chapter 10

COOLING METHODS

Methods of Cooling Air, Evaporative Cooling, Dehumidification, Silica Gel System, Alumina System, Lithium Chloride System, Design of System, Operating Methods, Steam Jet System, Centrifugal Compression Systems, Compressors, Refrigerants, Methods of Cooling, Condensers

BY using any of the following four methods, or any combination of them, *effective temperature* (see Chapter 3) may be reduced.

- a. *Sensible cooling:* Lowering of the dry-bulb temperature by the removal of sensible heat without change of the dew-point temperature.
- b. *Dehumidifying:* Lowering of the dew-point temperature by the removal of moisture without change of the dry-bulb temperature.
- c. *Evaporative cooling:* Lowering of the dry-bulb temperature through the evaporation of moisture without the addition or the subtraction of heat.
- d. *Air motion:* Increasing the air motion over the body.

As an example, let the condition be considered of 92 F dry-bulb, with a 40 per cent relative humidity, corresponding to a wet-bulb temperature of 72.8 F, and an effective temperature for still air of 81.1 F. This *effective temperature* may be reduced 3.1 F by any of the four basic methods mentioned, as follows:

First, by lowering the dry-bulb temperature to 85.5 F without changing the dew-point of 64.2; this gives an effective temperature of 78 F.

Second, by reducing the moisture content of the air to 46 grains per pound of dry air without changing the dry-bulb temperature; this gives an effective temperature of 78 F.

Third, by reducing the dry-bulb temperature to 83.8 F without changing the total heat of the air. This requires the evaporation of 14 grains of moisture per pound of dry air, and the effective temperature will become 78 F.

Fourth, by increasing the air movement from still air to 460 fpm, a velocity which will reduce the effective temperature 3.1 F from 81.1 F to 78 F.

Method to Employ

The best method of reducing the effective temperature in any specific case will depend on the accompanying circumstances and can be determined only by a thorough analysis made by a competent engineer. Generally speaking, the removal from the air of the sensible heat, or moisture, or both, by sensible cooling or dehumidifying is the most satisfactory method. Adequate results by the utilization of air motion or by evaporative cooling are difficult to obtain because of the dependence of both methods upon climatic conditions beyond the engineers' control although these methods are much less expensive than the first two

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURES^a

STATE	CITY	TEMP. F	STATE	CITY	TEMP. F
Ala.	Birmingham	84	Mass.	Boston	80
	Mobile	73		Cambridge	70
Ariz.	Phoenix	81		Fall River	76
	Tucson	80		Lowell	50
Calif.	Anaheim	60		Lynn	68
	Berkeley	69		New Bedford	70
	Fresno	72		Salem	68
	Fullerton	75		Worcester	76
	Glendale	68	Mich.	Detroit	77
	Los Angeles	75		Flint	70
	Oakland	69		Grand Rapids	84
	Ontario	70		Highland Park	77
	Pasadena	82		Kalamazoo	53
	Pomona	75		Lansing	64
	Riverside	78		Jackson	56
	Sacramento	72		Saginaw	82
	San Bernardino	65	Minn.	Duluth	55
	San Diego	82		Minneapolis	80
	San Francisco	62		St. Paul	77
	Whittier	75	Mo.	Jefferson City	82
Colo.	Denver	75		Kansas City	84
Conn.	Bridgeport	66		St. Joseph	84
	Hartford	73		St. Louis	85
	New Haven	76		Springfield	70
	Waterbury	72	Nebr.	Lincoln	87
D. C.	Washington	84		Omaha	87
Del.	Wilmington	83	Nev.	Reno	61
Fla.	Jacksonville	80	N. H.	Manchester	76
	Miami	80	N. J.	Jersey City	63
	Tampa	77		Newark	74
Ga.	Atlanta	87		Paterson	78
	Macon	80		Trenton	79
Ill.	Chicago	76	N. Y.	Albany	68
	Cicero	76		Buffalo	75
	Evanston	73		Jamaica	56
	Peoria	67		Mt. Vernon	74
	Rockford	59		New Rochelle	75
	Springfield	82		New York	72
Ind.	Evansville	86		Rochester	70
	Gary	75		Schenectady	60
	Indianapolis	80		Syracuse	74
	South Bend	61		Utica	69
	Terre Haute	82		Yonkers	70
Iowa	Cedar Rapids	78	N. C.	Asheville	74
	Des Moines	77		Charlotte	85
	Sioux City	62		Winston-Salem	82
Kans.	Concordia	57	N. M.	Albuquerque	65
	Kansas City	86	Ohio	Akron	76
	Topeka	88		Canton	50
	Wichita	72		Cincinnati	84
Ky.	Louisville	85		Cleveland	74
La.	Baton Rouge	85		Columbus	82
	New Orleans	85		Dayton	60
Me.	Augusta	60		Lakewood	82
Md.	Baltimore	67		Springfield	72
				Toledo	83

^aThese averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

TABLE 1. AVERAGE MAXIMUM WATER MAIN TEMPERATURE^a (CONTINUED)

STATE	CITY	TEMP. F	STATE	CITY	TEMP. F	
Okla.	Oklahoma City	82	Utah	Logan	44	
	Tulsa	85		Salt Lake City	60	
Oreg.	Eugene	60	Va.	Fredericksburg	75	
	Portland	64		Lynchburg	73	
Pa.	Altoona	74		Norfolk	80	
	Erie	75	Wash.	Olympia	58	
	Johnstown	74		Seattle	62	
	McKeesport	82		Spokane	51	
	Philadelphia	83		Tacoma	57	
	Pittsburgh	67	W. Va.	Charleston	85	
R. I.	Providence	68		Huntington	78	
S. C.	Charleston	80		Wheeling	78	
	Greenville	81	Wis.	La Crosse	54	
	Spartanburg	78		Madison	58	
S. Dak.	Rapid City	55		Milwaukee	70	
Tenn.	Chattanooga	84		Racine	68	
	Knoxville	89				
	Memphis	70				
	Nashville	90				
Texas	Amarillo	65	PROVINCE			
	Austin	90				
	Beaumont	86	Alta.	Calgary	64	
	Dallas	86	B. C.	Vancouver	60	
	Fort Worth	84	Ont.	London	50	
	Galveston	90		Toronto	63	
	Houston	84	P. E. I.	Charlottetown	48	
	Port Arthur	83	Que.	Montreal	78	
	San Antonio	76		Quebec	68	
	Wichita Falls	85				

^aThese averages taken from various city water main locations, with some actual values slightly higher and some lower than values shown.

mentioned. Cooling by evaporation is satisfactory only when the air to be cooled is very dry; air motion as a means of producing cooling effect is never entirely adequate in the range of high temperatures. Of the two, evaporative cooling, or adiabatic saturation of the air, is a much more dependable method which will make more reduction in the effective temperature than will an increasing air motion within permissible limits.

As an example of this, consider an outdoor condition of 96 F dry-bulb and 80 F wet-bulb. The effective temperature is 85.7 F and, if the still air is moved with a velocity of 300 fpm, the effective temperature will be reduced only 2.0 F while saturation at the wet-bulb temperature would reduce the effective temperature 5.7 F. At 300 fpm velocity this saturated air would reduce the effective temperature to 75.6 F, thus making a total improvement of 10.1 F.

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 this country and Canada.

Evaporative Cooling

Evaporative cooling is accomplished by passing air through a water

spray in which the water is being continually recirculated. The air, entering in an unsaturated condition, evaporates a part of the water at the expense of the sensible heat. As this is an adiabatic transfer, the total heat content of the air remains constant, while the dew point rises and the dry-bulb falls until the air is saturated. A system¹ of ducts and a propelling fan are used to distribute the air in a proper manner.

It will be seen that the reduction in dry-bulb temperature is a direct function of the wet-bulb depression of the air entering the dehumidifier and that the resulting air temperature is governed entirely by the entering wet-bulb temperature of the outside air.

Dehumidification

Dehumidification may be accomplished in three ways:

1. By cooling the air below the dew point and causing a part of the moisture contained to precipitate.
2. By extracting all or part of the moisture by absorption.
3. By extracting all or part of the moisture by adsorption.

As used in this discussion, the term *adsorption* pertains to the action of a substance in condensing a gas or vapor and holding the condensate on its surface without any change in the chemical or physical structure of the substance and with the release of sensible heat. The term, *absorption*, implies a change in the chemical or physical structure of a substance in the process of dehydrating air. Adsorption is distinctly a surface action and is thereby distinct from absorption. Adsorbers include lithium chloride, calcium chloride, silica gel, lamisilite, or any of the halides; absorbers include sulphuric acid.

Dehumidification by Refrigeration

Air conditioning imposes requirements on refrigeration equipment not usually found in general cooling work, so that specially designed apparatus is often needed to replace that normally used for industrial cooling. Standard equipment can be adapted to meet air conditioning requirements but extreme care must be taken to determine the limits of its applicability.

In industrial or process cooling systems the load is fairly constant, noise in operation is not of paramount importance, space is available or relatively cheap, condenser water is not a source of worry, and the cooling system is to a great extent separate and independent of other mechanical equipment. By contrast, air conditioning, especially as used for space cooling and comfort work in office buildings, theaters, and places where people gather requires special consideration of all these factors. Space in public buildings is limited and condenser water is expensive. Noise interferes with the occupants, and the cooling equipment must dovetail with the other air-handling apparatus. Most important, the load fluctuates tremendously and is seasonal.

A complete discussion of the thermodynamic problems of refrigeration is given in Chapter 2 on Refrigeration.

¹See Air Washer Performance in Chapter 11; also Theory of Atmospheric Cooling in same chapter.

Silica Gel System

Silica gel is a chemical composition made from sodium silicate and acid, the chemical formula being SiO_2 . It has an appearance greatly resembling that of clear quartz sand but it differs in structure in that the crystals are highly porous, with voids constituting 41 per cent by volume although the pores are microscopic in size. This material possesses the property of the pores are able to adsorb a substantial portion (about 25 per cent of its own weight) of moisture from the air without any increase in its volume. After the silica gel has become "saturated" or has adsorbed moisture to the limit of its capacity, the moisture may be driven from it by the application of heat, again without change in the structure, volume, or chemical composition of the silica gel. This cycle may be repeated indefinitely. When applied to air conditioning the silica gel which is exposed to the air reduces the moisture content in the air and releases sensible heat which may be readily removed from the air. A typical diagram of this system is shown in Fig. 10, of Chapter 2 on Refrigeration.

Practical Application of Silica Gel

Silica gel has two applications when used to replace refrigeration. In the one principally used, the air from which moisture is to be extracted is taken through silica gel beds by suction or pressure fans, and by means of this process the moisture becomes adsorbed by the silica gel and the air leaves at a lower dew point and a higher sensible temperature than those at which it entered. If this air is passed over surface coolers in which tap water or another cooling medium is flowing through tubes, a certain amount of sensible heat will be removed. The air leaves the surface cooler or interchanger with the same dew point with which it emerged from the silica gel beds, but with a lower dry-bulb temperature, although the dry-bulb temperature may be higher than the temperature of the air entering the silica gel beds.

In another method, the first two of the steps outlined are duplicated, and in addition the air is carried through a spray type washer. Because the air enters the washer with a low wet-bulb, and because adiabatic saturation will take place at a temperature close to the entering wet-bulb, considerable cooling of the air can be accomplished; but this can be done only with a consequent increase of the dew point.

It is necessary to reactivate the silica gel after it has adsorbed about 25 per cent of its own weight in the form of moisture. As reactivation requires a high temperature and since silica gel is only active at low temperatures, cooling of the beds must also be completed before they can be used again. This necessitates three stages in the silica gel containers and requires either three beds of silica gel or one bed divided and automatically put in position. The reactivation is usually done by means of gas or oil fires and the cooling of the beds by means of indirect water cooling or by means of small quantities of dehydrated air taken from the system beyond the interchanger.

Alumina System of Adsorption

Activated alumina contains a trifle over 91 per cent of aluminum oxide, Al_2O_3 , which material will adsorb nearly 100 per cent of the vapor in the

air up to about 8 or 10 per cent of the weight of the adsorbing material, after which the adsorption falls off gradually as the saturation point is approached. The application is quite similar to that employed for silica gel; that is, the material is exposed to the air flow and after reaching about 75 per cent saturation is reactivated by removing the moisture adsorbed by means of applied heat. The actual scheme generally followed in the use of this material for continuous service varies somewhat from silica gel inasmuch as the material is placed in three units which are used consecutively for the different steps. These steps permit each unit to operate as follows:

- In series with the preceding unit.
- Alone.
- In series with the following unit.

This plan allows for adsorption, reactivation, and cooling, in a manner similar to that used with silica gel.

Taking a single unit, when it is in the *a* step and operating with the preceding unit, the alumina adsorbs approximately 25 per cent of the moisture in the air and takes up about 1.3 per cent of its weight of water. During the second step when it is operating alone, it takes up 100 per cent of the moisture in the air until the weight of the water adsorbed is brought up to about 6.7 per cent. During the third step when the unit is operating with the succeeding unit, it extracts about 75 per cent of the moisture in the air until the water weight adsorbed comes up to about 10 per cent of the weight of the adsorber. The time allowable for reactivating is equal to the time occupied by the second unit adsorbing alone, plus the time when the second and third units are adsorbing in series, plus the time when the third unit is adsorbing alone, at the expiration of which time the first unit will be again required.

The temperature of air used for alumina reactivation is usually between 300 and 700 F and the air flow rate will have to be higher with the low temperature air than it will be with reactivating air of higher temperature. For example, air at 400 F for reactivating will, at 10 cu ft per hour per pound of alumina, require about 6 hours for reactivation. In the three unit system, after reactivation the cooling of the activated alumina may be carried out with considerable rapidity by using dry air from the adsorption unit for circulation through the unit which has just completed reactivation. The final temperature of the unit before it goes back into service should be not over 200 F. As a basis for computing the amount of cooling air required for reactivation, each cubic foot of cooling air has been found capable of removing 2.2 Btu when heated from 85 to 200 F and of providing a sufficient margin of safety in operation.

Lithium Chloride Adsorption System

Practically all salt solutions have the property of adsorbing or condensing moisture from a gas. The amount depends on the character of the solution, its gravity, temperature and viscosity. One of the best known dehydraters, calcium chloride, for instance, has been long known for this property. Its characteristics, however, present a limiting factor of about 30 per cent relative humidity, so that the resultant dew point of air brought in intimate contact with calcium chloride is usually too high for comfort work without further refrigeration.

The cycle of solid dehydraters or adsorbers and liquid adsorbers are fundamentally the same. A diagram of a lithium chloride adsorption system is shown in Fig. 1.

The liquid adsorber is brought in contact with air having a certain vapor pressure due to the moisture of water vapor that is in it, and the chloride, either sodium, lithium, calcium, or whatever is used, having a lower vapor pressure, adsorbs moisture in the form of water from the water vapor that is in the contacting air. Thermodynamically, it is definitely possible to measure this change and to know definitely that a change of state takes place because of the fact that there is a definite rise in temperature in the liquid adsorbing the moisture, which is definitely a function of the amount of water vapor condensed from the air stream.

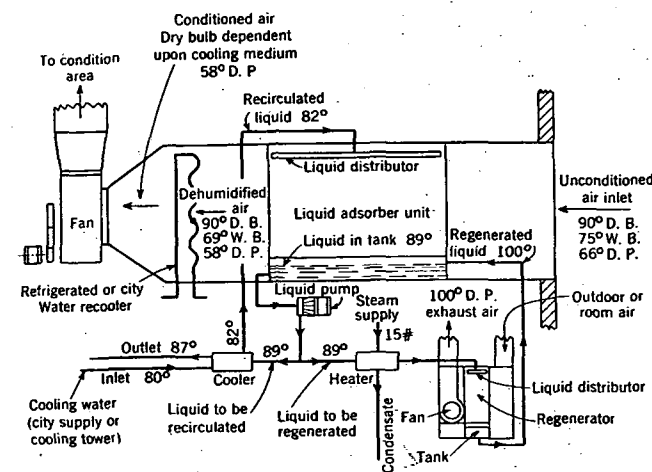


FIG. 1. DIAGRAM OF LITHIUM CHLORIDE ADSORPTION SYSTEM

Of course, the liquid adsorber, contrary to the solid adsorber such as silica gel, takes on two functions: one, as an adsorber, and that adsorption, as has been stated, tends to create a rise in temperature; on the other hand, the mass of liquid in the adsorber tends to keep down the rise in temperature, so that by cooling the adsorber below a certain temperature, a definite temperature can be maintained in both the liquid and the air, which is a function of the heat balance between the total heat of the air and the total heat of the liquid.

Adsorption of moisture by a liquid adsorber weakens the concentration of the liquid so that its adsorbing capacity is reduced and regeneration, or the driving off of the excess moisture in the liquid, must be performed, just as the driving off of the liquid adsorbed by a solid adsorber must be accomplished. However, with a liquid adsorber a very definite condition can be maintained; that is, a condition of density, by continuously withdrawing a certain small portion of the total liquid for intensive concentration and the concentrated liquid added to the mass of the liquid continuously, without varying to any marked degree the vapor pressure

of the total mass. This method of continuous regeneration and re-concentration tends to hold the relative humidity of the leaving air always at a very definite point.

There are two methods of regeneration. One is the boiling of the excess moisture by raising the temperature of the adsorbing liquid to above the boiling point of the particular concentration. As the salt in the solution does not vaporize, it is not carried off in the boiling process. The only care required is that in the small amounts of liquid diverted to the regenerator for concentration, too much moisture is not driven off, occasioning freezing or solidification of the salts, or solids.

The second method of regeneration which lends itself particularly well to ordinary low pressure heating systems, where steam pressure of about 12 lb per sq in., corresponding to a temperature of 242 F, is to raise the temperature of solution with ordinary steam coil interchangers to about 225 F, and then to pass the solution at this temperature over various types of scrubbers, over which ordinary air is passed. The increase in temperature of the liquid adsorber tends to increase its vapor pressure to such an extent that there is an exchange of vapor between the liquid adsorber and the air, as well as an equalization of temperature between the air and the liquid adsorber, so that the air is capable of taking up part of the moisture from the liquid adsorber to increase its density and to carry this excess moisture out into the atmosphere with the leaving air.

After this vaporation has taken place, the highly concentrated, hot liquid adsorber is taken through an interchanger through which the water used in cooling the main solution can be re-used to reduce the temperature of the concentrated solution to a point where it may be thrown back into the main solution tank at a slightly higher temperature than the main body of the solution. As approximately only 10 per cent of the solution is regenerated continuously, a difference of 5 to 10 F in the concentrating solution only creates a rise of 1 F or less in the main body of the adsorber, so that its effect on increasing the temperature of the air to be dried is negligible.

As can be seen, there are two conditions of continuous operation which have a tendency to raise the temperature of the liquid. One is the adsorption of vapor from the air, which, through a change of state, changes the latent heat of the vapor adsorbed to sensible heat, to raise the temperature of the liquid and consequently, the temperature of the air; secondly, the necessary heat added to the regenerator liquid in order to reevaporate and carry off the excess moisture which has been condensed in the first stage.

In the regeneration process, the air currents in this method have a tendency to carry away the excess heat of evaporation. In the first stage, that is, the main stage of adsorption, the temperature of the main body of liquid is continuously cooled, and the excess heat carried away by some type of interchanger, through one part of which either cooling tower or city water is utilized. Of course, this water may be refrigerated, but under most conditions that is unessential. With the ordinary liquid interchanger, using cooling tower water, the maintained condition of the liquid and the air can be kept at a point very close to the prevailing wet-bulb, possibly within 5 F, and certainly within 10 F.

Design of System

When designing air conditioning systems, the capacity of equipment is decided by selecting apparatus of sufficient size to maintain predetermined temperatures and humidities in treated spaces when arbitrarily established maximum atmospheric temperatures occur coincident with given conditions of population, lighting, and power consumption. These factors determine the maximum duty of the cooling system. The duty does not necessarily determine the size or capacity of the refrigeration apparatus. The refrigerating capacity is expressed in *tons*, each ton being equal to the absorption of the heat given up by one ton of ice at 32 F melting to water at 32 F in 24 hours. This is equivalent to heat absorption at a rate of approximately 200 Btu per minute, or 12,000 Btu per hour.

After the maximum duty is determined, the other factors concerning the installation must be investigated. The total heat to be removed by the cooling system has many sources, some substantially constant and others extremely variable. These sources can be roughly classified as follows, the first column indicating the order in amount and the second the order in variability:

- | | |
|--|--|
| 1. Fresh air supplied. | 1. Fresh air supplied. |
| 2. Population. | 2. Transmission through the structure. |
| 3. Transmission through the structure. | 3. Light and power consumed. |
| 4. Light and power consumed. | 4. Population. |

By combining these two columns, a third grouping is obtained as follows:

- | | |
|--|------------------------------|
| 1. Fresh air supplied. | 3. Population. |
| 2. Transmission through the structure. | 4. Light and power consumed. |

In this last arrangement, the first two items are governed by atmospheric conditions and they are therefore subject to tremendous fluctuations in value. As they generally form 40 to 60 per cent of the entire maximum load, the duty of the cooling system will be much less than maximum most of the time.

The transmission through the structure is especially influenced by the sun. (See Chapter 8.) In many cases, because of the heat flow resistance of the structure, the heat from the sun is retarded until it is compensated for by a reduced general temperature out-of-doors.

A survey of Weather Bureau records indicates that maximum temperatures occur less than 5 per cent of the cooling period and also that the duration of peak conditions is never more than three or four hours.

Two factors control the size of the refrigeration system, the evaporator or suction temperature, and the condenser or head temperature. With the knowledge that the system will operate most of the time with a load of not over 60 per cent of maximum, and that maximum demands will occur infrequently and only for short periods, some provision must be made to insure economical operation under average conditions. This can be done by overloading the machine under extreme demands and basing the design on normal or average loads. Flexibility in arrangement can be provided in several ways.

Variations in load change the efficiency of any machine and a refrigerating system can be costly and inefficient if improperly designed or operated.

Fortunately, the trouble can be concentrated in the compressor and the problem relieved of many complications. It is comparatively easy to furnish condensers and evaporators to carry the maximum load so arranged that they will function properly at small demands. They affect the compressor performance to some extent but most of the compressor problems are in the machine itself.

Variations in load are usually effected by lowering the suction temperature and pumping a larger volume of gas per ton through a greater pressure range. This is possible because the latent heat of the refrigerant remains nearly constant throughout the small range used and the specific volume varies rapidly with change in pressure. As the compressor must remove the refrigerant evaporated, the evaporator temperature fixes the displacement required. The objection to such method is that the total power consumed remains nearly constant and the power per unit of cooling increases rapidly as the total output is reduced. Such operation is satisfactory as long as the load is kept within 10 per cent of the rating of the compressor but this condition does not commonly occur in air conditioning applications.

Operating Methods

It is possible to divide the entire refrigeration system into a number of small units, which will allow cutting in and out of compressors and condensers as the load fluctuates.

A second method of providing for economy of operation is to have storage capacity which can be utilized during the peak period. A further reference to the Weather Bureau records indicates that maximum conditions prevail during the day for not more than three hours, and consequently the refrigerating system can be run for a longer period at maximum efficiency with tanks to store cold water or brine for supplementing the actual output of the refrigerating equipment when the load is more than the machine will carry. This situation brings complications. Storage tanks require space and extra apparatus, which increase the cost of the entire system, and further, it is difficult to determine what the size of the compressor should be because of the other variables which enter the problem. Depending upon the availability of storage space, the compressor could be designed for any reasonable percentage of the maximum load, so the smaller the compressor, the larger the storage space, and vice versa.

A third method is to provide in the compressor itself some means of reducing the capacity. This can be done by varying the speed and consequently the displacement of the compressor, or by varying the displacement, either by a partial by-pass of the cylinder or by a clearance pocket in the head of the cylinder when reciprocating compressors are used. It might be assumed that the efficiency would remain practically constant. This is not correct, inasmuch as the machine friction remains constant with the by-pass or clearance pocket method and this raises the power required per ton of refrigeration developed. Also, the volumetric efficiency of the machine falls off rather rapidly when the clearance pocket or partial by-pass is used. By varying the speed of the compressor, the efficiency of the power unit falls off as the speed is reduced, while the compressor friction remains constant.

Another method of operation is the automatic starting and stopping of the refrigerating machine, with the automatic control designed to function as the load varies.

All of the methods described are used from time to time.

The methods of varying the output of a refrigeration system which have been outlined apply to the reciprocating type of compressor, although variations in the speed of the compressor to change the refrigerating output are common to all types of mechanical refrigeration.

There is a further method of controlling the compressor output which is particularly adaptable to the centrifugal type of machine. This is accomplished by varying the amount of condensing water used with the fluctuation in demand load. Because of the characteristics of the centrifugal type of apparatus, as the condensing water quantity is reduced and the condensing temperature consequently raised, the discharge pressure of the centrifugal machine rises correspondingly and the horsepower input to the machine falls off. While this reduces the total power input to the machine, it does not necessarily reduce the power input per ton of refrigeration developed, as the power input does not drop with a rising discharge pressure as fast as the refrigerating effect produced drops. It is a method, however, which shows marked economies over the method generally used by the operating engineer, which is to lower the suction pressure in order to reduce the refrigerating output of the system.

Steam Jet System

So far the discussion has been confined to reciprocating, centrifugal, and rotary compressors. The steam jet type of compressor, under certain circumstances, is desirable for use in air conditioning. A complete flow diagram of the system is shown in Fig. 4 of Chapter 2 on Refrigeration. The power used for compressing the refrigerant is steam, taken directly from the boiler, thus eliminating the mechanical losses of manufacturing electric current. As the compression ratio between the evaporator and condenser under normal circumstances is large, the mechanical efficiencies of the equipment are somewhat lower than those of the positive mechanical type of compressor; also the condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures, and steam ejectors of the type commonly used in power plants for various processes will produce the necessary low absolute pressure to cause evaporation of the water.

In Chapter 2 on Refrigeration Fig. 4 shows a typical water cooling application. 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, and 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 into the evaporator, plus any entrained air taken out of the water circulated, to a somewhat higher absolute pressure, and the vapor and air mix with the impelling steam on the discharge side of the jet. The total mixture of entrained air, evaporated water, and impelling steam is discharged into a surface condenser at a pressure which permits the available condensing medium to condense it. The resulting condensate is removed from the condenser by a small pump, from which it can be discharged to the sewer or returned to the system in the form of make-up water, or part of it may be returned to the boiler feed pump.

As the normal temperature of water required for air conditioning purposes is between 40 F and 50 F, with an average temperature of approximately 45 F, this type of water cooling is particularly desirable, as the efficiencies and operating costs compare very favorably with other types of refrigerating equipment, especially in view of the fact that the cooling apparatus is, as a general rule, less expensive to install.

Approximately three times as much condenser water is required for the steam jet cooling system as would be necessary with other types of mechanical refrigeration, but as the system can be designed with a large number of jets, each of which can be cut off as the load falls below maximum, constant refrigerating efficiency is maintained and frictional losses and volumetric inefficiencies are kept at a minimum.

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 small secondary condenser, of course, is necessary to condense the steam used in the secondary jet.

Steam jet refrigeration has an advantage where cooling towers are used for supplying the condensing liquid, as there is a great saving in the amount of steam used per ton of refrigeration. As the outdoor weather conditions vary the load on the cooling system, the compression ratio between the condenser and evaporator can be reduced and less propelling steam need be used per ton of refrigeration developed. Roughly, in air conditioning work, mechanical compressors show a falling off of 30 to 40 per cent in the power input when using the most economical arrangement of compressors, as the load varies from 100 per cent to 25 per cent of the rated capacity; whereas with steam jet cooling equipment, the amount of steam required for producing the necessary refrigerating effect falls off in direct proportion to the load on the system. When steam refrigeration is employed with cooling towers, the efficiency increases as the output is reduced.

Compressors and Refrigerants

There are many different types of compressors, a number of refrigerants, different types of evaporators, condensers and arrangements of the cycle, and each type has its particular place and usage.

The generally used compressors are of the following types:

1. Reciprocating compressors.
2. Centrifugal compressors.
3. Rotary compressors.
4. Steam jet compressors.

Over-all efficiency of the compressor in smaller commercial installations is not as important a requirement as that the whole unit require little attention and make a minimum of noise. The noise level when the fan, sprays, and compressor are in full operation should not exceed 25 decibels. High compressor efficiency appears as an important factor only in the larger industrial air conditioning systems.

The refrigerants in most general use in commercial and industrial air conditioning are here listed in the order of their inoffensive odor characteristics:

1. Water vapor.
2. Carbon dioxide.
3. Dichlorodifluoromethane.
4. Dichloromethane, sometimes called methylene chloride.
5. Methyl chloride.
6. Ammonia.
7. Sulphur dioxide.

The various types of compressors bear varied relationships to the refrigerants used in both commercial and industrial air conditioning. *Reciprocating compressors* are generally used for any of the refrigerants listed except water vapor, dichloromethane, or other low pressure refrigerant, and they are used in both commercial and domestic air conditioning systems. They have been developed to a point where their efficiency is high and their operation very satisfactory. Relatively low speed operation makes them desirable for general use in large installations. They are of two types, vertical and horizontal, either single or double acting. The horizontal double-acting compressor is not generally used in air conditioning except when carbon dioxide is used as the refrigerant in the larger industrial systems. Vertical, single-acting, encased crank, reciprocating compressors of the uniflow type with valves in the pistons have proven reliable and are used in capacities from 1 hp to more than 100 hp. Reciprocating compressors can be used with more refrigerants than other types of compression units. For instance, when carbon dioxide is used as the refrigerant, a reciprocating compressor is required because of the extremely high pressures and the relatively high ratio of compression.

The production of refrigeration at temperature levels from 25 F to 55 F for general air conditioning involves special types of refrigerating compressors. Among these are:

1. Centrifugal compressors using a volatile refrigerant.
2. Centrifugal compressors using water as a refrigerant.
3. Steam jet or vacuum systems using water as a refrigerant.
4. Rotary compressors using a volatile refrigerant.

The first two types, *centrifugal compressors*, using dichloromethane or water vapor, can theoretically be used with any of the other refrigerants, but the resulting loss in efficiency with the higher pressure gases limits the centrifugal compressor to the two refrigerants named. At the present time the centrifugal compressors are limited to air conditioning systems of a minimum of 50 tons and more. Centrifugal compressors are usually built in two or more stages where the compression ratio is high, and their

design follows closely that of any other centrifugal equipment, such as general service pumps and fans.

Steam jet compressors which have recently entered the field are simple and compact and, having no moving parts, they produce practically no vibration but are not economical for water temperatures much below 40 F or where the cost of generating steam is higher than the cost of operation with other prime movers.

Rotary compressors are generally used for methyl chloride and dichlorodifluoromethane because of their relatively low pressure and compression ratios. These compressors find widest use for fractional tonnage duty.

The source of condensing water to some extent governs the type of refrigerant used. If condensing water is available at temperatures of not more than 70 to 75 F any of the refrigerants mentioned can be used economically, but if the available condensing water temperature is above 80 F, carbon dioxide becomes uneconomical as its critical temperature is approximately 88 F. A condensing water temperature over 80 F makes the power required for compression high. All refrigerants have critical temperatures and pressures sufficiently high so that their efficiency is not materially affected by the condensing water temperatures, except in so far as this temperature affects the compression ratio. Steam jet cooling systems can use water up to 85 F, or even slightly warmer.

The applicability of the various refrigerants is interesting. Carbon dioxide is limited by the condensing water temperature; the power consumption is slightly higher than that of other refrigerants; and the pressures are three to four times that of ammonia.

The condenser pressures of methyl chloride and dichlorodifluoromethane are approximately one-half that of ammonia.

Ammonia, probably the best known refrigerant, has the disadvantage of being toxic, and under certain circumstances explosive, corrosive, and irritating, even in small quantities in the atmosphere. Ammonia is used exclusively in the larger indirect or brine cooling air conditioning systems.

Sulphur dioxide is corrosive and irritating even in small quantities in the atmosphere and it is toxic under certain circumstances.

Dichloromethane operates at pressures below that of the atmosphere, and it is to some extent toxic.

Dichlorodifluoromethane under normal circumstances is non-toxic, non-irritating, and non-explosive, but under high temperatures it breaks down into several obnoxious, poisonous components.

Methyl chloride, under certain conditions, is explosive and slightly toxic.

The steam ejector water vapor system has none of the disadvantages of toxicity, explosiveness and corrosiveness encountered in the other refrigerants, but the system operates at less than atmospheric pressure. This, however, is not an important factor as there are no moving parts in the compressor and the possibility of leakage of air is remote as all of the equipment can be welded air and water tight. The supply of water is inexhaustible, and as a refrigerant, the make-up cost is negligible. The same boiler equipment can be used for heating in winter and for cooling in summer.

Electric Motors

The motors used for driving compressors can be roughly classified in three groups: synchronous, multispeed, or variable speed. Further information on motors and their controls may be found in Chapter 42.

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 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 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, but there are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used, due to fire or other risks especially in densely populated areas, the brine can be

cooled in an isolated room or building and then be circulated through the air conditioning equipment in perfect safety because it is used to cool the water or air, without any possibility of direct contact between the air and refrigerant.

When an indirect system of cooling is used, it will be found that the heat transfer rate of the water cooler is considerably higher as a general rule than that of a direct expansion cooler for the same requirements. With direct expansion interchangers, it is almost impossible to keep the entire system flooded with liquid, whereas with brine interchangers the cooling medium completely fills the space of the interchanger and perfect contact is insured.

Ice may be used for chilling water or air for conditioning work. Its application is limited because of the cost of ice, and the difficulty of handling it. The word "water cooling" is used advisedly in that the direct cooling of air by ice is, while not impossible, rather impractical. It might be said that ice coolers are economical for systems requiring a maximum of 20 tons per 24 hours where the load fluctuates considerably, and where ice is introduced only as it is required to cool water. The most general method of cooling water with ice is to spray the water over the surface of the ice, insuring as much contact as possible and approximating the same performance as the Baudelot type of cooler. Because of the large fluctuations in load in the air conditioning system, the higher cost of refrigerating effect when ice is used is offset by the fact that there are no motor and condenser inefficiencies under partial load. Also, because the cost of the mechanical refrigeration equipment for the small system is so much higher per unit of effect, the fixed charges are small enough to overbalance the extra cost of the ice.

Condensers

Condensers are usually either the double pipe type or the shell and tube type. Shell and tube condensers are almost identical with coolers. Double pipe condensers are arranged so that water passes through the inner of two concentric pipes, and refrigeration passes through the annular space in the outer pipe. Where possible, there should be counter flow of the refrigerant and the condensing water to maintain maximum temperature differences.

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 strike a balance so that the quantity of water insures economical compressor operation.

As part of the condenser, or attached to it, there must be storage space for liquid refrigerant. The installation of all equipment should be made accessible for inspection, repair, and cleaning. Both the coolers and condensers should have space for pulling tubes.

Because there is a decided tendency to conserve the water in city mains and most large cities are restricting the use of water, in order to use air conditioning systems and refrigeration equipment it is often necessary to install cooling towers. The 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

not only the maximum load at normal conditions, but also 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 in that the temperature of the condensing water varies directly with the outdoor temperature and, as pointed out, the refrigeration load also varies with this temperature. Certain economies are possible when a cooling tower is used which cannot be achieved by the use of condensing water from city mains, even where the city water temperature is extremely low. Normally, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of what the outdoor temperatures are. With the 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 off the cooling tower drops considerably, and it has been established that 50 per cent of the time the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water, therefore, 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 negligible, as the only water required is that used to make up the loss by evaporation in the cooling tower itself. See also Chapter 11.

PROBLEMS IN PRACTICE

1 ● Discuss the difference in results obtained in cooling and dehumidifying in an air washer from those obtained in a surface cooling coil.

Air leaves a dehumidifying air washer in a saturated condition at a dew point temperature which can be easily maintained at a constant level by controlling the spray water temperature. This saturated air may then be reheated to proper delivery temperature by reheating coils or by mixing with bypassed air.

For a set air velocity and a set mean refrigerant temperature, a given cooling coil is capable of absorbing a definite amount of heat. Whether the air leaving the coil is saturated or not depends then on the entering dry- and wet-bulb temperatures. From a practical operating standpoint, the easiest way to control the output of the cooling coil is by means of the dry-bulb temperature of the conditioned space. This means then that the final dew point will vary somewhat depending on entering air conditions.

Summarizing then, the air washers permit close control over both final dry-bulb and final dew point temperatures, while the surface coolers permit close control over the final dry-bulb only.

2 ● For condensing purposes, an air conditioning system uses city water which has an average 70 F supply temperature. The following table lists the number of hours per year during which definite wet-bulb temperatures and corresponding refrigeration rates pertain.

Wet-Bulb Temperature F	No. of Hours per Year	Refrigeration Required Tons
80	6	284
79 - 75	100	233
74 - 70	277	183
69 - 65	330	157
64 - 60	277	144
59 - 55	158	79
54 - 50	52	37
Total 1200 hours		

If the power requirements of a dichlorodifluoromethane refrigeration system are in accordance with the following data on partial load operation, determine the seasonal power cost at 2 cents per kwhr:

Tons of Refrigeration	284	233	183	157	144	79	37
Kw per ton	0.89	0.89	0.87	0.86	0.86	0.93	0.97

Seasonal power cost:

WET-BULB TEMPERATURE F	TON-HOURS	KWHR
80	6 × 284 = 1,704	1,704 × 0.89 = 1,517
79 - 75	100 × 233 = 23,300	23,300 × 0.89 = 20,750
74 - 70	277 × 183 = 50,700	50,700 × 0.87 = 44,100
69 - 65	330 × 157 = 51,800	51,800 × 0.86 = 44,500
64 - 60	277 × 144 = 39,900	39,900 × 0.86 = 34,300
59 - 55	158 × 79 = 12,500	12,500 × 0.93 = 11,600
54 - 50	52 × 37 = 1,920	1,920 × 0.97 = 1,860
Totals	181,824 ton-hours	158,627 kwhr

The 158,627 kwhr at 2 cents per kwhr will cost \$3,173.

The average consumption will be $\frac{158,627 \text{ kwhr}}{181,824 \text{ ton-hours}} = 0.873 \text{ kw per ton}$.

3 • Using the data of Question 2, if a cooling tower were installed for re-using the condensing water, estimate the annual operating cost of a dichlorodifluoromethane refrigeration system if the final temperatures of the water leaving the cooling tower and the kilowatt input per ton are the following:

Tons	284	233	183	157	144	79	37
Temperature of water leaving tower, F	86.7	81.8	76.5	72.1	66.4	61.3	55.6
Kw input per ton	1.10	0.94	0.85	0.80	0.74	0.59	0.62

WET-BULB TEMPERATURE F	TON-HOURS	KW PER TON	KWHR
80	1,704	1.10	1,875
79 - 75	23,300	0.94	21,900
74 - 70	50,700	0.85	43,300
69 - 65	51,800	0.80	41,400
64 - 60	39,900	0.74	29,500
59 - 55	12,500	0.59	7,370
54 - 50	1,920	0.62	1,200
Totals	181,824 ton-hours		146,545 kwhr

The 146,545 kwhr at 2 cents per kwhr will cost \$2,931.

The average consumption will be $\frac{146,545 \text{ kwhr}}{181,824 \text{ ton hours}} = 0.805 \text{ kw per ton}$.

4 • Using the data from Question 2, if city water costs 20 cents per thousand gallons, and if 1.25 gallons are used per minute per ton, estimate the annual water cost.

$$60 \times 1.25 = 75 \text{ gal per ton-hour.}$$

$$181,824 \text{ ton-hours} \times 75 = 13,620,000 \text{ gal per year.}$$

$$\frac{13,620,000 \times \$0.20}{1000} = \$2,724, \text{ the yearly cooling water cost.}$$

5 • If a steam ejector system were used to secure the refrigeration for the air conditioning system of Question 2, compute the annual steam cost if steam is sold for 53 cents per thousand pounds and if there is an average steam consumption of 20 lb of steam per hour per ton when used with a cooling tower system.

$$181,824 \text{ tons} \times 20 \text{ lb of steam per ton} = 3,636,480 \text{ lb of steam.}$$

$$\text{The } 3,636,480 \text{ lb at 53 cents per thousand pounds will cost } \$1,929.$$

6 • From the data given in the following table covering auxiliary equipment, make a comparison between the operating costs of the complete dichlorodifluoromethane system of Question 4 and the complete steam ejector cooling system of Question 5. A cooling tower is used for condenser water recovery.

Plant Operation	Dichlorodifluoromethane System	Steam Ejector System
Hours of operation	1200	1200
Cooling tower fan, bhp	17.8	35.6
Cooling tower pump, bhp	30.2	47.8
Chilled water, gpm	1200	1200
Discharge head on chilled water system, ft	75	75
Pump efficiency, per cent	75	75
Motor efficiency, per cent	80	80
Chilled water temperature, F	46	46

The flash tank or evaporator of the steam ejector system is of the open type, the flash water being pumped directly to the sprays of the washer used for cooling the air.

Dichlorodifluoromethane System:

Power requirements,	
Cooling tower fan	17.8 bhp
Cooling tower pump	30.2
Total	48.0 bhp

$$\text{Power for cooling tower system} = \frac{48.0 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 53,700 \text{ kwhr.}$$

The water cooler in a dichlorodifluoromethane system of the surface type requires no additional pumping head other than the friction drop through the cooler, which in this problem is estimated to be 10 ft. The total pumping head is, therefore, 75 + 10 = 85 ft. Power required for the chilled water system will be,

$$\frac{1200 \text{ gpm} \times 8.34 \text{ lb per gallon} \times 85 \text{ ft head}}{33,000 \text{ ft lb} \times 0.75 \text{ pump efficiency}} = 34.3 \text{ bhp.}$$

$$\frac{34.3 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 38,300 \text{ kwhr.}$$

Thus, the total power required by the auxiliary equipment will be

53,700 + 38,300 = 92,000 kwhr.	
The 92,000 kwhr at 2 cents per kwhr will cost	\$1,840
The power cost of refrigeration, from Question 4, is	2,931
The total annual power cost, using a dichlorodifluoromethane system, is	\$4,771

Steam Ejector System:

Power requirements,

Cooling tower fan	35.6 bhp
Cooling tower pump	47.8
Total	83.4 bhp

$$\text{Power for cooling tower systems} = \frac{83.4 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 93,300 \text{ kw hr.}$$

In the flash tank or water cooler of the steam ejector system, the water is at a pressure corresponding to the chilled water temperature required. In this case it is at 46 F, which corresponds to an absolute pressure of 0.1532 lb per sq in. or 0.3118 in. Hg. This increases the pumping head on the chilled water circulating pump by $14.7 - 0.15 = 14.55$ lb per square inch, or 33.5 ft. The total pumping head is, therefore, $75.0 + 33.5 = 108.5$ ft.

$$\frac{1200 \text{ gpm} \times 8.34 \text{ lb per gallon} \times 108.5 \text{ ft head}}{33,000 \text{ ft-lb} \times 0.75 \text{ pump efficiency}} = 43.7 \text{ bhp.}$$

$$\frac{43.7 \text{ bhp} \times 0.746 \times 1200 \text{ hr}}{0.80 \text{ motor efficiency}} = 48,800 \text{ kw hr.}$$

The total power required by the auxiliary equipment is

$$93,300 + 48,800 = 142,100 \text{ kw hr.}$$

The 142,100 kw hr at 2 cents per kw hr will cost

\$2,842

The cost of the steam, from Question 5, is

1,929

The total annual power cost, using a steam ejector system, is

\$4,771

These calculations indicate that for the assumptions made, both the dichlorodifluoromethane system and the steam ejector system would cost 2.6 cents per ton-hour to operate. In order to obtain a complete analysis it would be necessary to compare the fixed charges which include interest, depreciation, obsolescence, and maintenance. These are customarily computed at 15 per cent of the initial cost per annum. To this cost must be added the cost of refrigerant make-up per year. In the steam system this is negligible, but in the dichlorodifluoromethane system it may be approximated at from $\frac{1}{4}$ to $\frac{1}{2}$ of the refrigerant charge per year.

7 • If, in a given air conditioning installation, fixed charges are estimated at 15 per cent per annum, what increased first cost would be justified for an auxiliary appliance on a refrigeration machine which could save \$125.00 per year in operating cost?

$$\$125.00 \text{ capitalized at 15 per cent is, } \frac{\$125.00}{0.15} = \$834.00 \text{ increased first cost justified.}$$

8 • Electrically driven Freon condensing units are to be used in an air conditioning system, requiring 20 tons refrigerating capacity for conditions of maximum load. An overall analysis of the seasons operating conditions shows an average load factor of 62.5 per cent, and allowing for variable time intervals of operation of refrigeration units installed, three-quarters of the operating season, or 750 hours, would require operation of the equipment at one-half load, and one-quarter of the operating season or 250 hours full load capacity of the refrigeration equipment would be required.

The increased first cost of 2-10 hp, 10 ton condensing units over 1-20 hp, 20 ton condensing unit is, \$830.00 installed price, to the customer.

The increased first cost of a 2-speed compressor motor of 20 hp size over a constant speed 20 hp size motor including increased starter cost is \$210.00. The efficiency of the 2-speed motor above is 83 per cent at full load full speed, and 79 per cent for full load at $\frac{1}{2}$ speed. At $\frac{1}{2}$ -speed, full load is $\frac{1}{2}$ total bhp of full load full speed.

Discuss the considerations involved in making a decision as to whether a single unit with a 20 hp motor of the 2-speed type would be used in preference to 2-10 hp constant speed units.

The cost of 2-10 hp 10 ton units in excess of 1-20 hp, 20 ton unit with 2-speed motor, is \$830.00—\$210.00 or \$620.00, increased first cost. At 15 per cent fixed charges, this represents an increased annual cost of \$93.00 for 2 compressors over one compressor. The advantage of 2 compressors instead of one compressor on an installation of this type, is in the breakdown service provided in the event one compressor is shut down for repairs in the system could be operated at one-half capacity utilizing the duplicate machine. The motor efficiency of the constant speed unit would be higher at full load than would be the efficiency of the 2-speed motor at low speed. Offsetting this latter advantage however, is the fact that the condenser on the condensing unit would provide a lower refrigerant condensing temperature for $\frac{1}{2}$ load operation with the same final condensing water temperature than would be the case with duplicate units each furnished with its own compressor and condenser. Operation at a lower condensing temperature would provide for a power saving compensating for the lower efficiency of the 2-speed motor when operated at slow speeds. It is, in a case of this kind, purely a question as to whether or not the purchaser would deem an investment of \$620.00 more and an increased fixed charge of \$93.00 a year, advisable to get breakdown service thru the installation of duplicate units. In most cases, this increased first cost would not be warranted because of the fact that satisfactory indoor conditions could not be obtained at full load if only one-half the refrigeration capacity were available.

9 • For an increased first cost of \$210.00 for a 20 hp, 2-speed motor over a 20 hp constant speed motor to be used on a Freon automatic condensing unit for air conditioning duty, justify the increased investment based on a load factor of 62.5 per cent, on an operating basis of 1000 hours total of which $\frac{3}{4}$ of the operation of the refrigeration equipment would be at $\frac{1}{2}$ load and $\frac{1}{4}$ of the operation season at full load. Motor efficiency 2-speed, 83 per cent, at full speed; and 79 per cent at half speed. Motor efficiency constant speed motor 83 per cent. City water is to be used for condensing purposes and is to be automatically controlled from the discharge pressure by means of automatic regulating valve. Compute increased first cost justifiable based on 15 per cent fixed charges and number of years required to pay off increased first cost.

At full load for 20 bhp load and 83 per cent efficiency, the kw input to the compressor motor would be 20 bhp times 0.746 kw per bhp divided by 83 per cent motor efficiency or 18.0 kw.

At $\frac{1}{2}$ speed,

$$\frac{10 \text{ bhp} \times 0.746 \text{ kw per hp}}{0.79 \text{ efficiency}} = 9.45 \text{ kw input at } \frac{1}{2} \text{ speed.}$$

For a constant speed motor installation, 1000 hours operation at 18 kw load, requires 18,000 kw hr per season for operation.

With 2-speed motor 750 hours at $\frac{1}{2}$ speed and 9.45 kw input = 7080 kw hr.

For full speed operation, 250 hours at 18 kw input = 4500 kw hr.

The total kw hr required with 2-speed motor = 7080 + 4500 = 11,580 kw hr.

From the above it is seen that the power consumption is reduced by the use of a 2-speed motor from 18,000 kw hr per year to 11,580 kw hr or approximately a power saving of 35 per cent annually.

For power at \$.02 per kw hr, 6400 kw hr saving per year would mean a saving of \$128.00 per year, which at 15 per cent would justify an increased first cost of \$853.00.

Further, the fixed charge added due to the increased first cost of a 2-speed motor would be 15 per cent of \$210.00, or \$31.50 per year, thus the net annual saving after correcting for fixed charge due to increased investment would be \$128.00 less \$31.50 fixed charge or a net annual saving of \$96.50 with a 2-speed motor based on a power cost of \$.02 per kw hr. Thus the increased first cost of \$210.00 would be returned to the purchaser in $\$210.00 \div \96.50 or 2.18 years.

Similarly a \$.03 power the gross saving exclusive of additional fixed charges to increased cost of 2-speed motor would be \$192.00; net saving per year \$160.50 and therefore, the 2-speed motor would pay for itself in 1.31 years.

For \$.04 power, the gross saving would be \$256.00; net saving, \$224.50 and the 2-speed motor would be paid for in 0.935 years.

It is safe to assume that any appurtenance which will pay for itself in less than 5 years when computed on the above basis, will be accepted by practically all purchasers of air conditioning equipment, provided an analysis of the above type is properly presented.

10 • In a locality where the electric power rate is based on a demand charge, it is desired to install the smallest possible compressor motor which will provide summer cooling for a 300-seat restaurant which operates 6 hours per day from 11 a.m. to 2 p.m., and from 5 p.m. to 8 p.m. The refrigeration load at the peak is 28 tons. If the load factor for both the noon and evening meals is 70 per cent, discuss the type of equipment which would take the greatest advantage of the reduced power rate at low kilowatt demand.

A storage system using a chilled water storage tank would permit the installation of a refrigeration system having the smallest motor.

For a 28-ton system operating 6 hours per day at a 70 per cent load factor, on the maximum day the total heat removed would be,

$$28 \text{ tons} \times 6 \text{ hr} \times 0.7 = 117.5 \text{ ton-hours per day.}$$

If a compressor were to operate 24 hours at a constant rate, its average capacity would be $\frac{117.5 \text{ ton-hours}}{24 \text{ hours}} = 4.9 \text{ tons}$, or approximately 5 tons. If operated 12 hours per day, the compressor capacity would have to be increased to 10 tons.

A water storage tank would store the refrigeration and allow off-peak operation, so a smaller compressor motor could be used. However, the suction temperature at which the compressor would be operated would be lowered approximately 5 to 10 F. This would increase the horsepower per ton of refrigeration, when dichlorodifluoromethane is used, approximately 10 per cent for a 5 F reduction and 24 per cent for a 10 F reduction in the suction temperature. Rather than store the water at too cold a temperature, it would be more economical to install a larger storage tank and use a higher temperature.

A 5-ton compressor running during periods when there are no customers, namely, during the 15 hours from 8 p.m. to 11 a.m. will have stored $15 \text{ hr} \times 5 \text{ tons}$, or 75 ton-hours, of refrigeration in the storage tank by 11 a.m. As one ton-hour equals 12,000 Btu, $75 \times 12,000 \text{ Btu}$, or 900,000 Btu, will have been stored.

If the apparatus dew-point temperature is 54 F, and the chilled water is supplied to the air washer at 48 F, it will leave at 54 F. If the water in the storage tank is at 40 F, the temperature difference between the stored water and the water entering the washer will be $48 \text{ F} - 40 \text{ F} = 8 \text{ F}$. This is equivalent to an available 8 Btu of cooling effect per lb of water stored. Therefore, $\frac{900,000}{8}$ or 112,500 lb of water must be stored. This is

$$\frac{112,500}{8.34} = 13,500 \text{ gal water to be stored, which equals } \frac{13,500}{7.5} = 1800 \text{ cu ft of water.}$$

The storage tank to hold this water might be 6 ft high, $7\frac{1}{2}$ ft wide, and 40 ft long. Should this volume prove impractical, a proportionately smaller tank could be used if the water storage temperature were reduced. Should a 10-ton refrigeration system be used, the water quantities and tank capacity could be reduced by one half, and the refrigeration plant need not be started until 8 a.m. daily, which might prove of additional advantage.

If refrigeration is stored by freezing ice on coils, considerable storage space will be saved but more power input per Btu of cooling will be required.

Chapter 11

HUMIDIFICATION AND DEHUMIDIFICATION

Air Washers, Atmospheric Water Cooling Equipment, Cooling Towers, Design Wet-Bulb Temperature, Cooling Ponds, Natural Draft Deck Type Towers, Mechanical Draft Towers, Winter Freezing, Apparatus for Industrial Conditioning

EQUIPMENT for humidifying and dehumidifying is of varied character and its functions will be discussed in this chapter. An air washer is essentially a chamber in which air is brought into intimate contact with water, the object being (a) to wash the air or (b) to regulate the moisture content of the air and at the same time wash it. The air comes in contact with the water by passing it through water sprays or by passing it over surfaces wetted by a continuous flow of water; hence the classification: spray, scrubber, and combination spray and scrubber type washers.

A washer chamber may be constructed of wood, or stone, but it is most often constructed of sheet metal. The lower portion of it is specially designed as a tank to receive the water dropping through the chamber and to serve as a reservoir from which the water may be recirculated.

It is desirable that air leaving a washer contain no water in suspension. For this reason eliminators are provided at the washer outlet. These may be in the form of plates or baffles upon which the free moisture is deposited as the air is deflected through several changes from its original direction of flow. In some washer units steel wool filter sections serve as eliminators. However, specially designed plates are used more generally than other devices because they offer the least resistance to the flow of air, while still performing effectively the function of free moisture elimination. They also have the advantage of acting as scrubber surfaces when flooded.

It is essential to uniform performance in a washer, that air enter evenly distributed over the washer inlet. To insure this, a perforated plate or eliminator plates are installed at the inlet. Eliminator plates are now more generally used. They serve a second purpose in preventing the escape of spray through the washer inlet.

Water is supplied to scrubber type units through flooding nozzles. The capacity of these nozzles varies with the manufacturer although a fair value of 5 gpm may be used. The nozzles are spaced on one-foot centers across the top of the washer over the scrubber plates.

Water is supplied to spray type units through atomizing nozzles generally arranged in banks across the washer. The nozzles spray either in the direction of the air flow, that is, downstream, or against the air flow, or upstream. Nozzle capacities vary with the manufacturer, from $1\frac{1}{2}$ to 2 gpm at a water pressure of about 25 lb per square inch which pressure

is required for effective atomization. The spacing of spray nozzles is determined by the water requirements of the particular installation. A spray type washer may contain one, two or three banks of nozzles depending upon its application.

When an air washer is used for cleaning air it removes impurities and dusts. In general it does not function as efficiently in this service as a filter. For non-microscopic soluble dust its efficiency averages about 50 per cent, unless the concentration of dust is high. Its effectiveness in removing greasy microscopic dust is practically negligible as is also its deodorizing ability.

When a washer is used to regulate the moisture content of air it adds moisture to (humidifies) or removes moisture from (dehumidifies) the air to achieve the desired moisture content.

When air passes through a washer wherein water is circulated without the addition or removal of heat, the air tends to become saturated at its entering wet-bulb temperature. What occurs here is partial or complete adiabatic saturation. The total heat content of the air is unchanged, inasmuch as the dry-bulb temperature of the air drops in proportion to the amount of additional water evaporated. This action is also known as evaporative cooling. A measure of the washer's effectiveness under these conditions is its saturating efficiency which is equal to the drop in dry-bulb temperature in per cent of the entering wet-bulb depression. Other things being equal, the saturating efficiency of a spray type washer is a function of the number of spray banks and the direction in which they spray. The following table gives a general comparison:

3 banks—2 upstream—1 downstream.....	100% saturation efficiency
2 banks—2 upstream.....	95% saturation efficiency
2 banks—1 upstream—1 downstream.....	85% saturation efficiency
1 bank —upstream.....	80% saturation efficiency
1 bank —downstream.....	65% saturation efficiency

When air passes through a washer wherein the circulated water is either cooled or heated before being returned to the spray chamber, a heat interchange between the air and water occurs, and the air tends to become saturated at the temperature of the leaving water. The extent to which the leaving air and leaving water temperatures approach each other is an index to the effectiveness of the washer under the operating conditions. The total heat absorbed by the water in the process equals the total heat given up by the air, or the heat given up by the water equals the heat absorbed by the air. Depending on whether the moisture content of the air is increased or decreased during the operation, humidification or dehumidification occurs. Heat will be added to or removed from the air as the water supplied is of a higher or a lower temperature than the wet-bulb temperature of the entering air.

For dehumidifiers the ratio of the difference between the leaving wet-bulb and the leaving water to the difference between the entering wet-bulb and the entering water may be figured as follows:

3 banks—1 downstream—2 upstream.....	0%
2 banks—2 upstream.....	5%
2 banks—1 upstream—1 downstream.....	15%
1 bank —upstream.....	20%
1 bank —downstream.....	35%

Humidifiers may be figured on the same basis as dehumidifiers; the leaving water temperature, of course, will be higher than the wet-bulb temperature of the leaving air.

The problem of cooling or heating the circulated water before returning it to the washer chamber is external to the unit. It will suffice here to note that heating is generally accomplished by passing the water through closed hot water heaters or by injecting steam into the water circuit; cooling, by passing the water through closed coolers or over refrigerating coils in a Baudelot chamber. Often in a cooling and dehumidifying application, the refrigerating coils are located within the washer chamber.

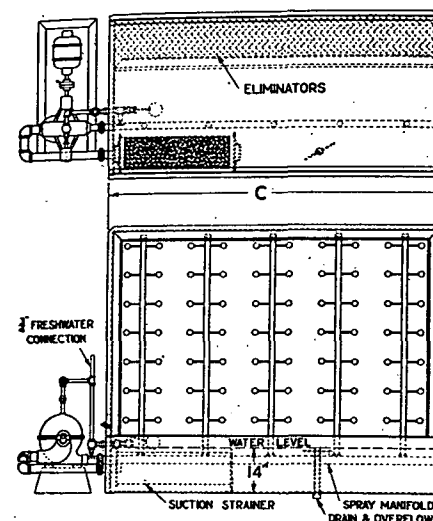


FIG. 1. TYPICAL SINGLE BANK AIR WASHER

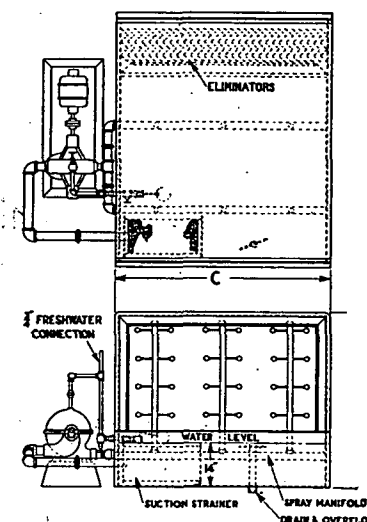


FIG. 2. TYPICAL TWO BANK AIR WASHER

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 air and the cooling or heating medium (water, brine, etc.). A multi-stage washer is equivalent to a number of washers in-series arrangement. Each stage is in effect a separate washer.

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 feet per minute through the gross cross-sectional area of the unit above the water level in its tank. At this rating spray type washers handle about 2-1/2 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. Two of the more important items are:

a. Choose a washer for air velocities above approximately 300 fpm and below approximately 600 fpm. Velocities outside this range are likely to result in faulty elimination of entrained moisture.

b. When a high saturating efficiency is required, select a two or three bank spray type unit, having a total water capacity of not less than 15 gpm per 1000 cfm.

The area of a washer may be dictated by space limitations outside the washer, such as headroom, or by space requirements inside washer, 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, (the *leaving* eliminators require about 1 ft 6 in., *entering* eliminators about 1 ft).

The resistance to air flow through an air washer varies with the type eliminators, number of banks of sprays, direction of spray, type of scrub-

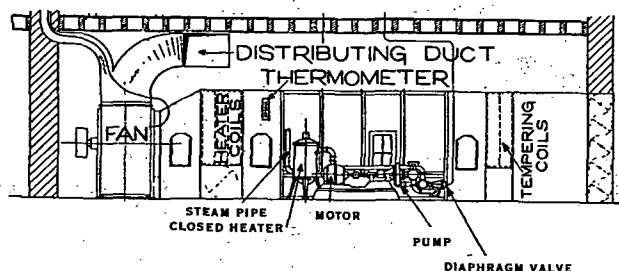


FIG. 3. AIR WASHER WITH SPRAY WATER HEATING ARRANGEMENT

ber plates, and, if cooling coils are located in unit, by their size and type. Washers should be selected to limit static resistances below 0.50 in.

Power Requirements

The approximate power requirement for passing 10,000 cfm of air through a humidifier of the spray type by a fan of 78 per cent mechanical efficiency is given in Table 1, this being the fan brake horsepower for various velocities and static pressure losses. Allowance should be made for variations in static pressure due to the use of different diffuser plates or inlet louvers and for variations in fan efficiencies.

ATMOSPHERIC WATER COOLING EQUIPMENT

To successfully operate a refrigerating plant or a condensing turbine, the heat from the compressed refrigerant or the discharged steam must be removed and dissipated. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger. If the plant is situated on the banks of a river or lake, an intake may be had upstream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of water is a city supply or well water, the discharge water may be run into the nearest sewer or open waterway. Lacking an unlimited water supply, or in cases where city water is too expensive or where the water available contains

dissolved salts which would quickly form scales on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-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. Because the changes in rate do not occur in direct proportion to changes in the governing factors, data on the performance of atmospheric water cooling equipment are largely empirical.

TABLE 1. APPROXIMATE FAN BRAKE HORSEPOWER

Requirements for passing 10,000 cfm of air through humidifiers at various velocities and static pressures. Mechanical efficiency of fan—78 per cent.

VELOCITY FPM	30 DEG ELIMINATORS SPACED ON 1-1/8 IN. CENTERS		45 DEG ELIMINATORS SPACED ON 2-1/4 IN. CENTERS	
	Static Pressure In. Water	BHP	Static Pressure In. Water	BHP
500	0.20	0.40	0.40	0.80
550	0.24	0.48	0.48	0.97
600	0.29	0.58	0.58	1.15
650	0.34	0.68	0.68	1.35

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 1.) Because it is impractical 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 possible expenditure of power.

Cooling Towers

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 quantity of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design, however, in that the size of the equipment is determined by the quantity of air that must 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.

Sizes of Equipment

Assuming a definite quantity of water to be cooled, the size and design of atmospheric cooling equipment are affected by the following factors:

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.

TABLE 2. CONDENSER DESIGN DATA

Gas	MAXIMUM PRESSURE DESIRED IN CONDENSER	GAS TEMPERATURE IN CONDENSER F	LEAVING HOT WATER TEMPERATURE F	
			Best Design	Average Design
Steam.....	28 in. vacuum.....	99.7	97	93
Steam.....	27 in. vacuum.....	114.3	110	105
Steam.....	26 in. vacuum.....	126.0	120	114
Ammonia.....	185 lb gage head pressure.....	96.0	92	88
Carbon dioxide..	1030 lb gage head pressure.....	86.0	88	81
Methyl chloride.....	102 lb gage head pressure.....	100.0	96	92
Dichlorodi- fluoromethane	117 lb gage head pressure.....	100.0	96	93

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 lb 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 lb 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 deg 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. 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. Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.
2. 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 1931, inclusive, there were but 6 hrs 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 refrigeration 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 a refrigerating 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 1, Chapter 8, shows safe 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 efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmospheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

$$\text{Percentage cooling efficiency of atmospheric water cooling equipment} = \frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is 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 parts of the cooling equipment are:

Compressor refrigeration.....	220 to 270 Btu per minute per ton
Condenser turbine.....	950 to 980 Btu per pound of steam
Steam jet refrigerating apparatus.....	1030 to 1150 Btu per pound of steam
Diesel engine.....	2800 to 4500 Btu per horsepower

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, as natural air movement over the surface of the water will cause evaporation

TABLE 3. EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

EQUIPMENT	COOLING EFFICIENCY—PER CENT		
	Minimum	Usual	Maximum
Spray Ponds.....	30	45 to 55	60
Spray Towers.....	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers.....	35	50 to 70	90
Mechanical Draft.....	35	55 to 75	90

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 off. 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.

Spray pond efficiencies are 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 nozzles from 3 ft to 6 ft above the edge of the basin, to supply from 5 lb to 12 lb 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 0.1 gpm per square foot in a small pond to 0.8 gpm per square foot in a large pond.

Increasing the pressure, spacing the nozzles farther apart, or increasing the elevation of the nozzles will increase the cross-section of spray cloud exposed to the air, and therefore increase the quantity of air coming in contact with the water. 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 if they have an earthen or a concrete basin, or they may be placed on roofs having special waterproof roofing. 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 formations may be a considerable nuisance in a spray pond. Such growths are killed by the periodic addition of potassium permanganate to the pond water. Addition of the dissolved chemical should be made until the water holds a faint pink color for at least 15 min.

Spray Cooling Towers

Where not more than 30,000 Btu per minute are to be dissipated, the spray cooling 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 cooling 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

louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years most 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 15 ft to 80 ft high and from 6 ft to 30 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 also add to the area of water surface exposed to the air, but since they furnish a resistance to air flow, too many decks are 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. These boards or louvers extend outward and upward, and in most designs the top edge of each louver extends above the bottom edge of the one above it.

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.

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 on days 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 1 Chapter 8, 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 should it need a wind velocity of more than 5 mph. It is obvious that natural draft towers and other natural draft equipment must be so located that they are not obstructed by trees, buildings, or other wind deflectors.

Mechanical Draft Towers

Mechanical draft towers usually consist of vertical 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 checker-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 circulated through the tower from bottom to top by forced or induced draft fans. Since the air flows counter to the water, the air is in contact with the hottest of the water

just before leaving the top of the tower, and each unit of air picks up more heat than a similar unit would on natural draft equipment, so the mechanical draft tower cools water by using less air than the other types of equipment need. As movement of the air through the towers is obtained by power-consuming fans, it is essential that the air used be reduced to a minimum so as to secure the lowest possible operating cost.

The efficiency of a mechanical draft tower is increased 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 louvered 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 fpm to 600 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. Such apparatus does not have the counter flow advantage of the vertical mechanical draft water cooling equipment, and therefore requires a much larger excess of air for proper operation. Air velocities and operating powers are considerably above those required by vertical mechanical draft water cooling equipment.

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. Because the evaporation of the water leaves behind the salts which the water contained, high concentration of salts may make chemical treatment of the make-up water 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 below 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

TABLE 4. COMPARISON OF VARIOUS TYPES OF ATMOSPHERIC WATER COOLING EQUIPMENT

Figures indicate order of desirability

	COOLING POND	SPRAY POND	SPRAY TOWER	DECK TOWER	MECHANICAL DRAFT	INDOOR TOWER
Cost.....	x	2	1	3	4	5
Area.....	5	4	3	2	1	x
Height.....	1	2	3	4-5	4-5	x
Weight per sq ft.....	x	x	1	3	4	2
Independence of wind velocity.....	6	3	4	5	1-2	1-2
Drift nuisance.....	1	6	5	4	2-3	2-3
Make-up water required.....	1	6	5	4	2-3	2-3
Pumping head.....	1	2	3	4-5	4-5	6
Maintenance.....	2	1	3	4	5	6
Suitability for congested districts.....	x	5	4	3	1	2
Water quantity required for definite result.....	6	5	4	1-2	1-2	3

*Not comparable.

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.

A comparison of various types of water cooling equipment is given in Table 4.

APPARATUS FOR INDUSTRIAL CONDITIONING

Apparatus for industrial air conditioning may be divided into two distinct groups, namely, (1) *humidifiers* for increasing the moisture content of the air and for producing cooling by evaporation and (2) *dehumidifiers* for removing moisture from the air and for producing cooling by contact with water or surfaces at a lower temperature than the air.

Strictly speaking, humidity control alone, whether it involves humidification or dehumidification, is not *air conditioning*. To be entitled to this classification according to the definition in Chapter 44, the process should include the *simultaneous control* of temperature, humidify and air motion.

Industrial humidifiers may be divided into the following general types, according to the method of operation:

1. *Direct*, which spray into the room.
2. *Indirect*, which introduce moistened air.
3. *Combined* direct and indirect.

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 which cannot be permitted 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 to the wet-bulb temperature. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is quickly evaporated and the resulting vapor is rapidly and thoroughly diffused. This effective distribution of fine spray over the maximum possible area insures complete and extremely rapid vaporization even at the highest humidities.

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.

The spray and high-duty types of humidifiers have many features in common but the latter, because of its finer spray and greater capacity, is often considered better adapted for producing high humidities.

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 cleaning and servicing of individual units without affecting the room as a whole, group control of the water and power may be employed.

Where large quantities of power are generated in a limited space and where a comparatively high relative humidity is required, it is often feasible and economical to use a combination of direct and indirect humidification. The indirect humidification provides the desired quantity of ventilation and cooling, and the additional direct humidification provides for increase in humidity without interfering with the ventilation or the cooling effected by the indirect system.

In general, it may be stated that direct humidification is most satisfactory where high humidities are desired but where little cooling, ventilation or air motion is required. Therefore, the indirect system is most applicable where either low or high relative humidities are desired with maximum cooling and ventilation effect. For conditions that require an unusually large amount of heat to be absorbed by ventilation, together with the maintenance of high humidities, it is often preferable to make use of the combination system of indirect and direct humidification. If the indirect system alone were used it would mean an unusually large volume of air to be handled, which might interfere, due to air motion, with production, even though it would result in greater cooling effect. If direct humidification alone were used, no ventilation would be obtained, with consequently higher room temperatures.

PROBLEMS IN PRACTICE

1 ● Determine the per cent saturating efficiency of an air washer used for adiabatical saturation or evaporative cooling if the following temperatures are obtained on the entering and leaving sides of the washer:

Entering air, 84.3 F dry-bulb and 71.2 F wet-bulb.

Leaving air, 73.4 F dry-bulb and 71.2 F wet-bulb.

The equation for determining saturation efficiency is:

$$E = \frac{(db_1 - wb_1) - (db_2 - wb_2)}{db_1 - wb_1} \times 100$$

where

E = saturation efficiency in per cent.

db_1 = dry-bulb temperature of entering air.

wb_1 = wet-bulb temperature of entering air.

db_2 = dry-bulb temperature of leaving air.

wb_2 = wet-bulb temperature of leaving air.

Therefore

$$E = \frac{(84.3 - 71.2) - (73.4 - 71.2)}{84.3 - 71.2} \times 100 = \frac{10.9}{13.1} \times 100 = 83.2 \text{ per cent.}$$

2 ● How may relative humidity be controlled?

a. If constant room temperature is to be maintained:

1. To maintain a constant relative humidity, the dew point must be kept constant.
2. To increase the relative humidity, the dew point must be raised.
3. To decrease the relative humidity, the dew point must be lowered.

b. If constant dew point is to be maintained:

1. To maintain a constant relative humidity, the room temperature must remain constant.
2. To increase the relative humidity, the room temperature must be lowered.
3. To decrease the relative humidity, the room temperature must be raised.

c. With varying dew-point temperatures:

1. To maintain a constant relative humidity, the room temperature must vary directly and in almost equal amount with the dew point.
2. To increase the relative humidity, the difference between room temperature and dew point must be decreased.
3. To decrease the relative humidity, the difference between room temperature and dew point must be increased.

d. With varying room temperatures:

1. To maintain a constant relative humidity, the dew point must vary directly and in almost equal amount with the room temperature.
2. To increase the relative humidity, the difference between dew point and room temperature must be decreased.
3. To decrease the relative humidity, the difference between dew point and room temperature must be increased.

3 ● In industrial air conditioning plants, what are the four sources of heat which must be taken into consideration in the design of a system?

- a. Heat transfer from the outside air.
- b. Body heat from employees.
- c. Sun effect.
- d. Heat equivalent of power consumed in driving machinery, in lighting, and in manufacturing processes in general.

4 ● Why do cooling towers give best results when the humidity of the air is low?

The cooling of water by dropping it through air depends mostly upon the evaporation of the water. If the relative humidity of the air is low, the water vapor will be readily absorbed and carried away, while if the humidity of the air is high, its capacity to pick up water vapor is less and the water is cooled less with the same exposure to air.

5 ● What performance tests should be given air washers?

- a. Capacity.
- b. Resistance.
- c. Visible entrainment of free moisture.
- d. Humidifying efficiency.
- e. Cleaning effect.

6 ● What are the several different types of water-cooling towers?

- a. Those with forced draft.
- b. Those with natural draft open to the atmosphere.
- c. Those with natural draft closed to the atmosphere.
- d. Those with combined natural and forced draft.

7 ● What are the different types of air washers?

- a. Spray. b. Wet scrubber. c. Combination spray and scrubber.

8 ● What is the saturation efficiency for an air washer with the common variations in spray arrangement?

For three banks, two up-stream and one down-stream.....	100%
For two banks, both up-stream.....	95%
For two banks, one up-stream and one down-stream.....	85%
For one bank, up-stream.....	80%
For one bank, down-stream.....	65%

9 ● Upon what air velocity are air washers usually rated?

500 fpm, through the area above the tank.

10 ● What wet-bulb temperature for the outside air is usually selected in air conditioning design when cooling is to be accomplished?

One which is not exceeded more than 5 to 8 per cent of the time in the locality where the plant is to be situated.

11 ● Where should the suction connection be placed in a cooling pond?

As far below the surface as possible and as far away from the discharge as practicable.

12 ● What chemical is used to kill algae formations in spray ponds?

Potassium permanganate.

13 ● What is the usual amount of spray water delivered to a cooling pond per square foot of pond area?

From 0.1 gpm on small sizes to 0.8 gpm on large sizes.

14 ● What is the usual amount of water delivered in cooling towers per square foot of area?

From 0.6 to 1.5 gpm.

15 ● About how much water is lost by evaporation in atmospheric cooling?

About 1 gal per 1000 gal for each degree of cooling range.

16 ● How is freezing obviated in cooling pond sprays?

The pressure and quantity of water is lowered so that the drops become of increased size and do not freeze so readily.

Chapter 12

UNIT HEATERS, VENTILATORS, COOLERS, AND AIR CONDITIONERS

Classification of Unitary Equipment and Related Systems, Advantages, Uses, Functions, Types of Construction, Locations, Capacities, and Accessories to Unitary Equipment

IN other chapters, complete descriptions have been given of heating, cooling, ventilating, humidifying, and dehumidifying systems. These descriptions have covered the detailed principles of each and have, in general, described the assembled equipment that has been used to make these the complete systems. The success of such completely engineered, heating, cooling, and air conditioning systems has inevitably lead to the production of smaller factory-assembled equipment employing a majority of the principles of these complete systems. As a result, present day practice involves the use of this unitary equipment in the majority of smaller installations where capacity demands are within the limits of such units. Thus unit heaters, unit ventilators, unit coolers, and unit conditioners have come to occupy a place of their own in the industry.

With the growth of this unitary industry, it becomes increasingly evident that there is no sharp line of demarcation, on the basis of capacity, between a *unit* and a so-called *central station* system. The various industrial codes have thus found it necessary to describe a *unit* as a factory-made encased assembly, shipped substantially complete, or built and shipped in sections so that the only field work necessary is the assembling together of the sections, without resorting to any field fabrication. These units consist of assemblies of functional elements indicated by their names, such as *Air Conditioning Unit*, *Room Cooling Unit*, *Humidifying Unit*, etc.

A unit such as defined may be complete in itself, employing its own direct means of air distribution and sources of refrigeration or heating, in which case, it thus represents a complete self-contained unit system. Or it may be coupled with separate means of air distribution such as duct work and outlets, in which case, it will still be considered as a unit system, in distinction to the generally accepted term of a field fabricated central station system. The manufacturer of the unit is responsible for the output and performance of the unit under rated conditions, whereas the contractor installing the complete unitary system is normally held responsible for the complete performance of the system.

Unit equipment as described previously justifies its existence due to the following features:

1. Lower cost per unit capacity.

Standardized design and volume production makes possible low cost factory assembly thereby eliminating individual design and handling of every part for each installation.

2. Flexibility and mobility of equipment.

Unitary equipment can be readily located in existing buildings without the necessity of running large ducts through floors and many partitions. Such equipment can be shifted to meet changing requirements. Tenants may obtain the advantages of conditioning when the entire building is not equipped with a conditioning system. In industrial process work, the flexibility of unitary equipment is of great advantage.

3. Lower installation costs.

The fact that the equipment arrives on the job in an assembled condition, coupled with the lesser problems of duct work and connecting piping, materially reduces installation costs.

4. Small capacities.

The small capacities available in unitary equipment have brought the advantages of controlled air conditions to a number of small offices, stores, shops, and individual rooms where specially designed and built central system equipment would have been uneconomic.

This chapter will limit itself in general to a description and discussion of the functions and construction of the units themselves indicating the accessory features necessary to complete unitary systems.

SUB-DIVISION OF UNITARY EQUIPMENT

While for descriptive purposes, it would be better to sub-divide unitary equipment on a purely functional basis, such as heating, cooling, ventilating, etc., there has grown up very definite branches of this industry in which the engineering and application of the equipment vary quite widely. Thus common acceptance in the industry recognizes the following sub-divisions:

1. *Unit heaters* consisting of an encased heating surface through which air is forced by means of a fan or blower, located either in or closely adjacent to the heated space, and normally employed only for industrial and commercial applications.

2. *Unit Ventilators* which are similar in principle to unit heaters but are designed to use outside air with or without provision for recirculation of the air. While unit heaters are largely used for commercial and industrial applications, unit ventilators are intended primarily for school, offices and semi-commercial applications.

3. *Unit Coolers* which are similar to unit heaters except that a cooling medium is used in place of a heating medium and provision is made to catch and remove the condensate. A distinction is made here between unit-coolers, normally applied to the cooling of products for their preservation or processing and unit conditioners used for cooling for comfort.

4. *Unit Conditioners* which consist of equipment to provide control of heating with humidifying, or cooling with dehumidifying, coupled with air circulation; all compactly housed in a single casing.

5. *Miscellaneous Unit Equipment and Accessories* such as filtering equipment, attic fans, humidifying units and special controls.

These sub-divisions will be discussed in detail.

UNIT HEATERS

A *unit heater* consists of the combination of a heating element and fan or blower having a common enclosure and placed within or adjacent to the space to be heated. Generally no ducts are attached to inlets or outlets, although it is common practice with many unit heaters to equip them with directional outlets or adjustable louvers. While unit heaters were first applied to industrial heating, the constant development and improvement of this equipment both as to appearance and quietness has broadened its field of application until today unit heaters are available for fields extending from large factory spaces to recreation rooms, churches, and offices.

While unit heaters are designed primarily to handle all recirculated air, they may be installed to handle either partial or total out-door air. Compared with the older method of heating by means of radiation, properly designed and applied unit heaters should:

1. Circulate air in the building at a rapid rate but without objectionable draft.
2. Reduce the temperature differential between the floor and ceiling.
3. Direct the heated air so that uniform temperature distribution be obtained throughout the heated space.
4. Prevent or remove the cold stratum of air commonly found at the floor level.
5. Reduce the number of heating elements required and thereby decrease the cost and extent of the piping necessary.
6. Maintain a closer control of room temperature either manually or by means of simple thermostats.
7. Produce an economy in heating costs resulting from the sum total of the above advantages.

TYPES OF UNITS

A wide variety of types of unit heaters is available. They all employ some form of convector, supplied with either steam or hot water, although occasionally equipped for gas or electric heat. Air is always forced over these convectors by a fan of either the propeller or centrifugal type. Heating surfaces may be in the form of steel pipe coils, non-ferrous tubes or pipes with extended surfaces, cast iron, and pressed or built up sections of the cartridge or automotive type.

Two major sub-divisions of design are commonly recognized. Floor mounted units which withdraw the cold air from the floor and discharge the heated air above the working zone and the suspended type where the unit is located in an elevated position withdrawing the air from this level and discharging the heated air down into the working zone. In closely occupied spaces where direct air drafts into the working zone are not permitted, the floor mounted unit will give more uniform temperature distribution. On the other hand if opportunity is provided to deliver the heated air from suspended units down into the working zone excellent temperature distribution is possible.

Housed fan, high velocity discharge units with outlets adjustable to deliver air in several directions are able to project their heating effect over distances of from 30 ft to as much as 200 ft from the unit. This makes possible the location of these units at considerable distances from each

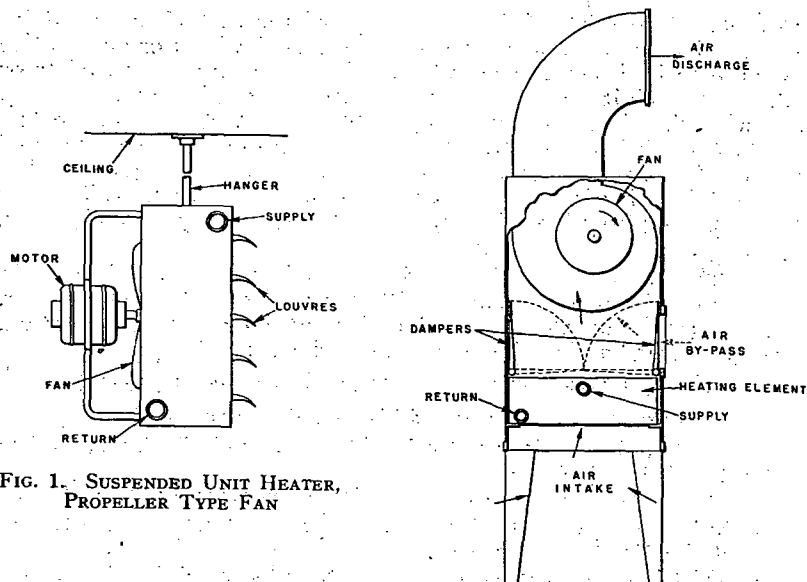


FIG. 1. SUSPENDED UNIT HEATER, PROPELLER TYPE FAN

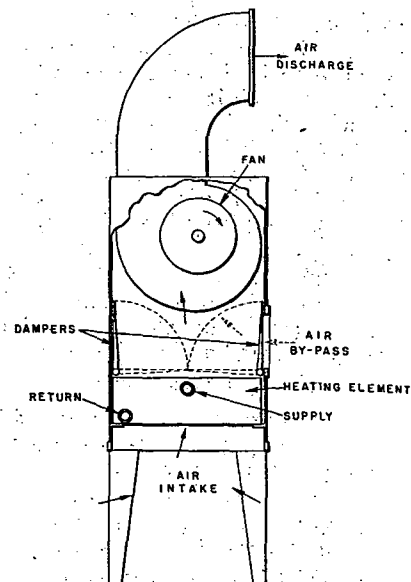


FIG. 2. FLOOR MOUNTED UNIT HEATER, HOUSED TYPE FAN

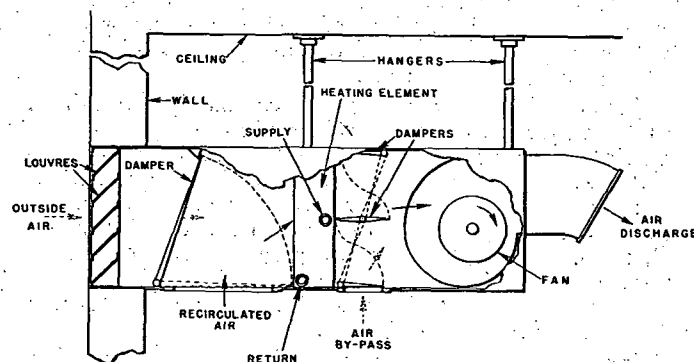


FIG. 3. SUSPENDED TYPE UNIT HEATER, HOUSED TYPE FAN

other, thus greatly reducing the piping and loss of floor space as a result of the heating equipment. Propellor-type fan units with outlet velocities of from 300 to 1000 fpm are usually placed from 30 up to 100 ft apart.

Fig. 1 shows a typical suspended type unit heater with propellor type fan. Fig. 2 illustrates a typical floor mounted type unit heater with by-pass feature. Such units are available either with or without this air by-pass. For purposes of this control feature see Chapter 14. Fig. 3 shows suspended high-velocity type unit heater connected to outside air intake with damper to control volume of ventilation.

HEATING MEDIUM

The convectors of unit heaters or ventilators may be supplied with either hot water or steam. When water is used, it should be circulated mechanically and the pump rate and friction loss should be based upon test data for the particular unit to be employed. Normally the friction within the convector is too great for gravity circulation. The heat output of a given heater will normally be less when using water than with steam at the same temperature, unless high rates of circulation are maintained.

Either high or low pressure steam may be used but it is essential that proper venting of the air from the coil be obtained at all times. Where unit heaters are used with outside air in freezing climates, it is unwise to use steam of less than 5 lb pressure at the coil, due to the possible danger of freezing of condensate in the tubes. It is essential that properly constructed traps and some form of thermostatic air by-pass be used with high pressure steam as well as adequate condensing legs to prevent steam in the returns. Increasing the return temperature tends to increase return line corrosion, especially at points where overheated condensate or steam enters the return line.

When low pressure steam is used with unit heaters and ventilating units, it is important that proper means be provided for the removal of the heavy condensation. Such units should not be applied to low pressure gravity return systems except where the difference in heater level and boiler water line is large enough to compensate for the pressure loss through the convector at its highest condensation rate. The use of vacuum or return pumps and receivers is advisable with jobs of any considerable size, as the best method of taking care of the condensate and at the same time providing for proper venting of the units directly into a vacuum return line system, or into an open vented return system, the latter having some advantage in preventing the formation of any vacuum in the unit itself, which sometimes tends to hold up condensate and causes freezing.

ESTIMATING HEAT LOSSES

The heat losses of a building to be equipped with unit heaters are determined in the same manner as for any other heating system, excepting so far as the unit heaters may prevent air stratification and thus reduce the temperature difference between the ceiling and floor. (See Chapter 7.)

Unit heaters may be arranged to recirculate the air or to supply warmed air from the outside for ventilation or to make up air exhausted.

If all or a part of the air is to be taken in from out-of-doors, the heat necessary to warm this air from the outside temperature to the inside temperature must be added to the transmission or other losses. Units of the number and size needed to furnish the total heat required are then selected from the manufacturers' rating tables, using these ratings at the steam pressure to be used and at the temperature at which the air will enter the convector.

AIR TEMPERATURES¹

For recirculating heaters with intakes at the floor level, the temperature to be maintained in the room should be considered as the temperature of the air entering the heater. Where outside air is introduced, the temperature of the mixture must be calculated and used as the entering air temperature to the heater. Where suspended heaters are used without any intake boxes extending down to the floor level, a higher entering air temperature should be used than that at which the room is to be maintained.

With suspended unit heaters taking air at some distance above the floor, the temperature variation from floor to ceiling may reach as much as 1 deg for each foot of elevation during the periods when the maximum capacity of the heaters is required. Thus this allowance should be made in calculating the capacity of suspended heaters. Unit heaters taking in recirculated air at the floor level should maintain temperature differentials of less than 0.5 deg per foot of elevation when the maximum capacity of the heaters is required. This temperature difference per foot of elevation is less than the corresponding variations for spaces heated by direct radiation.

Unit heaters save fuel because of their ability to circulate air at a lower average temperature than the air circulated by direct radiators; however, the unit heaters must circulate more air in any given time than is needed with direct radiators. This requires the selection of heaters having a liberal air capacity for the required heat output, which in turn means a relatively low final temperature. Extremely low final temperatures can be had only at the expense of larger heaters and increased power, so that an economic limit is imposed. In general, for heating purposes it is advisable to use a delivery temperature not more than 70 F above the average room temperature desired, and considerably less where possible. Since the delivery temperature increases with increase in steam pressure with high pressure steam, units should be selected that have a minimum heating surface in order to provide outlet air not in excess of 70 F above average room temperature.

OUTPUT OF HEATERS

It is standard practice to rate unit heaters in Btu per hour at a given temperature of air entering the heater and at a given steam pressure maintained in the coil. Steam at 2 lb pressure and air entering at 60 F

¹Temperature 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).

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. JOURNAL SECTION, Heating, Piping and Air Conditioning, November, 1934).

are used as the standard basis of rating². The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heat capacity for any condition of steam pressure and entering air temperature may be calculated approximately from any given rating by the use of factors in Tables 1 and 2. Table 1 is for blow-through and Table 2 is for draw-through unit heaters. These tables are accurate within 5 per cent.

The ratings customarily published for unit heaters apply only for recirculation and free discharge, unless otherwise noted in the rating tables. If outside air intakes, filters, or ducts on the discharge side are used with the heater, proper consideration should be given to the reduction in air and heat capacity that 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 employed, so that no specific percentage of reduction can be assigned for all heaters for a given added resistance. In general, however, disc or propeller fan units will have a larger reduction in capacity than housed fan units for a given added resistance, and a given heater will have a larger reduction in capacity as the fan speed is lowered. When confronted with this problem the ratings under the conditions expected should be secured from the manufacturer.

When steam supplied to the heaters contains superheat, the capacity of the heater will be but slightly less than with saturated steam at the same pressure. Recent tests indicate that the reduction of capacity from this cause is negligible for superheat up to 50 deg and will not exceed 3½ per cent for any degree of superheat.

DIRECTION OF DISCHARGE

Heaters may be distributed through the central portions of a room discharging toward exposed surfaces, or may be spaced around the walls, discharging along the walls and inward as well, especially when there are considerable roof losses.

In general, it is better to direct the discharge from the unit heaters in such fashion that rotational circulation of the entire room content is set up by the system rather than to have the heaters discharge at random and in counter directions.

Various types and makes of unit heaters are illustrated in the *Catalog Section* of this edition. Usually hot blasts of air in working zones are objectionable, so 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 to the head line as possible, yet not into the working zone. In general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

²See A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Heaters (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

TABLE 1. 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-lb Gage and Entering Air Temperature of 60 F)

TEMPERATURES OF ENTERING AIR													
Steam Pressure Lb per Sq. In.	-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	1.355	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 and 2 lb pressure.

TABLE 2. 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-lb Gage and Entering Air Temperature of 60 F)

STEAM PRESSURE LB PER SQ. IN.	TEMPERATURES OF AIR ENTERING HEATER											
	-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 and 2 lb pressure.

BOILER CAPACITY

The capacity of the boiler should be based on the rated capacity of the heaters at the lowest entering air temperature that will occur, plus an allowance for line losses. Ordinarily for recirculating heaters the lowest entering temperature will occur at the beginning of the heating period and is usually taken as 40 F, while for ventilators taking air from outdoors the lowest entering temperature will be the extreme outdoor temperature expected in the district. No greater allowance in boiler capacity beyond the calculated heat demand need be added in order to supply unit heaters than for any other type of system.

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 attendance 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, however, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller heating units instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their coils are designed for the purpose and when proper provision is made for returning the condensate. If ventilators are to take in air that may be at a temperature below freezing, however, a steam pressure of not less than 5 lb should be maintained on the convector or a corresponding differential in pressure between the supply and returns be maintained by means of a vacuum.

QUIETNESS

In selecting unit heaters, attention should be given to the degree of quietness required for the installation.

No given fan speed may be applied as a measure of relative quietness to fans of different designs and proportions. Quietness is a function of type, diameter, blade form and other variables besides speed, and all these must be taken into account. In general small fans may be run at higher motor speeds than large fans with equal quietness; and centrifugal fans are more easily made quiet than disc or propeller fans.

PIPING CONNECTIONS

Piping connections for unit heaters are similar to those for other types of fan-blast heaters. Typical connections are shown in Figs. 4 and 5.

One-pipe gravity and vapor systems are not recommended for unit heater work.

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. Pressure-drop must be compensated for by elevation of the heater above the water line of the boiler or of the receiver.

In pump and receiver systems the air may be eliminated by individual

air valves on the heaters, or it may be carried into the returns the same as for 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 vented 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

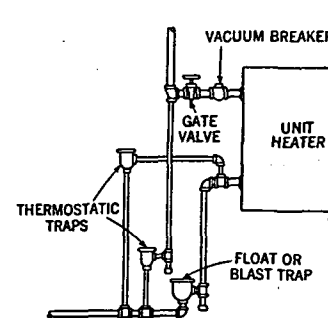


FIG. 4. UNIT HEATER CONNECTIONS WHERE CONDENSATION IS RETURNED TO VACUUM PUMP OR TO AN OPEN VENTED RECEIVER

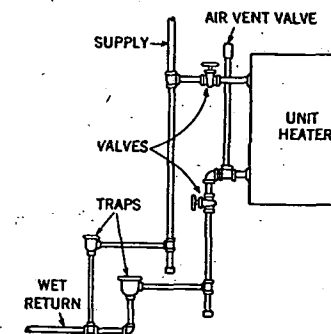


FIG. 5. UNIT HEATER CONNECTIONS WHERE CONDENSATION IS RETURNED TO BOILER THROUGH WET RETURN

high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap, with some danger of return pipe corrosion and the problem of its elimination at some other point in the system.

The connections for steam and return piping to unit heaters must always be calculated on the basis of the high heat emission or condensation rate of such devices. The pipe-size tables given in Chapter 32 may be used for unit heater work by multiplying EDR values by 240 to get Btu values.

OTHER TYPES OF UNITS

All Electric

The foregoing discussion relates generally to units in which steam or hot water is used as the heating medium. On rare occasions electrical resistances are used as the heating element. These are applied only where electric power is abundant and cheap and where other forms of fuel are scarce and expensive. (See Chapter 39.)

Direct Fired

A recent development in gas burning equipment is the direct-fired industrial unit heater. These heaters are of the warm-air type and are

equipped with fans which cause the air to pass over the heating surfaces at a fairly high velocity and then direct the warm air in to the space to be heated. As is the case with the steam-fed unit heaters, the gas-fired appliances may be used for heating stores, shops, and warehouses. They usually are suspended in the space to be heated and in most instances leave the entire floor and wall area free for commercial use. Partial or complete automatic control also may be secured on appliances of this type. This type of heater is often used for temporary heat during building construction or where the installation of a steam or hot water plant is for some reason not justified. For permanent installations, it is usually advisable to provide an exhaust duct from the gas-fired unit heaters to remove products of combustion from the occupied space. While this is not necessary in large open industrial plants, in smaller closed rooms, it becomes essential.

Turbine Driven

Where high pressure steam is available it is sometimes used to drive a steam turbine direct-connected to the unit heater. The exhaust from this turbine, reduced in pressure, is then passed into the heating coil where it is condensed and returned to the boiler.

INDUSTRIAL USES

In addition to their prime function of heating buildings, unit heaters may be adapted to a number of industrial processes, such as drying and curing, with 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 given off. When such conditions are severe, it is necessary that the 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 7.)

Information on the control of unit heaters will be found in Chapter 14.

UNIT VENTILATORS³

Unit ventilators while designed primarily for ventilation must incorporate controlled heating. Since these units are generally located in the occupied space, they must be pleasing in design and must harmonize with the furniture or with the decorative scheme. A typical unit ventilator is illustrated in Fig. 6. They usually consist of a semi-decorative cabinet containing the following necessary or optional parts:

1. Outside air inlet.
2. Inlet damper for closing the opening to the outside air inlet when the unit is not in use.
3. Adhesive or dry type filters for cleaning the air (optional).

³A roof ventilator is sometimes termed a unit ventilator. For information on roof ventilators, see Chapter 4.

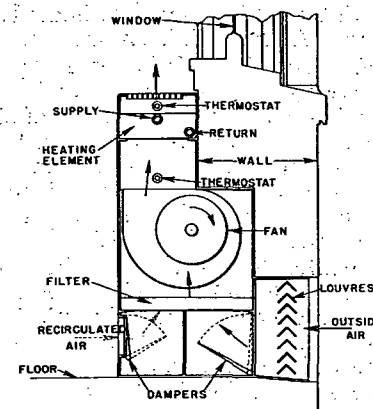


FIG. 6. TYPICAL UNIT VENTILATOR SHOWING ONE OF MANY ARRANGEMENTS OF DAMPERS AND HEATING COILS

4. A heating element usually of special design and intended for low pressure steam.
5. Motor and fan assembly.
6. Mixing chamber where warm and cold air streams are brought together. (No mixing chamber is normally provided where sectional type convectors are used.)
7. Outdoor air inlet and recirculating air mixing damper (optional).
8. Device for ozonizing air (optional).
9. Discharge grille or diffuser.
10. Temperature control arrangement.

The primary functions of a unit ventilator are:

1. To supply a given quantity of outdoor air for ventilation or to mix indoor and outdoor air.
2. To warm the air to approximately the room temperature if the unit is intended for ventilation only, or to a higher temperature if it is intended to take care of all or a part of the heat transmission losses from the room.
3. To control the temperature of the air delivered so as to prevent both cold drafts and overheating.
4. To deliver air to the room in such a manner that proper distribution is obtained without drafts.
5. To recirculate room air for the purpose of heating or promoting comfort when ventilation is unnecessary.
6. To perform all its functions without objectionable noise.

In addition to these functions, unit ventilators frequently are arranged so that the air supplied may be cleaned by means of filters of either the dry or viscous type. If filters are used, the proper allowance must be made for the increased resistance offered to the air flow. Humidifiers in unit ventilators are rather difficult to control and are only furnished upon special order.

1. *Air Supply for Ventilation.*—The outdoor air supply for ventilation is delivered by motor-driven fans operated at comparatively low speeds; the back of the cabinet being connected to the outside through rust-proof

louvers and screens. Air quantities may be estimated on the basis of data given in Chapter 3. (See A.S.H.V.E. Ventilation Standards.)

2. *Warming Incoming Air.* The air is heated by passing it through specially designed convectors. The amount of heating surface to be provided in the unit is determined by the volume of air to be heated and the temperature range. If the unit is to be used for supplying air for ventilation only, the convector must be sufficient in capacity to maintain a final air temperature of about 70 F. If the unit is to be used for heating as well as for ventilation, the convector must be sufficient to maintain the necessary final air temperature for the conditions involved.

3. *Control of Temperature.* This is accomplished by varying the temperature of the air discharged from the unit (1) by the automatic operation of a mixing damper which controls the relative quantities of air being blown through the heating unit or by-passed around it, (2) by operation of valves on different layers of convector surfaces, or (3) by variation in the temperature of the circulating heating medium.

The outside air inlet damper and recirculating damper (where one is provided) should be so connected that there will be an uninterrupted supply of air to the fans at all times the unit is in operation. These dampers may be operated by hand or by pneumatic or electric motors manually controlled from some central point.

These dampers may also be linked together, in the form of mixing dampers and be controlled by a thermostat in the cold air intake, by a differential thermostat acted upon by both the cold air and the recirculated air, or by a thermostat in the two streams of air after they are mixed, so as to keep the relative proportion of air taken in from out-of-doors commensurate with outside temperatures and to prevent drafts of cold air being blown through the unit into the room.

Provision should be made for the inlet damper to close automatically whenever the fans are shut down, and not to open until the room is properly heated when the fans are again started. The minimum temperature of the air delivered by the machine should be regulated automatically by a thermostat in the outlet air which controls the temperature of the heated convector, or this minimum temperature may be maintained by properly mixing the inside and outside air by means of the mixing dampers under thermostatic control referred to above. Another thermostat in the recirculated air intake to the unit or elsewhere in the room controls by-pass dampers or the supply of heating medium, or both, so as to control the temperature of the air leaving the unit according to the heat requirements of the room. In addition to these thermostats, a room thermostat is needed to control any other heat sources for the room. (See Chapter 14.)

Thermostats for controlling by-pass dampers must be of the intermediate type to hold the dampers in intermediate positions to prevent objectionable drafts. When direct radiators are used in conjunction with unit ventilators, the control is usually arranged so as automatically to open the valves to the direct radiators when the room temperature falls about 2 deg below the setting of the thermostat for the unit ventilator. Another arrangement opens the radiator valve whenever the unit ventilator control reaches the full heating position. Further information on this subject is contained in Chapter 14.

4. *Distribution.* This function is governed by the proper selection and location of the unit. Diffusion and distribution are dependent upon a relatively high velocity air stream discharged in a generally vertical direction, and in order to insure satisfactory diffusion in the room the less the difference between the temperature of the air discharged from the unit and that of the room air, the better. With a final temperature above 110 F, excessive stratification of the air may be experienced. Troublesome drafts may be eliminated to a large extent if a static pressure is built up in the room.

5. *Recirculation of air* requires less fuel than does the use of all outside air and aids in heating up quickly. Certain units are designed to recirculate all air at all times, except when the admission of outside air is needed to regulate room temperatures. Under this arrangement, the outside air for ventilating purposes is obtained solely from infiltration, but the amount thus obtained may or may not be sufficient to meet legal ventilating requirements for public buildings. Recirculation of the air in schools is therefore prohibited by ordinance in many communities. Ventilating systems in schools should be arranged for taking in a sufficient quantity of air to constitute, with infiltration, not less than 10 cfm per occupant of a room.

6. *Quiet Operation.* Since the unit ventilator is generally set in close proximity to the room occupants, it must operate with exceeding quietness.

SPLIT AND COMBINED SYSTEMS

In a *split* system the unit is used primarily for ventilation. Air is delivered to the room at very near the room temperature, and enough separate direct heaters are placed in the room to warm it to the desired temperature, independently of the unit. Their principal advantage lies in offsetting the cooling effect of window and wall surfaces long before these can be heated to room temperature and in retaining heat for this purpose after the ventilation is shut down.

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 heating surface 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.

A combined system employs the unit ventilator alone, its capacity being sufficient both for ventilation and for supplying the heat loss. Direct heating surface is omitted altogether. It becomes necessary then that the fan be running whenever the room is to be heated but this also gives assurance of ventilation, especially if automatic dampers are used in the air intake from out-of-doors and in the recirculating intake arranged so as to give a certain quantity of air from the outside (commensurate with weather conditions) whenever the unit is operating and after the room is heated. 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 practically no heating.

LOCATION OF UNIT

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.

VENTS⁴

The size and location of the vent outlet is important. In many cases the sizes for public buildings are regulated by law, but the location of the vents generally 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.

The cross-sectional area of the vent flue itself may be figured on the basis of 15 sq in. of flue for each 100 cfm. Thus the vent flue area of a flue for a room equipped with one 1200 cfm unit ventilating machine would be 180 sq in. The area of vent flue opening from the room may be figured on the basis of 25 sq in. per 100 cfm.

In school buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, heating and 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.

There has been much controversy over the use of corridor ventilation in school building practice, one group holding the view that when each classroom has a separate vent flue there is a minimum fire risk and less likelihood of cross-contamination, while others emphasize the economy features of the corridor discharge and minimize the fire, contamination, and other hazards.

CAPACITIES

Unit ventilators are available in air capacities ranging from 450 cfm to 5000 cfm and with corresponding heat capacities (above that required for ventilation purposes based upon an outside temperature of zero and an inside temperature of 70 F) ranging from 30 Mbh to 144 Mbh (1 Mbh = 1000 Btu per hour). Some manufacturers furnish a unit with several heating capacities for each air capacity, thus enabling the engineer to select the unit best adapted to the heating and ventilating load. Capacities

⁴Investigation 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).

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. JOURNAL SECTION, Heating, Piping and Air Conditioning, June, 1935).

ties should be determined in accordance with the A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators⁵. Typical capacities are given in Table 3.

The amount of heat to be supplied by the unit ventilator will depend on the amount of air passed through the unit and the temperature range through which the air is heated. The weight of air (W) to be circulated per hour is fixed by the ventilating requirements.

If no direct heating surface (radiation) is installed, the combined heating and ventilating requirements must be taken care of by the unit ventilators, and the total heat to be supplied is obtained by means of the following formulæ:

When all of the air handled by the unit is taken from the outside,

$$H_t = 0.24 W (t_y - t_o) \quad (1)$$

$$W = d \ 60 \ Q \quad (2)$$

$$t_y = \frac{H}{0.24 W} + t \quad (3)$$

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.

t_o = outside temperature.

t_y = temperature of the air leaving the unit.

W = weight of air circulated, pounds per hour.

0.24 = specific heat of air at constant pressure.

From Equations 1, 2 and 3:

$$H_t = H + 0.24 \ d \ 60 \ Q \ (t - t_o) \quad (4)$$

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. $H = 24,000$; $d = 0.075$

$Q = 1000$ cfm; $t = 70$ F; $t_o = 0$ F.

Substituting in Equation 4:

$$H_t = 24,000 + 0.24 \times 0.075 \times 60 \times 1000 (70 - 0) = 99,600 \text{ Btu}$$

$$t_y = \frac{24,000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2 \text{ F}$$

When part of the air handled by the unit is taken from the room and the remainder from the outside,

⁵Adopted 1932. See A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.

TABLE 3. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

CUBIC FEET OF AIR PER MINUTE	TOTAL CAPACITY IN SQUARE FEET OF EQUIVALENT DIRECT HEATING SURFACE (RADIATION)		CAPACITY AVAILABLE FOR HEATING THE ROOM IN SQUARE FEET OF EQUIVALENT DIRECT HEATING SURFACE (RADIATION)		FINAL AIR TEMPERATURE (DEG FAHR)
	EDR	Mbh	EDR	Mbh	
600	285	68	95	23	105
750	350	84	115	28	105
1000	455	110	150	36	105
1200	565	136	190	46	105
1500	705	169	235	56	105

$$H_t = 0.24 W_o (t_y - t_o) + 0.24 W_i (t_y - t) \quad (5)$$

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.

$$W_o = d_o 60 Q_o \quad (6)$$

$$W_i = d_i 60 Q_i \quad (7)$$

where

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.

$$t_y = \frac{H}{0.24 (W_o + W_i)} + t \quad (8)$$

$$H_t = H + 0.24 d_o 60 Q_o (t - t_o) \quad (9)$$

Equations 5, 6, 7, 8, and 9 may be used in the same manner as is illustrated above for Equations 1, 2, 3, and 4. It may be noted in Equation 9, 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 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 requirements H_t reduce from 99,600 Btu to 24,000 Btu, or to about one fourth. Such a unit handling one third of its air volume from the outside and two thirds from the room would show a total heat requirement of $24,000 + \frac{99,600 - 24,000}{3} = 59,200$ Btu. 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.

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24 W (t_y - t) \quad (10)$$

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:

$$H_v = 0.24 W (t_y - t_o) \quad (11)$$

In this case t_y should be equal to or slightly higher than t . If the unit ventilator were of such capacity as to exactly provide 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.

UNIT COOLERS

Unit coolers as applied to industrial product conditioning and processing are very similar in construction to unit heaters except that the heat transfer medium is supplied with refrigeration instead of with steam or hot water. They are normally installed within the space to be served, or at least closely adjacent thereto. Occasionally they are provided to receive outside air in which case this air is almost always filtered or washed to prevent any possible contamination of the product.

Unit coolers are provided in two major types similar to unit heaters, either floor mounted with housed fan, or suspended with propeller type fans. Normally, air outlet velocities are lower than for heating, due largely to the effect of high velocities on the product. Consequently unit coolers are occasionally supplemented with duct distribution.

Product cooling originally was accomplished by means of stationary pipe coils. This was later supplemented with the forced fan bunker systems in which air was passed over banks of coils. The present trend in this field is toward a more accurate control of both temperature and humidity, thus placing these units in the classification of complete air conditioning units as discussed in the next section. However, in the majority of these cases dry-bulb temperature is controlled separately from the control of humidity, thus classifying these units as unit coolers.

The principal field for unit coolers is in cold storage plants, fur storage, fruit packing houses, provision stores, brewery fermentation and stock rooms, candy plants, and other industrial process work. In replacing bunker and wall coils in meat storage plants, unit coolers give distinct advantages in compactness, lower first cost and maintenance expense, ease of defrosting, freedom from drip and the maintenance of sanitary conditions, as well as uniform temperature and humidity under variable load conditions. Unit coolers by means of their positive air circulation prevent dead-air spots, frequently objectionable in this industry.

Typical unit coolers are shown in Figs. 7 and 8. The former indicates a

suspended type unit cooler which may be designed with or without a moisture eliminator. If high air velocities are maintained, an eliminator will be necessary to prevent the drops of moisture from being carried through with the air. The condensation that occurs is collected in a drip pan and removed from the system through a drain pipe. Fig. 8 indicates a typical floor-mounted unit of the housed fan type. The illustration shows a common form of distributing outlet designed to give low outlet velocities together with a controlled distribution. In process work, it is

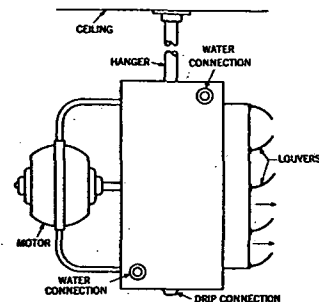


FIG. 7. CEILING TYPE COOLING UNIT

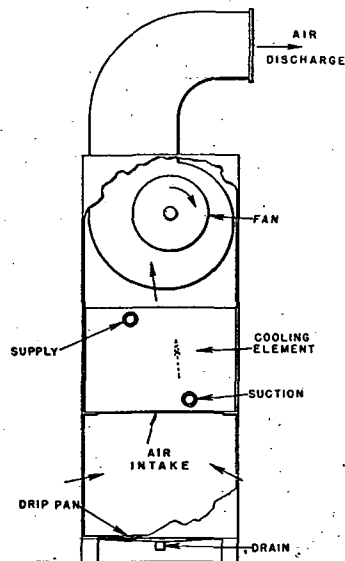


FIG. 8. FLOOR MOUNTED SURFACE TYPE COOLING UNIT WITH DISTRIBUTING OUTLETS

often important that direct air distribution does not impinge on the product.

Unit coolers are normally constructed of galvanized steel or non-ferrous material in order to reduce the corrosive effect of their constant wetted condition.

Unit coolers are often called upon to operate in rooms where a temperature below freezing is maintained and low refrigerant temperatures are required. This results in the collection of frost on the heat transfer surface which in turn leads to a rapid loss in capacity and requires eventual defrosting. Such defrosting is accomplished by the following methods:

1. When the room is above freezing the source of refrigeration is cut off and the fan allowed to operate until the unit has defrosted.

2. A reversal of the refrigeration system may be provided and the so-called hot gas defrosting method used. This is accomplished by reversing the flow of the hot gas so that it is delivered directly from the compressor to the evaporator of the

cooling unit. As soon as the ice and frost has been melted, the system is again returned to its normal cycle.

3. Where brine is used as a refrigerant, heated brine may be sent through the cooler to remove the ice.

4. When the room is at very low temperatures, warm air defrosting is sometimes used by providing for the admission and removal of warm air from outside the cooled space.

5. The surface may be sprayed with a strong brine solution.

In order to prevent the collection of frost in low temperature rooms where high latent heat loads are present, unit coolers equipped with a constant brine spray are frequently used. These are normally of the

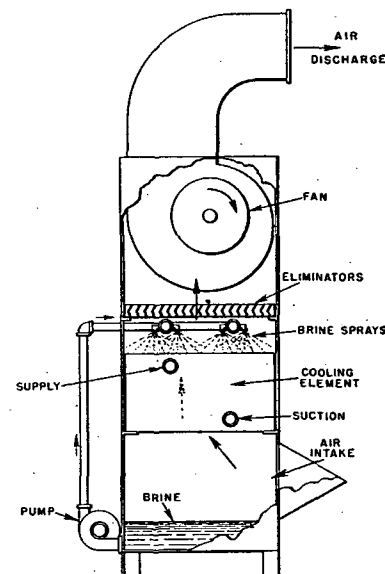


FIG. 9. BRINE SPRAY TYPE COOLING UNIT

housed fan type similar to Fig. 9, but equipped with a pump for recirculating brine over the coil. It is, of course, necessary to strengthen the brine at intervals to maintain a non-freezing mixture.

Ratings of unit coolers may be expressed in Btu per hour, or in tons of refrigeration and should specify the quantity, temperature and humidity of the air entering the unit with a stipulated refrigerant temperature within the coil. When chilled water or brine is used, the rate of circulation of the cooling media as well as its entering temperature must be given.

UNIT AIR CONDITIONERS

According to the definition given in Chapter 44, to qualify as an air conditioning unit such equipment must simultaneously control at least

the temperature, humidity, and motion of the air within the conditioned space. Air conditioning with unit equipment has gained in popularity during the last few years and some equipment offered does not fulfill the basic requirements of a true air conditioning unit. It is to be noted that air conditioning equipment must have the ability not alone to alter the temperature or humidity within the conditioned space, but it must *control* these conditions. Thus, some equipment that attempts to provide cooling or humidification does not have the capacity or means of actually controlling these conditions.

In selecting and specifying the unit types, the results desired should be carefully determined. If the simultaneous control of temperature, humidity, air motion, and distribution within the enclosure is not required then the simpler and less expensive unit heaters and unit coolers may be considered. Complete air conditioning should simultaneously *control* the following features:

1. Air temperature.
2. Air humidity (moisture content).
3. Air motion and distribution.
4. Air purity.

Three general classifications of unit air conditioners are recognized:

1. All-year conditioning involving the control of these conditions both in summer and winter and thus requiring both heating and cooling as well as humidifying and dehumidifying.
2. Winter conditioning in which heating and humidifying alone are controlled together with air motion and air purity.
3. Summer conditioning involving cooling and dehumidifying with air distribution and cleaning.

The means of accomplishing these functions are outlined herewith:

Heating

The normal air conditioning unit derives its heating function from a heating coil, usually of the non-ferrous finned tube type supplied with either steam or hot water. Steam may be supplied directly from a self-contained and built-in oil or gas fired unit, from a separate domestic steam boiler, or even from an outside source of a central heating plant. Hot water is supplied either from a separate hot water boiler or in rare instances from a domestic water heater.

In domestic or household conditioning units, adapted from a warm-air heater to which humidification is added, or possibly all-year-round conditioning, a direct-fired air interchanger is frequently used. The source of heat in this case may be from the combustion of coal, oil, or gas. A wide variety of designs and structures are used. In such direct-fired systems, the bulk and volume of the heat transfer surface is necessarily large in proportion to the rest of the equipment.

Where electric power is low in cost, electric heat has been furnished in unit air conditioners either in the form of encased heaters, or open wire heaters. (See Chapter 39.) Radiant electric heaters are seldom used except as their radiant heat is absorbed by some receiving wall and there transmitted to the air in the form of convected heat.

Another method of applying electric heat is by means of the reversed refrigeration cycle, whereby electric energy is used to compress a refrigerant and to deliver the heat of compression, withdrawn from a lower temperature source, to the conditioned space by locating the condensing coils in the air circulation circuit of the unit conditioner. While this method of heating has gained wide interest, it is practical only in a limited number of applications. Its economic practicability depends largely upon the availability of a source of heat during the winter season at as high a temperature as is possible. Thus, while it is frequently applicable in mild climates, it is not of great value in the northern climates where sub-freezing temperatures are common. It has the other unfortunate property that when the outdoor temperature is the lowest and the heat requirement the greatest, the heating capacity of the system is a minimum. This means that in many installations, an auxiliary and separate source of heat must be supplied to carry through these peak winter conditions. Such factors as these have limited a more general application of this principle. (See Chapter 2 on Refrigeration.)

Humidifying

A very wide variety of humidifying means have been used to furnish humidification in winter to air conditioning units. The oldest and best known is by means of a direct spray which is used in many different ways. The simplest system is where the spray water is furnished from a constant water source, such as city water, and is permitted to run to waste. Under such conditions, the spray may be either of the direct atomizing type where, by means of the nozzles, the water is broken into fine particles or of the so-called target spray type, where a fine stream of water under pressure is caused to impinge upon a flat surface or target. Such methods are normally rather inefficient in the use of water.

Any spray system where water is run to waste must of necessity atomize as completely as possible to provide for evaporation of the maximum amount of water. This normally calls for moderately high pressures and for very fine orifices or passages in order to produce fine atomization. This in turn involves the danger of occasional clogging of the nozzles with resulting service problems. Recently a number of self-cleaning nozzles have come on to the market which give promise of reducing this hazard. Such methods even at best are wasteful of water inasmuch as only a comparatively small proportion of the water used is actually evaporated, the remainder running to waste. In spite of the water wastage this method is normally more economical in the case of small conditioning units than the installation of a pump to recirculate and reuse the waste water.

In the larger conditioning units where much larger quantities of water are used and the amount of water wastage becomes an appreciable item, it frequently is profitable to install a recirculating pump. Comparatively few units employ this recirculating system. It is important to note that when spray water is recirculated in such equipment in industrial areas and in large cities, it becomes necessary that water treatment be supplied, in order to reduce the corrosive action resulting from the absorption of acids and impurities from the air.

In some units, in order to increase the humidifying capacity and to utilize a greater portion of the spray water, the atomized spray is permitted to impinge against a heated surface thereby forcing its evaporation. While this is practical in some instances, there is danger of scale formation where hard water is employed. It should be pointed out that in any atomizing system if extremely hard water is used, the air will carry with it a content of infinitely fine solids which should be removed by a filter before discharging into the conditioned space. If it is not so removed, this may result in the formation of an apparent *dust* upon furniture and exposed surfaces.

One of the simplest methods of humidification in winter is by means of a direct steam spray. This is seldom used in unit conditioners for comfort applications due to the resulting odors. In industrial applications, however, it finds frequent use. The steam is usually introduced to the air through a perforated tube or through some type of porous material.

If a small atomizing spray is not used in comfort conditioning units, the evaporative pan type of humidifier is usually employed. This consists of a container offering as much water surface as possible and equipped with means of heating the water. This heat may be applied either electrically, by steam, hot water, or by circulation of the water from a heated space through the evaporating pan. Since the humidification is accomplished by surface evaporation only, it is essential that the air stream be directed across the surface and that the evaporating surface be large. While this system eliminates the dusting hazard when hard water is used with a spray system, hard water tends to scale the heating surface and results in loss of capacity and the need for frequent cleaning.

The rate of evaporation per unit surface exposed is low. Thus it frequently becomes difficult to provide sufficient surface for adequate capacity. The higher the temperature of the water, the lower the relative humidity of the air; the greater the velocity over the surface, the greater is the rate of humidification. The evaporative pan type of humidification limits the water wastage and is usually supplied with water through a float valve. Due to the collection of salts in this evaporating pan such humidifying systems require occasional drainage and cleaning.

Other methods of humidification attempted in air conditioning units are through the use of wetted fabrics, porous earthenware plates, or other capillary surfaces. These methods rely upon the capillary absorption of the moisture up from the liquid level into the portion exposed to the air. They have a tendency to lose their effectiveness due to the resulting deposit of mineral salts at the evaporating surfaces thereby clogging the pores and reducing the contact of the air with the water. Also they frequently become foul and often support bacterial growth.

The capacities of any of the above methods of humidification are affected by the temperature and relative humidity of the air passing through the humidifier. The higher the temperature, the lower the relative humidity, the greater will be the rate of humidification.

Cooling and Dehumidification

Those units that employ recirculated water sprays will undoubtedly use such sprays as their means of cooling and dehumidification by

furnishing refrigeration to the water in circulation. Occasionally where an adequate source of cold well water is available, this may be used as a direct spray and run to waste.

Other methods of dehumidification accomplished by direct contact with the transfer medium are by means of the so-called *adsorption* and *absorption* systems. (See Chapter 2 on Refrigeration.) It must be recognized that these methods of dehumidification do not in themselves provide cooling. The substance removes the water vapor from the air thereby heating it. This highly dehumidified air may then be cooled either by partial rehumidification or by direct contact with a cooling medium of cold water or direct expansion refrigerant. There are now on the market solid adsorbents such as silica gel, and activated alumina, and liquid adsorbents such as calcium chloride in solution and lithium chloride which is one of the most promising examples.

Finally the most common direct means of cooling and dehumidification is through the use of ice. In such units the ice is brought into as intimate contact as possible with the air handled. Provision is made for the removal of the moisture as rapidly as it is formed from the melting of the ice. One method employed is to mount the ice on metal grids and then draw the air over the cold grids as well as over the projecting pieces of ice formed between the grids. Of course, ice is also used to cool water which is circulated through the sprays.

In conditioning units, the use of surface cooling is probably more common than direct spray or other direct transfer means. The type of surface employed may, of course, be cast or fabricated from tubes. In present-day practice finned tubes or plate fins through which tubes are passed form the most generally used cooling surface. The detailed fabrication of this surface and the arrangement of the tubes will depend largely upon the type of refrigerant for which it is intended.

The simplest construction is where chilled water or brine is used as the refrigerating medium. With direct expansion refrigerant it is usually necessary to provide a special arrangement of headers so that proper distribution of refrigerant through all the surface is obtained. In some cases, ordinary brine coils can be used when operated as a flooded refrigerant system. In some units a combination of a direct spray and a refrigerant surface is used, the spray being directed against the surface. Such systems claim the advantage of air washing together with the maintenance of a clean and effective cooling coil.

It should be noted that when surface coolers are used, adequate protection in the form of filters or at least lint screens are necessary to prevent fouling of the surface from the air borne dirt. Surface not so protected frequently becomes completely matted with lint, grease, and similar dirt.

The sources of refrigeration used with these surface type conditioning units are discussed in Chapter 2 on Refrigeration. However, they may be divided into the following groups:

1. Direct expansion refrigerant in which the liquid refrigerant is evaporated within the coils of the unit. The vapor from these coils may be recompressed in centrifugal, rotary, or reciprocating type compressors, and the refrigerant again returned to the evaporator coil.

2. Indirect refrigeration by means of:
 - a. Cold well water.
 - b. Cold city water.
 - c. Artificial refrigerated water provided by direct expansion of refrigerant in a water cooler, direct steam jet refrigeration, or by the melting of ice.

It should be recognized that direct expansion of refrigerant within the coils of a unit conditioner, represents a certain hazard unless the refrigerant is of the non-irritating, non-inflammable type. Therefore, in many cities rather stringent codes have been set up limiting the use of such direct expansion unit conditioners. In the case of hospitals, or places where a large number of people are present, direct expansion is prohibited and the so-called *indirect* systems are required. In these systems the refrigerant is used to cool a brine which is then circulated through the cooling coil.

Filtering—Air Cleaning

A variety of methods are employed as a means of controlling air purity. In unit systems where filtering alone is considered satisfactory, the degree of filtering varies widely and in proportion to the actual needs. If the air is chiefly recirculated with but little outside air used for ventilation, filtering requirements are largely limited to keeping the coils in a clean and operable condition. Thus such units are frequently furnished with simple lint screens of low resistance and formed of moderately close meshed wire. Where outside air is used for ventilation, more complete filtering of dust particles is necessary and for this purpose, there are a large number of filters available on the market. Some of these filters are of the so-called *throw-away* type, constructed of inexpensive material so that when they become dirty or clogged they may be thrown away and replaced with new ones. All of these filtering methods are described in detail in Chapter 16.

A number of air conditioning units are being offered for the relief of hay fever sufferers and render relief in this disease both by maintaining proper air conditions and by removing most of the air borne pollens from the ventilated air. Such filters, however, must be designed for this purpose and must have an exceptionally high efficiency.

The degree of filtration and its efficiency depends entirely upon the design of the unit or the unitary system. For high efficiency of air cleaning, large and expensive filtering area is required. Small filters mean either low efficiency of cleaning or else high resistance with increased fan power or decreased air capacity.

Ventilation

Inasmuch as air purity is one of the factors that constitutes true air conditioning, ventilation or the introduction of outside air is an essential part of any air conditioning unit or system. While a unit that recirculates all its air capacity is still considered an air conditioning unit, the better type system provides for the introduction of a certain proportion of out-door air. In some instances one of several units may operate entirely on outside air, while in other cases only a portion of the air handled by the unit is drawn from out of doors. In such cases a damper is provided either

in the unit or in the duct connections for controlling the proportion of out-door air.

Location of Unit Conditioners

The characteristics of the conditioned space, the building construction, the type of system employed, and the source of duct connections, power, piping, and refrigeration influence directly the location of the unit conditioners. Primarily a unit conditioner is either of the portable or fixed type. The portable unit is usually a simple summer conditioner either with or without ventilation, but includes the refrigerating compressor. If this compressor is of the air-cooled type, the unit must be located adjacent to a window or other source of outside air. On the other hand, if it is of the water cooled type, its location should be made convenient to sources of water and drainage. Thus portable units are almost invariably located within the conditioned space.

Non-portable units, or units of fixed location may or may not be located within the occupied space. Naturally units that are located in such spaces must be built with decorative cases in order to harmonize with the surroundings. Such units are normally of comparatively small capacity varying from a fraction of a ton up to as much as five or six tons. Many conditioning units and particularly those of the larger sizes are located externally to the occupied and conditioned space and are connected thereto by means of delivery and return ducts. Such an arrangement permits the location of the conditioning unit convenient to either the sources of refrigeration or outside air. It frequently permits the use of the basement or of space less valuable than that on the level or floor of the occupied zone. Oftentimes the same type of unit may find application in an exposed position for one job and in a concealed location for another. Thus it can be seen that it is not possible to define a unit merely on the basis of its location. Frequently conditioning units are built into the structure or into the architectural design of a room so that they are entirely concealed except for the discharge and return grilles or openings which are designed so as to correspond to the decorative scheme of the room.

Air Distribution

With portable units or units exposed within the conditioned space, the distribution is usually through grilles or louvres built entirely into the equipment. The discharge of the air from this unit, in general, should be upward immediately at the conditioner, with sufficient horizontal component to carry it to the most remote point in the room. Such a distribution permits the cool air to drop slowly over the entire zone and return to the inlet of the unit below the breathing line and along the floor. The location of doorways, air vents, and heat exposed walls should be carefully observed, as they have a marked effect on the direction of the air flow and on its uniformity of temperature. Velocities below the breathing line should be kept low, normally not over 40 to 70 fpm. Such velocities are usually best checked by a Kata thermometer rather than by means of an anemometer.

With the suspended type of unit conditioners located within the conditioned space, sufficient outlet air velocity should be provided to give

adequate induction and mixing with the room air thereby preventing the immediate dropping of the air stream and resulting objectionable cold drafts. This is one of the limitations to the size of such suspended units inasmuch as it is difficult to distribute from one point a large refrigeration tonnage without cold drafts.

Where the units are located without the conditioned space, air distribution is more frequently provided through multiple outlets located in ducts from the conditioning unit. The location of these outlets is quite critical and is influenced both by the building construction, economies of connections and by the distribution of load. Normally with a cooling system, it is desirable to blow the conditioned air toward the exposed walls or windows. This delivers the cool air to the point where it is most needed. These problems are discussed in considerable detail in Chapter 19.

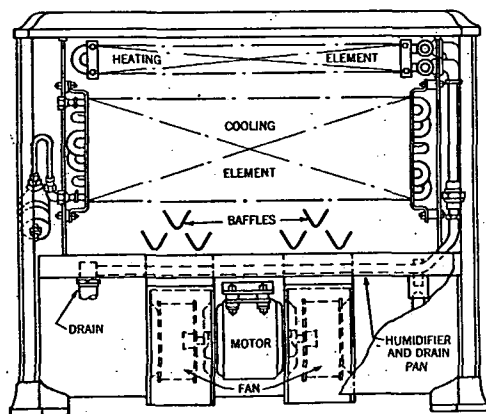


FIG. 10. FLOOR TYPE UNIT CONDITIONER FOR HEATING AND COOLING

There are a wide variety of outlet types used. Most of these have fixed delivery characteristics, thus requiring careful consideration in their location. Some types of outlets are now available with adjustable vanes thereby permitting some alteration in the delivery of the air stream after installation. This frequently eliminates objectionable down drafts resulting from the impingement of the air stream against posts, pillars, lighting fixtures, and beams.

Construction of Apparatus

The types and designs of air conditioning units in production and proposed are legion. New designs are constantly appearing, with new improvements, greater capacities, wider range of application and superior construction. It will be impossible to cover in this chapter the many types of construction on the market. Illustrations of current makes and models will be found in the *Catalog Data Section*. A few typical designs of conditioning units will be described in detail.

Fig. 10 shows an all-year floor type conditioning unit for exposed location and with direct expansion coil supplied with refrigerant from

a remotely located compressor. A cooling coil for use with chilled water can be substituted for the direct expansion coil indicated. The fans below the separate cooling and heating elements deliver the air against deflectors thereby obtaining distribution across the face of the element and preventing condensate from dripping down into the fans. The plate upon which the fans are mounted serves as the drip pan from which the water is conducted to the drain. Separate elements are used for heating and for cooling. Thus this unit may be used automatically for heating and cooling without manual control. When the unit is used for summer conditioning only, the heating coil may be omitted for the installation. The illustration indicates an evaporative type humidifier and drain pan.

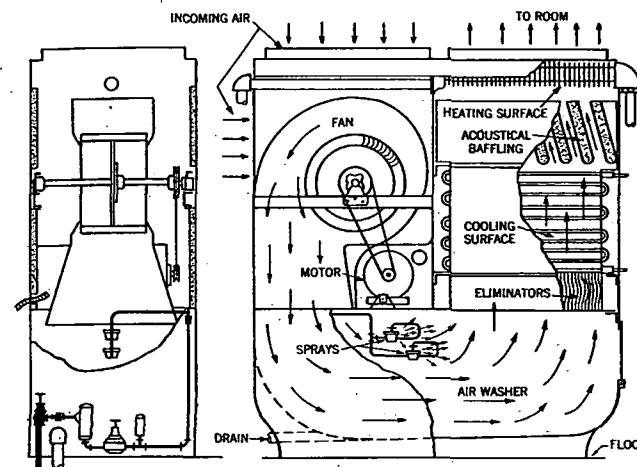


FIG. 11. CONDITIONING UNIT WITH TOP INLET AND OUTLET

Other units are on the market in which a target spray humidifier is substituted for the evaporative type thereby providing a unit for summer cooling and dehumidification, at the same time supplying humidification in winter for application in rooms with other existing heat sources.

Still another unit is available in which the fans are mounted at the top of the unit delivering directly through a grille and drawing their air supply through the cooling and heating coils. Other variations in proportion and details of construction of this general arrangement are common. With this type of unit conditioner, ventilation is usually provided by means of a separate duct connected to the inlet of the unit.

An entirely different arrangement shown in Fig. 11 places both the air inlet and the discharge at the top of the unit. The fan at one side discharges the air downward to the bottom where it turns and passes horizontally through an atomizing spray air washer. The path then continues upward through eliminators, a cooling surface and a heating surface before it leaves the unit. With steam or hot water connected to the heating element, tempered water to the sprays and refrigerated water to

the cooling element, this unit gives controlled temperature, humidity, air cleaning, and air movement in both summer and winter. Air washing may be connected in summer, or in intermediate season to remove room odors. Excess water is run to waste. Acoustical treatment of the housing and outlet baffles permits installation where noise requirements are exacting.

One of the recently developed units, designed particularly for low cost installation is shown in Fig. 12. Twin fans with wheels mounted on extensions of the motor shaft take the air from the floor, send it downward

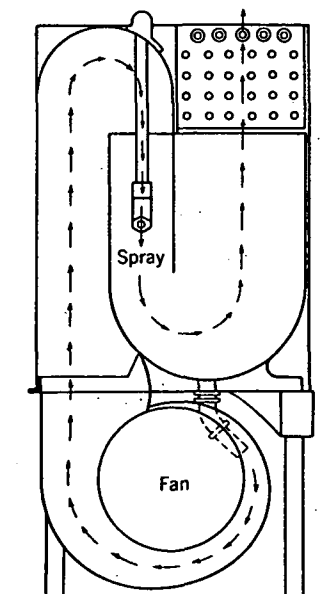


FIG. 12. SIMPLE SPRAY AND COIL UNIT CONDITIONER

through a passage containing a water spray and then upward through a double set of coils for cooling and heating, the air leaving the cabinet through a top grille. The spray nozzles are supplied with city water, the excess collecting in the air reversal chamber and running to the drain. The cooling coil uses water either from city mains or from other low temperature sources, or a special direct expansion coil may be provided. This unit like others may be equipped with automatic controls.

Fig. 13 illustrates a common type of suspended type unit conditioner for exposed location utilizing a propeller type fan and suitable for summer conditioning only. Such units are equipped with either a direct expansion coil or one for chilled water or brine circulation. The outer cabinet is commonly of wood-grained steel or baked enamel and is insulated from the cool air chamber to prevent external condensation. The drip from the coil is collected in an insulated drip pan and carried to a drain. The inlet to the unit is provided with a lint screen to protect the cooling

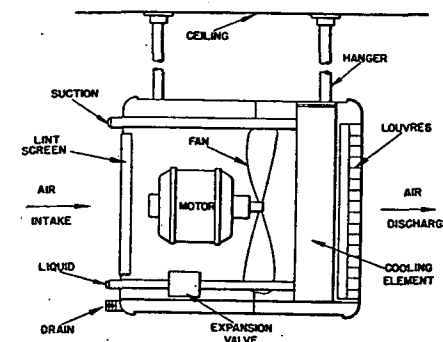


FIG. 13. SUSPENDED PROPELLER FAN TYPE UNIT CONDITIONER

surface. Such units are normally used for recirculation only but may be connected for ventilation through short full-size ducts. Similar units are available with twin housed fans of the same general construction, although usually such fans draw the air instead of blow it through the coils.

A self-contained completely portable unit conditioner is illustrated in Fig. 14. For the operation of this unit, it is only necessary that it be located adjacent to a window or shaft to which air connections can be made and to plug in the motors to a convenient light socket. In this unit, the conditioned air enters on the side, passing through a grille, filter, and cooling coil and is delivered vertically to the room through a special motor and fan assembly. Refrigeration is furnished by a reciprocating compressor driven from a motor located in the base. This compressor utilizes an air cooled condenser. Air is drawn into the base by a fan mounted on the compressor motor, so arranged that the air passes through the refrigeration condenser and is again discharged out through the window connection. A novel feature of this design is that the condensate

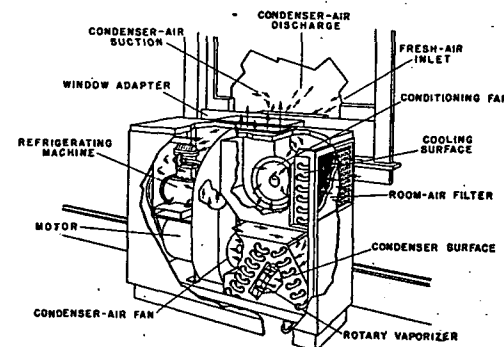


FIG. 14. PORTABLE SELF-CONTAINED AIR COOLED CONDITIONING UNIT

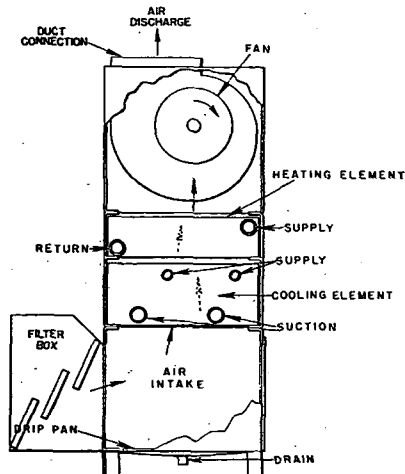


FIG. 15. VERTICAL REMOTE TYPE ALL-YEAR-ROUND CONDITIONING UNIT

from the cooling coil is sprayed over the condenser surface and there vaporized, thus eliminating the need for drain connections. One advantage of this type of conditioning unit is that it may be removed from the occupied space during the winter season when cooling is not needed.

Another type of portable unit for occupied space locations differs from the former in that the compressor is water cooled and a connection to water and drain must be provided in addition to the electric connection. Water and drain lines are carried in a composite hose, especially built for this purpose and connections are usually made to a nearby washbowl. In order to reduce the starting load, one model has two separate motors brought on to the line at delayed intervals thereby decreasing the initial

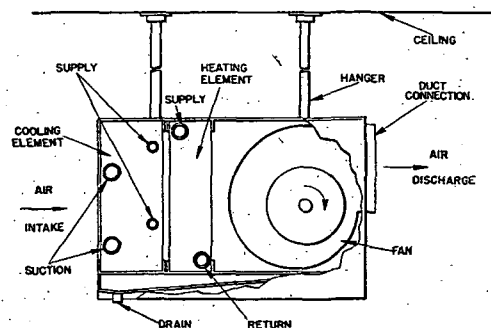


FIG. 16. HORIZONTAL REMOTE TYPE ALL-YEAR-ROUND CONDITIONING UNIT

line surge and reducing light flicker. These water cooled units either eliminate or reduce the need for out-door air connections. Due to the necessity of water and drain connections they are not as portable as the air cooled type.

Remotely located conditioning units vary widely in details of construction. Fig. 15 and 16 indicate one type built in sections thereby permitting interchangeability of application with a minimum change in parts. The vertical unit shown in Fig. 15 consists of a fan section, housing one or more fans, mounted on a coil section in which are located a heating coil and a cooling coil, which may be built for either direct expansion refrigerant, chilled water, or brine. These two sections are supported on a third or drip-pan section. The distributing duct system is attached to the fan outlets and returned fresh air connections are made to the drip pan. A filter box is illustrated attached to the drip pan section. By

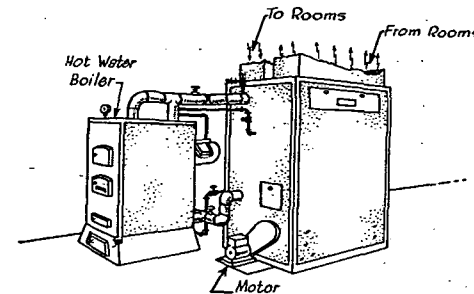


FIG. 17. RESIDENTIAL CONDITIONING UNIT WITH HOT WATER BOILER

eliminating the vertical type drip pan and substituting a horizontal drip pan, this unit is converted into a horizontal suspended type conditioning unit for connection to duct work, both with and without filters.

The previous figures have largely confined themselves to the illustration of all-year-round or summer conditioning units for office or commercial application. The smaller conditioners have, of course, been applicable to residences. There remains a field of unit conditioners primarily adaptable to residence work. They are in general the outgrowth or adaptation of mechanical warm air systems to conditioning, which are covered in Chapter 23. However, the following illustrations will cover details not included in that chapter.

Fig. 17 illustrates a conditioning unit attached to a hot water or steam boiler and as shown may furnish winter conditioning or, by connecting a cooling coil to a source of refrigeration, will supply year-round conditioning. Such a system lends itself particularly to the so-called *split* system from which a portion of the residence may be conditioned summer and winter, while the garage or servants' quarters will have only heating in winter, the heat being furnished directly from the boiler and not through the conditioning unit.

Fig. 18 shows a gas fired furnace unit equipped with an air washer, and represents a winter conditioning unit furnishing filtering, heating, and

humidification. If the air washer is supplied with refrigerated water, this unit would become a year-round conditioner.

For fuel oil the equipment shown in Fig. 19 can be installed to obtain warm, filtered, and humidified air. An oil burner and heat exchanger provide the heat. A cooling section may be inserted between the fan and the heat interchanger, cold water, or refrigerant being circulated through the cooling element. For thermostatic control, a room thermostat is provided to start the oil burner when the temperature falls, the rising temperature in the heat interchanger causes the thermostat to start the fan. As soon as the temperature in the house rises to normal, the room thermostat shuts down the oil burner and operates the thermostat controlling the fans.

BASIS OF RATING

In the past, the unit air conditioning industry has been handicapped by the lack of any standard method of rating. Recently a tentative code has been prepared defining a complete rating method⁶.

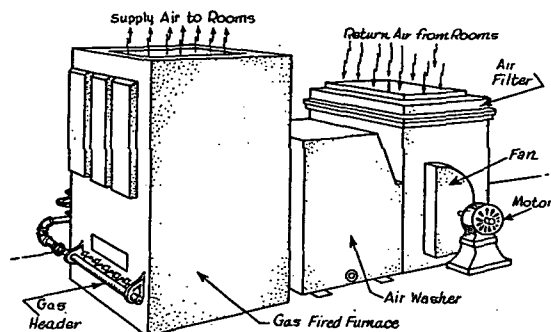


FIG. 18. GAS FIRED FURNACE UNIT

On the basis of this new code, air conditioners are to be classified primarily as *free delivery type* and *pressure type*, where delivery will be measured in cfm standard air at specified fan speed. Pressure type units will specify the delivery against various total fan static pressures. Cooling capacity will be expressed in total Btu per hour and this total will be subdivided into sensible heat cooling effect and dehumidification or latent heat effect. Cooling capacities will be given on entering air temperatures of 85 F, 50 per cent relative humidity with 40 F refrigerant temperature for comfort conditions and 45 F, 85 per cent relative humidity with 30 F refrigerant temperature for commercial applications.

The duty for heating surfaces will be specified in Btu per hour for 70 F entering air temperature based on two pounds gage steam pressure or 180 F entering water temperature with 20 F drop. Humidification will be specified in pounds of water evaporated per hour at 70 F and 30 per cent

⁶Code prepared by Joint Committee of American Society of Refrigerating Engineers, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Refrigerating Machinery Association, National Electric Manufacturers Association and Air Conditioning Manufacturers Association.

relative humidity entering air conditions. The *Catalog Data Section* gives ratings of current models offered by leading manufacturers.

METHODS OF CALCULATING CAPACITIES

The methods of calculating heating and cooling loads for conditioning units are similar to those described under Chapters 7 and 8. Certain manufacturers have adopted simplified and approximate methods which through experience they have found applicable to unitary equipment. Such methods involve certain averaging approximations which are suitable for estimating purposes but many of which require rechecking on a more accurate basis before actual installations are made.

The greatest error in calculating cooling loads is apt to be introduced in the failure to appreciate the magnitude of the latent heat loads and the

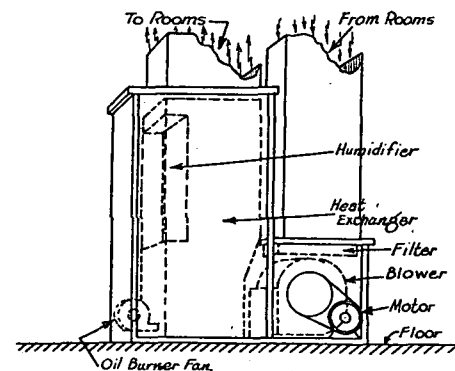


FIG. 19. OIL FIRED UNIT

relationship between the latent heat removal and the refrigerant temperature. It is extremely important that the proper balance be obtained between the refrigerant temperature, the conditioning unit surface and the conditions to be maintained within the occupied space. Unsatisfactory conditions often result through the attempt to apply unit conditioners with a source of refrigeration which gives too high a surface temperature. Such conditions may be obtained when well water of too high a temperature is used or when a direct expansion evaporator is connected to a refrigeration compressor of inadequate size. The high refrigerant temperature even though it may give adequate dry-bulb temperatures, due to over-size conditioning units, will not give proper humidity control due to its inability to furnish sufficient dehumidification.

The use of surface coolers with widely extended fins may lead to similar results since the large ratio of extended surfaces gives an average surface temperature considerably above the refrigerant temperature within the tubes. All these factors must be kept in mind when making the selection of equipment. It is furthermore vital that the calculation of a cooling load be based on an accurate survey before the selection of the equipment is made.

COSTS

Due to the rapid development of the air conditioning industry and the great progress that is being made each year, it is impossible to give any cost figures that will be of value. There are, however, certain factors that influence the cost of unit air conditioner installations.

1. Since the cost of the total job involves material cost plus installation labor and since through the use of unitary equipment, material costs can be kept to a minimum, every effort should be made to simplify installation.
2. Self-contained units in the small sizes now available, probably represent the lowest cost individual installations. They have, however, their limitations.
3. The floor type all-year-round unit conditioners for the occupied space with a remotely controlled compressor, heating sources being either the existing heat system or steam connections to the unit, probably afford the lowest cost all-year-round service for most individual rooms. This is particularly true in the case of residences. With offices, this will probably be true if the compressor can be located immediately adjacent to the conditioned space, as for example in a closet or nearby store room. The expenses increase rapidly as the distance from the unit increases.
4. For multiple rooms or offices, the remotely located conditioner with connecting ducts probably represents the most economical installation. Such installations are also particularly adaptable to stores, residences and small commercial installations.

Costs of operation vary widely depending entirely upon the cost of power and water. Water costs in the larger installations are being materially reduced through the use of cooling towers and special types of condensers. The normal expense of operating the cooling system is considerably in excess of that of winter heating both as to the first cost and as to operation. Consequently, the more rapid growth of air conditioning has been along commercial lines where it has represented an actual profitable investment resulting in increased business returns rather than along the lines of residential comfort cooling where it still represents a luxury in comfort.

MISCELLANEOUS UNITARY EQUIPMENT

There are a number of units available which were not covered in the previous discussion that accomplish only one or two of the functions of air conditioning.

Attic Fans

Attic fans, used during the warm months of the year to draw large volumes of outside air through a house, offer a means of using the comparative coolness of outside evening and night air to bring down the inside temperature of a house.

Because the low static pressures involved are usually less than $\frac{1}{8}$ in. of water, disc or propeller fans are generally used instead of the blower or housed types. The fans should have quiet operating characteristics, and they should be capable of giving about thirty air changes per hour. The two general types of attic fan installations in common use are:

Open attic fans, in which the fan is installed in a gable or dormer and one or more grilles are provided in the ceilings of the rooms below.

Fresh air, which enters the house through open windows, is drawn into the attic through the grilles, and is discharged out-of-doors by the fan. An attic stairway may be used in place of the central grille. It is essential that the roof and the attic walls be free from air leaks.

Boxed-in-fans, 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. Fresh 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.

The locations of the fan, the outlet openings, and the grilles should be chosen after consideration of the room and attic arrangement 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, 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 operating routine which will secure best results with an attic fan is an important consideration. A typical routine might require that in the late afternoon when the outdoor temperature begins to fall, the windows on the first floor and the grilles in the ceiling or the attic floor should be opened, and the second story windows should be kept closed. This will place the principal cooling effect in the living rooms. Shortly before bedtime, the first floor windows may be closed and those on the second floor opened, to transfer the cooling effect to the sleeping rooms. A time clock may shut off the fan before waking time, or the fan may be stopped manually at a later hour.

A disadvantage arising from the passing of a great amount of outside air through a house is the dust nuisance, which varies considerably in different locations. Persons suffering from allergic diseases caused by air-borne pollens will have their troubles increased with attic type coolers.

Some typical data on an attic fan installation in an average six-room house of frame construction containing 14,000 cu ft and located in the southern part of this country are:

Installation cost.....	\$75 to \$400, average \$250
Fan data.....	9000 cfm average, 280 rpm if belt driven, 570 rpm if direct connected, 500 watts input
Operating period.....	April 15 to October 15, intermittently as weather conditions demand
Power consumption.....	500 kwh per year for 8 months' operation

Humidifiers

Humidifying units may be installed as part of an air conditioning unit system, or may be installed individually to furnish additional humidity. Fig. 20 illustrates a humidifying unit for installation in connection with a warm air heating system, and as such it is located at the intake of the

furnace. The air passes through a lint filter, then through the fans and finally through an air washer or spray system. Surplus spray is eliminated and the air delivered to the hot air system. In other cases, similar spray type apparatus is used to deliver humidified air through ducts to openings beneath existing radiators in a steam heated house.

For other steam heated homes, there is a humidifying unit as illustrated in Fig. 21. This unit is normally placed at some central location on the first floor, and receives the air from the floor into fans, delivering it through a heating coil and up through a target spray atomizer. Surplus

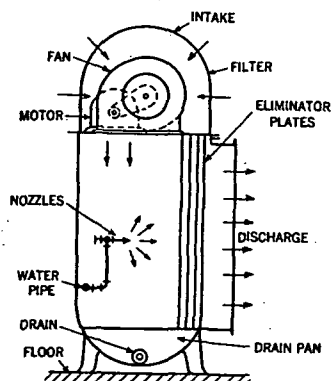


FIG. 20. HUMIDIFYING UNIT FOR WARM AIR FURNACE

moisture is removed by means of an eliminator filter and the humidified air is delivered upward through the other half of the floor grille. Since a large percentage of the heater capacity is transformed into the latent heat of humidification, this unit does not eliminate any existing steam radiation. It may also be used with hot water systems but its capacity is considerably reduced.

CONTROLS

The controls of all unitary equipment represent a vital part of their successful operation. This is particularly true in the case of conditioning equipment where a very close inter-relationship exists between the thermostatic room controls and the refrigerating unit controls.

The proper selection of controls and the proper adjustment is extremely important to prevent short cycling of compressors. Furthermore, the proper adjustment of direct expansion valve controls is likewise extremely important.

A detailed discussion of control problems is contained in Chapter 14.

PROBLEMS IN PRACTICE

1 ● Distinguish between a unit and a central type of air conditioning system.

In a unit system, the air treating apparatus consists of factory assembled equipment which is shipped substantially complete or in sections and is installed without field fabrication except for the duct connections between the equipment and the point of delivery of the air. Usually the air treating equipment is located closely adjacent to the conditioned space and serves a limited area. A central type of air conditioning system localizes the air treating equipment for the entire area at one point and involves the field assembly of a large number of individual elements. Manufacturer of the unit is responsible for the out-put and performance of the unit under rated conditions, whereas, the contractor installing the completely unitary or central type equipment is held responsible for the complete performance of the system.

2 ● Is it satisfactory to use superheated steam in unit heaters?

Superheated steam can be satisfactorily used in unit heaters provided the capacity is based on the saturated steam temperature and not on the total temperature. If unusually high superheat is used, trouble may be experienced from the excessive expansion and contraction of the heating elements.

3 ● Is it satisfactory to install one unit heater as the total load on a coal fired boiler?

Such an arrangement is impractical if the unit heater is started and stopped in keeping with the room temperature. However, if the room temperature controls the steam pressure and the unit heater is arranged to start when there is steam in the mains and to stop when there is no steam in the mains, such an installation will be satisfactory.

4 ● Will a unit heater with a slow speed fan be more quiet than one with a high speed fan?

The speed of the fan is no indication of quietness. Quietness is a function of the type, diameter, blade form, speed, and location of the fan.

5 ● Is it satisfactory to use steam at pressures less than atmospheric for unit heaters or unit ventilators?

If the air inlet temperature is above freezing, steam at any pressure may be used in the heating element of the unit heater or unit ventilator. If the inlet temperature is below freezing the heating element should be filled with steam of at least 5 lb pressure (or with a positive 5 lb pressure differential between supply and return) and the steam supply should never be throttled or the heating element may be frozen.

6 ● In general, what is the primary function of a unit ventilator?

To maintain the desired room air conditions as to temperature, air change, and air cleanliness, without drafts regardless of variations in outdoor temperature, occupancy, sun heat, and wind.

7 ● What are the usual working parts of a unit ventilator?

The fan and motor assembly, a set of heating elements, outdoor and indoor air dampers, filters (optional), outlet grille, some method of varying the outlet temperature in keeping with the room requirements, and, in the case of some unit ventilators, a method of limiting the outlet temperature to a minimum of 60 F.

8 ● Do all unit ventilators introduce a constant amount of outdoor air?

Certain types employ full recirculation except when outdoor air is obtained by throttling the steam valve on the heating element so the proportion of outdoor air to room air is varied. This is a very economical type of unit ventilator but in some communities it cannot be used because of existing laws which require that some fixed amount of outdoor air be introduced whenever the room is occupied. Certain types of units are designed to always take in a minimum quantity of air from the outside and to automatically vary this with the weather.

9 • Why are metal surface cooling elements instead of liquid spray chambers used in the design of most unit air conditioners and unit coolers?

The first cost of the surface cooling type of unit is considerably less than the cost of spray type equipment. Further, the requirements of many industrial air conditioning jobs and of all comfort cooling jobs where unit equipment is applicable can often be effectively met with the use of surface type units, with a reduction in the space required for making the installation. Where space conditions are especially limited, the cross-sectional area of the surface cooler can be reduced because the resulting increase in velocity over the coil surface increases the effectiveness of the surface, whereas an increase in velocity through a liquid spray would reduce its effectiveness.

10 • Why are air conditioning units with metal cooling surfaces not desirable for all industrial jobs?

Wherever unusually close control of relative humidity is required, a spray type unit will prove to be more satisfactory. Relative humidity control and accurate temperature control, however, can be maintained without difficulty with the use of metal surface units.

11 • Why is accurate control of relative humidity with surface coolers more or less complicated?

A surface cooler cannot add moisture to the air, and moisture is removed only when the surface temperature is below the entering dew-point temperature. Any change in condition of the entering air will result in a change in the dry-bulb depression of the leaving air. This change in entering condition requires not only a readjustment of the air volume but also a change in the coil temperature, if accurate control over the relative humidity is to be maintained.

12 • What in general are the characteristics of unit conditioner operation using surface coils?

For a constant entering dry-bulb temperature and a constant refrigerant temperature any increase in the entering wet-bulb temperature will produce a rise in the leaving dry-bulb temperature with an accompanying reduction in the wet-bulb depression of the leaving air. The sensible heat removed by the unit decreases and the latent heat increases, while the total heat removed also increases. When the dry-bulb temperature of entering air is increased, with constant refrigerant temperature and constant wet-bulb temperature of entering air, the wet-bulb depression of the leaving air increases, and since it is this depression which determines the maintained relative humidity it must be carefully considered when selecting the unit.

13 • If a drop in the dry-bulb temperature of entering air reduces the capacity of the unit, is there not danger of selecting a unit which is too small, if its selection should be based on an excessive entering dry-bulb temperature?

Yes. If the total cooling load is largely internal (such as from occupants and lights) as distinguished from the cooling load of outdoor air, and the unit is selected on the basis of a too high dry-bulb temperature of entering air, then, in the event of under capacity, it might be possible to maintain the room temperature by reducing the quantity of outdoor air. But this increases the recirculated air taken into the unit, reducing the dry-bulb temperature of entering air and, therefore, reducing the sensible heat capacity of the unit. This reduction in capacity may offset the gain obtained by reducing the amount of outdoor air taken in. Further, since the total tonnage required for any installation is equal to the total internal heat load plus the total heat removed from the outdoor air, and since the outdoor air might have a wet-bulb temperature equal to the designed wet-bulb but less than the designed dry-bulb temperature, then the sensible heat capacity of the unit will be less than that required. It follows that unit air conditioners and coolers should not be selected on a basis of the maximum possible dry-bulb temperature of entering air.

Chapter 13

RAILWAY AIR CONDITIONING

Ventilation, Heating, Cooling, Calculation of Cooling Load, Humidity Control, Temperature Control, Power Supply, Costs

THE general principles of air conditioning as applied to buildings also apply to railway cars, but due to space and weight limitations and the severity of the service, equipment designed for stationary work in buildings is seldom suitable for car installations. Equipment for railway use must be safe, reliable, compact, light in weight, accessible for inspection and repairs, as nearly automatic in operation as possible, and have low initial, operating, and maintenance costs. To properly air condition a car, ventilating, filtering, heating, cooling, humidifying, and control equipment must be provided together with an adequate power supply. Air from the interior of the car is mixed with air from the outside and passed through the air conditioning unit where it is heated or cooled, humidified or dehumidified and delivered to the interior of the car through suitable ducts and grilles.

VENTILATION

One of the important problems in connection with air conditioning of cars is that of ventilation. In non-air conditioned cars, ventilation is accomplished by exhaust fans, roof ventilators and open doors and windows. This provides an ample supply of outside air and in addition a large amount of smoke and dirt which may be excluded in an air conditioned car.

An average car contains approximately 5000 cu ft of air which is being contaminated by the occupants who are continually liberating heat, carbon dioxide, moisture, odors, and some organic matter from the breath, skin and clothing. The heat and moisture can be removed by cooling and dehumidifying, but the others can be handled only by proper ventilation. In the average car from 2000 to 2500 cfm of air should be delivered by the air conditioning unit. Some of this air may be recirculated, but a portion of it should be brought in from the outside. The amount of outside air required depends upon the type of car (dining, club, cafe), number of passengers, air temperature, humidity, smoke, and odors, and will vary from 15 to 90 per cent. This percentage of outside air should be kept as low as possible to still maintain the air in the proper condition in order to minimize the heating or cooling load.

In equipping old cars, the air is often distributed to the car from the conditioning unit through one or two side ducts, built on the outside of

monitor-roofed cars and on the inside of turtle-back or arched-roof cars. In some instances, a center duct is used and in others the air is discharged directly from the conditioning units into the upper part of the car either through the end bulkheads or from a unit placed overhead in the center of the car. For details of air distribution and duct design see Chapters 19 and 20. Care should be taken to keep the air velocity in the ducts below the point where the noise would be objectionable.

Suitable grilles should be used at duct outlets to reduce the velocity and to direct the air so as to insure thorough mixing without objectionable drafts. The incoming air should be introduced as near room temperature as possible and should have a velocity of not over 120 fpm when it reaches the passengers. The outlet grille nearest the recirculating air grille should deflect 45 deg in one direction towards the center of the car to prevent the incoming air from passing directly to the recirculating grilles. The other outlet grilles installed along the ducts should deflect a portion of the air 45 deg to the right and 45 deg to the left and to all points between.

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 wash room or lavatory, to be exhausted to the outside through a ventilator.

For compartments a supply duct outlet grille of suitable size and design should be provided and provisions made in the door or partition for removal of the recirculated air through a special grille, which will allow the air to pass from the room to the main recirculating air grille. The exhaust grille should be designed and arranged so as to obstruct the vision of passengers.

Lower berths in sleepers 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.

Recirculating air grilles are usually of the straight flow type, arranged to hold two 2 by 16 by 20 in. filters and hinged on an angle iron frame to permit easy access for cleaning and repairs. They should be located so that objectional drafts will not be created by the return air. The outside air intakes should be of ample size and provided with filters and dampers or shutters for regulating the amount of outside air.

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. Cars 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. Greater care must be exercised to secure proper circulation without more objectional drafts than when ducts are used.

Filters made of metal wool, spun glass, hemp, paper, cloth, and wire screen are in use. Some types may be cleaned, retreated and returned to service while other types are discarded when dirty.

HEATING

The majority of cars in service are heated by circulating low pressure steam or vapor through pipes located along the side walls near the floor. When an overhead air conditioning unit, using air from the outside and discharging it into the car, is installed it is necessary to provide a heating coil to warm the air during cold weather. Most of the heat required in the car can be supplied by the overhead unit but in extreme weather some heat must be supplied at the floor to keep the lower part of the car at a comfortable temperature. It is also desirable to have sufficient floor radiation to keep the car from freezing up while standing in the yard with the air conditioning unit shut off. Usually from 30 to 40 per cent of the heat required is supplied from the overhead unit and the balance from the floor heating system. The amount of heat required depends upon the type and construction of car, especially the amount and kind of insulation, outside temperature, wind velocity and direction, train speed, number of passengers, and inside temperature desired. In severe weather, -10 to -20 F, an average of approximately 200 lb of steam per car per hour is required. Pullmans will require approximately 250 lb per hour, coaches 150 to 175 lb and baggage cars 150 lb.

COOLING

Three general types of cooling or refrigerating equipment are being used with satisfactory results. These are the ice, steam jet, and mechanical compressor systems. These systems when arranged for car use, function the same as in stationary service, but must be more compact and lighter in weight. See Chapters 2 and 10 for the general principles of the various systems.

The mechanical compressor systems are divided into two general classes, the direct drive and the electro-mechanical, differing only in the method of driving the compressor. With the direct drive system, the compressor is driven from the car axle through a combination of pulleys, belts, gears, a shaft, and a speed control device. With the electro-mechanical system the compressor is driven by an electric motor. The refrigerant most commonly used in the mechanical compressor systems is Freon.

Another type of system which has possibilities of being adapted to railway air conditioning uses methylene chloride as the refrigerant, dry ice, and a small circulating pump in place of the conventional compressor. The refrigerating unit consists of a condenser built into an insulated dry ice box. The methylene chloride is liquified by the low temperature produced by the dry ice and is circulated by the pump through the evaporator coils where part of it is vaporized by the heat from the air passing over the evaporator coils. The mixture of gas and liquid passes back to the condenser where the heat is removed and the gas liquified. This system will probably not offer many possibilities until an adequate supply of dry ice can be assured at a stable and reasonable price.

A system using a compressor driven by an internal combustion engine operating on propane has also been considered for railway air conditioning. The engine, compressor, condenser, starting motor and battery are

mounted together in a single unit and supported on a track on the car under-frame on pneumatic supports. Propane fuel sufficient for several days operation is carried in three drums mounted in a rack under the car. Freon is used as the refrigerant. The unit may be used with a direct expansion air conditioning unit or it can be used with a heat exchanger with cold water cooling coils.

The capacity required in the refrigerating system depends upon a number of factors such as size and type of construction of car, thickness and kind of insulation used, amount of heat produced within the car by motors, lights, and other appliances, amount of outside air, outside air temperature and humidity, intensity of solar radiation, number of passengers, and inside temperature desired. A check of a number of cars with ice-activated systems indicates an average ice consumption, day and night, of approximately 275 lb per hour. This means an average capacity of 3.3 tons of refrigeration. It has also been observed on test that in the sunshine with an outside dry-bulb temperature of 85 to 90 F and a wet-bulb temperature of 73 to 75 F, a six ton compressor unit operates approximately 50 per cent of the time. As the average temperature during the cooling season is below 90 F the observed performance of the six ton compressor unit compares with the 3.3 tons average capacity determined for the ice systems. It was also observed on test that with temperatures above 90 F, high humidity, and in the sunshine, a six ton unit operates almost continuously. The sun load on a bright sunshiny day is about one ton. For average cars on sunshiny days, with high temperatures and humidity, from 65,000 to 80,000 Btu per hour will have to be removed from the interior of the car to maintain an inside effective temperature within the comfort zone. This means that a refrigerating capacity of from 5.5 to 7 tons will be required.

CALCULATION OF COOLING LOAD

Due to the many variables involved the calculated heat gain will be more or less of an approximation, but if a careful study is made, results sufficiently accurate for all practical purposes may be obtained. The following example illustrates a typical cooling load calculation.

Example and Solution.

Type of car	=	Arched-roof steel coach.	
Roof area	=	70.3×12.5	= 880 sq ft.
Floor area	=	70.3×9.8	= 687 sq ft.
Window area	=	$38 \times 2.5 \times 2$	= 190 sq ft.
Net side wall area	=	$(2 \times 7.5 \times 70.3) - (190)$	= 865 sq ft.
End area	=	$2 \times 7.5 \times 9.8$	= 147 sq ft.

Average roof section: $\frac{1}{16}$ in. steel sheet, 1 layer tar paper, 2 in. air space, $\frac{1}{2}$ in. hairfelt, $\frac{3}{8}$ in. rigid fiber board insulation.

Average window section: two layers of $\frac{1}{8}$ in. glass with $1\frac{1}{2}$ in. air space between.

Average floor section: 1 in. composition, $\frac{1}{16}$ in. steel, $\frac{1}{2}$ in. air space, 1 in. hairfelt, $\frac{1}{4}$ in. steel.

Average side and end section: $\frac{3}{16}$ in. steel, $\frac{1}{2}$ in. hairfelt, 3 in. air space, $\frac{1}{4}$ in. wood.

End area is taken as total area at body and bulkheads with no allowance made for glass as car is vestibuled and glass in body ends is not subjected to direct solar radiation.

Outside dry-bulb temperature	t_o	= 95 F.
Outside wet-bulb temperature	t'_o	= 75 F.

Outside effective temperature	ET_o	= 83 F.
Outside relative humidity	rh_o	= 40 per cent.
Inside dry-bulb temperature	t	= 80 F.
Inside wet-bulb temperature	t'	= 66.5 F.
Inside effective temperature	ET	= 73.5 F.
Inside relative humidity	rh	= 50 per cent.
Density of air at t	d	= 0.07356.
Moisture per pound of dry air at t_o and t'_o	G_o	= 99 grains.
Moisture per pound of dry air at t and t'	G	= 77 grains.
Moisture per pound of dry air to be removed M	$= \frac{G_o - G}{7000} = \frac{99 - 77}{7000} = 0.00314$ pounds.	
Evaporator condensate temperature		= 53 F.
Latent heat of water at 53 F	L	= 1060 Btu per pound.
Specific heat of air	C_p	= 0.241.
Number of seated passengers	P_s	= 68.
Number of attendants	P_a	= 1
Heat given off by each passenger		= 400 Btu per hour.
Heat given off by attendant		= 650 Btu per hour.
Outside air introduced	Q	= 500 cfm.
Roof temperature in sun	= 40 F above ambient.	
Wall temperature in sun	= 25 F above ambient.	
Sun. 15 deg from Zenith.		

Solar heat = 4.75 Btu per minute per square foot. 30 per cent of heat passing through glass is reflected and reradiated, 70 per cent remaining. 80 per cent of heat at surface passes through glass. Then $4.75 \times \text{Sine } 15 \text{ deg} \times 0.8 \times 0.7 = 0.7$ Btu per square foot per minute passes through glass. Evaporator motor = $\frac{1}{2}$ hp. Evaporator motor efficiency = 65 per cent. The total heat gain will include the gains from leakage through roof, floor, side walls, end walls, windows, sensible and latent heats from outside air, heat from occupants, heat from evaporator fan motors and solar radiation.

Calculation of Transmission Coefficients. (See Chapter 5.)

Roof coefficient = U_r = Btu per hour per square foot per degree Fahrenheit.

Outside air film	$f_o = 3.5$	$1/f$	= 0.286
$\frac{1}{16}$ in. steel	$k = 308$	$\frac{x}{k} = \frac{0.0625}{308}$	= 0.0002
2 in. air space	$a = 1$	$1/a = 1.0000$	
$\frac{1}{2}$ in. hairfelt	$k = 0.25$	$\frac{x}{k} = \frac{0.5}{0.25}$	= 2.0000
$\frac{1}{8}$ in. rigid fiber board insulation	$k = 0.33$	$\frac{x}{k} = \frac{0.125}{0.33}$	= 0.3800
Inside air film	$f_i = 1.65$	$1/f$	= 0.608
Total			= 4.4287

$$U_r = \frac{1}{4.4287} = 0.223 \text{ Btu per hour per square foot per Fahrenheit.}$$

Side and end wall coefficient = U_s = Btu per hour per square foot per degree Fahrenheit.

Outside air film			= 0.2860
$\frac{3}{16}$ in. steel	$k = 3.08$	$\frac{x}{k} = \frac{0.1875}{308}$	= 0.0006
$\frac{1}{2}$ in. hairfelt	$k = 0.25$	$\frac{x}{k} = \frac{0.5}{0.25}$	= 2.0000
3 in. air space	$a = 1$	$1/a$	= 1.0000

$\frac{3}{4}$ in. wood	$k = 0.8$	$\frac{x}{k} = \frac{0.75}{0.8}$	= 0.9350
Inside air film			= 0.6080
Total			= 4.8296

$$U_s = \frac{1}{4.8296} = 0.207 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

Window Coefficient = U_w = Btu per hour per square foot per degree Fahrenheit.

Outside air film			= 0.286
$\frac{1}{8}$ in. glass	$k = 1.13$	$1/k$	= 0.889
$1\frac{1}{2}$ in. air space	$a = 1.06$	$1/a$	= 0.940
$\frac{1}{8}$ in. glass			= 0.889
Inside air film			= 0.608
Total			= 3.612

$$U_w = \frac{1}{3.612} = 0.278 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

Floor Coefficient = U_f = Btu per hour per square foot per degree Fahrenheit.

Outside air film			= 0.2860
$\frac{1}{16}$ in. steel			= 0.0002
$\frac{1}{2}$ in. air space	$a = 1.2$	$1/a = 1/1.2$	= 0.8320
1 in. hairfelt	$k = 0.25$	$1/k$	= 4.0000
$\frac{1}{16}$ in. steel			= 0.0002
1 in. composition	$k = 6$	$1/k$	= 0.1666
Inside air film			= 0.6080
Total			= 5.8930

$$U_f = \frac{1}{5.893} = 0.170 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

Heat Leakage:

Through roof	= $880 \times 0.232 = 204$
side walls	= $865 \times 0.207 = 180$
floor	= $687 \times 0.170 = 117$
windows	= $190 \times 0.278 = 53$
end walls	= $147 \times 0.207 = 31$

585 Btu per hour per degree Fahrenheit.

Total heat gain = $585 (t_o - t) = 585 \times 15 = 8775$ Btu per hour.

Sensible heat gain from make-up air = $Q \times 60 \times d \times C_p \times (t_o - t)$
 = $500 \times 60 \times 0.07356 \times 0.241 \times 15 = 7950$ Btu per hour.

Latent heat gain from outside air = $L \times 60 \times Q \times d \times M$
 = $1060 \times 60 \times 500 \times 0.07356 \times 0.00314 = 7350$ Btu per hour.

Heat gain from passengers = $(68 \times 400) + (1 \times 650) = 27,850$ Btu per hour.

Heat gain from evaporator motor = $\frac{0.50}{0.65} \times 0.746 \times 3415 = 1960$ Btu per hour.

Heat gain from solar radiation:

Heat gain through roof = $U_r \times A_r \times 40 = 0.232 \times 880 \times 40 = 8150$ Btu per hour.
 Heat gain through side wall = $0.207 \times 433 \times 25 = 2230$ Btu per hour.

Heat gain through window = $0.7 \times \frac{190}{2} \times 60 = 4000$ Btu per hour.

Total heat gain from solar radiation = 14380 Btu per hour.

Summary of Total Heat Gains:

Leakage	8,775
Sensible heat from outside air	7,950
Latent heat from outside air	7,350
Heat from passengers	27,850
Heat from evaporator fan motors	1,960
Solar radiation	14,380
Total	68,265 Btu per hour.

This would require a refrigerating capacity of $\frac{68,265}{12,000} = 5.7$ tons.

HUMIDITY CONTROL

The temperature to be maintained in a car depends upon the outside temperature and the desired humidity 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 on 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 compressor is bypassed.

During the heating season humidification is desirable from a comfort standpoint, but the proper amount of humidification required would doubtless cause the windows to become frosted. A steam or water spray controlled by a humidistat will provide the necessary moisture if humidification is desired.

TEMPERATURE CONTROL

The control of the air conditioning equipment should be simple but at the same time as nearly automatic as possible. The use of a centralized control panel for all control switches, fuses, relay, etc., will simplify the installation and operation. Generally, separate thermostats are used for heating and cooling control. The best location for the thermostats depend upon the car layout and method of air distribution and can best be determined for any particular type of car and equipment by careful consideration of the several factors involved. The floor heat thermostats are usually located near the floor. The overhead heat and cooling thermostats are placed in the upper part of the car, sometimes in the air ducts. All thermostats should be located so that the air can circulate freely around them. Maintenance of uniform comfort conditions for cooling,

floor and overhead heating, has been satisfactory with provisions for high, medium, and low thermostat settings and in some cases, two settings have been satisfactory for cooling. In many cars the following points have been found to be satisfactory, 71 and 76 F for cooling, 60, 71 and 74 F for floor and overhead heating.

The heating and refrigerating equipment should be interlocked so that they cannot both operate at the same time. While heating, the control should be so arranged that in case of a steam failure the blower fan will stop or the outside air intake closed to prevent cool air from being introduced into the conditioned space.

POWER SUPPLY

One of the most important problems to be solved in connection with railway car air conditioning is that of power supply. The majority of cars now in service are electric lighted and equipped with fans. Power is furnished by storage batteries and axle generators of from 2 to 5 kw capacity. When air conditioning is installed the electrical load is increased approximately 1 kw for ice systems, 3.5 kw for steam systems and 10 kw for electro-mechanical compressor systems. Steam ejector systems require approximately 230 lb of steam per car per hour for a six ton unit. All of this power as well as the power required to move the extra weight must be supplied by the locomotive enroute and if a number of cars in the train are air conditioned, the effect on train performance should not be overlooked.

In the case of mechanical compressor systems the power is taken from the locomotive draw bar, unless the compressor is driven by an internal combustion engine, while with the steam system most of the power is furnished in the form of steam which is a load on the locomotive boiler but not on the cylinders. The ice system takes the smallest amount of power from the locomotive since no power is required to produce refrigeration. The demand for power for cooling comes, however, at the time of year when steam for heating is not required and the demand for lighting is at a minimum.

Inasmuch as axle generators and mechanical driven compressors will not operate below a certain cut in speed and will not carry full load until a speed considerably higher than cut in speed is reached, it is quite apparent that the characteristics of the run such as schedule speed, top speed, number of stops, length of stops, percentage of time of operation at slow speed, length of run, etc., must be considered in the selection of equipment. Fig. 1 shows the tractive resistance of a 75 ton car with six wheel trucks without an axle generator, and with a 4 kw generator and for the same car with an increase in weight of five tons and with a 20 kw axle generator load.

The direct-drive compressor system has only the friction of the drive during starting and at low speeds. After the compressor cuts in the load increases with the speed. The compressor output increases with the speed until maximum output is reached and then the drive efficiency decreases as the speed continues to increase so that the power input to the drive continues to increase and the greater power requirements come at the higher speeds.

Consideration must also be given to the power requirements for refrigeration while the car is at a standstill or running at slow speeds. The electrical energy required for the ice activated and steam systems is easily supplied from the storage battery. With the ice system a supply of ice is necessary and steam for the steam system can be supplied from the locomotive or from a stationery plant. The majority of the electro-mechanical systems are equipped with A.C.—D.C. motors. While standing in the yards and stations the A.C. motor is connected to a 220 volt, 3 phase circuit. The majority of these equipments are so arranged that while operating on A.C. power the D.C. motor may be used as a generator for battery charging. If an auxiliary circuit is not available the D.C. compressor motor may be operated from the storage battery.

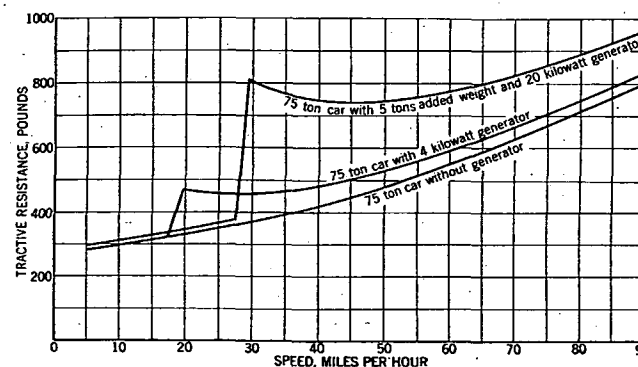


FIG. 1. TRACTIVE RESISTANCE OF 75 TON PASSENGER CAR WITH SIX WHEEL TRUCKS

The direct drive mechanical compressor systems are also equipped with A.C. motors for operation from auxiliary circuits. These equipments can only be operated when connected to the auxiliary circuit or while the train is running above the cut in speed of the drive.

COSTS

It is rather difficult to calculate the costs of car air conditioning due to the many variables involved. A study of reports covering a large number of cars of all kinds, operating under various conditions in all parts of the country, indicate a great variation in costs and weights for all systems. The following tabulation shows the average values for all cars considered.

	ICE	STEAM	MECH. COMP. ELECTRO	DIRECT DRIVE
Av. cost.....	\$4000.00	\$8200.00	\$8200.00	\$8200.00
Av. weight.....	8500 lb	11300 lb	9600 lb	8600 lb
Av. maintenance per 1000 car-miles.....	\$2.00	\$2.20	\$3.85	\$3.22
Av. ice capacity.....	4500 lb			
Av. ice consumption per hour.....	275 lb			

Based on an average cost of coal, water and locomotive lubrication of \$0.001133 per pound of fuel burned and the following assumptions: generator and drive efficiency 80 per cent, locomotive mechanical efficiency (cylinders to tender drawbar) 90 per cent, 3.5 lb coal per cylinder horsepower and 6 lb of water per pound of coal, the cost of generating 1 kw of electrical energy would be \$0.00738, and 100 lb of steam \$0.01888. This is based on the assumption that the cost of coal, water and locomotive lubrication would increase in direct proportion to the amount of coal burned. As the cost of locomotive lubrication would not increase in direct proportion to the amount of coal burned, the extra cost of lubrication included in the figure should offset the increase in cost of locomotive maintenance due to the additional load. Using an average capa-

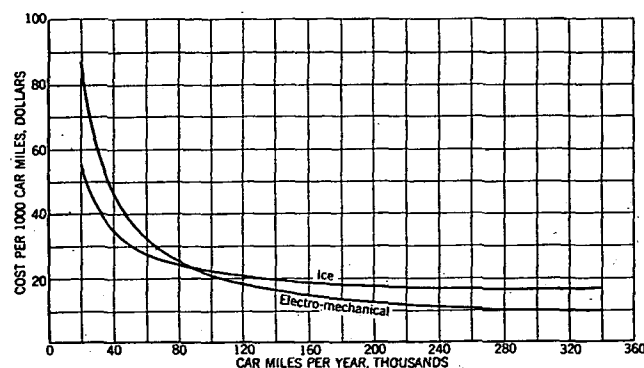


FIG. 2. COSTS RAILWAY PASSENGER CAR AIR CONDITIONING

city of 3.3 tons, the cost per hour of operation would be approximately \$0.04 for steam and mechanical compressor systems and \$0.55 for ice, using a price of \$4.00 per ton for ice.

These figures are based on the assumption that the motor driven compressor is receiving power direct from the generator and that the efficiency of the direct drive is the same as the combined efficiency of the axle generator, drive, and motor. Of course part of the current for the electric drive would be taken from the battery and therefore the battery efficiency would have to be taken into consideration. Likewise the lower efficiency should be considered of the direct drive at the higher speeds and it would be necessary to know the characteristics of the run under consideration to make an exact comparison.

For a given run the total kilowatt hours required for the electro-mechanical system would

$$= \frac{E_g \times I_g \times T_g}{1000} + \frac{E_b \times I_b \times T_b}{e_b \times 1000}$$

where

E_g = voltage with generator cut in.

E_b = voltage with generator cut out.

I_g = current with generator cut in.
 I_b = current with generator cut out.
 T_g = time generator cut in.
 T_b = time generator cut out.
 e_b = battery efficiency.

$$\text{Cylinder horsepower hour} = \frac{\text{kilowatt hours} \times 1000}{e_g \times e_d \times e_1 \times 746}$$

where

e_g = generator efficiency.

e_d = drive efficiency.

e_1 = locomotive mechanical efficiency.

With these formulae and the coal and water rates of the locomotive the total amount of fuel and water required to supply the power enroute can be calculated.

Fig. 2 shows the relation between the cost per thousand car miles and total miles run per year for the ice and electro-mechanical systems. These curves are based on the cost figures previously given and the following formula:

$$\text{Cost per thousand car miles} = \frac{A}{CM \times D} + M + \frac{10000}{45}$$

where

A = Annual fixed charges (dollars) calculated as follows:

a. Depreciation	12.5 per cent
b. Interest	6.0 per cent
c. Taxes and insurance	1.5 per cent
Total	20.0 per cent

CM = thousand car-miles per day.

D = days in service per year.

M = maintenance cost per thousand car miles.

C = operating cost per hour.

45 = average miles per hour.

No charges for precooling are included. The curves for the steam and direct drive mechanical compressor systems would be between those shown for the ice and electro-mechanical compressor systems.

PROBLEMS IN PRACTICE

1 • What is the capacity of the air conditioning unit in the average car? 2000 to 2500 cfm.

2 • How much steam is required for car heating on the coldest days? Pullmans 250 lb per hour, coaches 150 to 175 lb per hour, baggage cars 150 lb per hour.

3 • What types of cooling systems are used? Ice, steam jet, and mechanical compressor systems.

4 ● What refrigerating capacity is required?

5.5 to 7 tons per car.

5 ● What would be the annual operating cost for a car cooling system, using ice, if the car travels 150,000 car miles?

\$2,850.

Chapter 14

TEMPERATURE AND HUMIDITY CONTROL

Apparatus Sensitive to Temperature, Apparatus Sensitive to Relative Humidity, Accessory Apparatus, Temperature Control Systems, Control of Automatic Fuel Appliances, Individual Room Control, Zone Control, Industrial Processes, Air Conditioning Systems

AUTOMATIC controls can be installed on any type of heating, ventilating, or air conditioning system to maintain desired conditions automatically, and with maximum operating economy. The variety of automatic control equipment available is such that a suitable control system can be devised without difficulty, provided that the conditions to be maintained are known and the control equipment is properly chosen. This chapter outlines briefly the various types of control apparatus and indicates the method of their application to typical heating, ventilating, and air conditioning systems. Specific control devices and systems are described in the *Catalog Data Section* of THE GUIDE.

Controls are applied for the following reasons:

1. To maintain conditions required for human comfort and efficiency.
2. To maintain conditions required for industrial processes.
3. To obtain economy in operation.
4. To provide necessary safety measures.

CONTROL APPARATUS

The various pieces of control apparatus may be grouped under the following general headings:

Apparatus Sensitive to Temperature

Temperature-sensitive devices which will respond to changes in temperature, and which will motivate equipment to compensate for the changes, are usually called *thermostats*. They have many specialized forms for use in specific control applications. Thermostats are the detectors of a control system which identify changes in desired temperature conditions and automatically call for compensating action.

Thermostats are actuated by various means, all of which have the common characteristic of responsiveness to small changes of temperature. The actuating element may be a piece of bi-metal in straight, helical, or spiral form (Fig. 1), which, by bending slightly as the temperature changes, actuates an electric or pneumatic *switch* to govern the controlled

apparatus; or the actuator may be a diaphragm, bellows, or tube filled with a volatile liquid (Fig. 2) in such way that expansion and contraction with changes in temperature will operate the controlled apparatus by a direct mechanical, electric, or pneumatic connection.

A *room or wall thermostat* in its simplest form contains a single temperature-sensitive element which is so set that it maintains, by actuating the controlled system, a single temperature. A variation of this is the *two-temperature* or *dual thermostat* having two temperature-sensitive elements, one of which is set for a higher temperature than the other. Such a thermostat is used on day-night systems where the night temperature

enclosed in a tube to keep it from direct contact with the liquid. They are used in pipes, vats, and tanks, and are called *immersion thermostats*.

Sometimes *surface thermostats* are used in place of duct or immersion thermostats. These devices, so constructed as to respond to changes in temperature of the surface of the duct or vessel containing a fluid, are clamped or screwed to such surfaces in a manner which will provide as rapid as possible heat transfer between the surface and the sensitive element.

Other forms of thermostats are available for specialized purposes, but those described previously are the ones in general use for heating, ventilating and air conditioning systems.

Apparatus Sensitive to Relative Humidity

Devices which are responsive to changes in the relative humidity of the surrounding air, and which will motivate equipment to compensate for the changes, are called *humidistats* or *hygrostats*. These may vary considerably in their sensitive elements, but they all operate through connecting equipment which automatically causes humidifying apparatus to supply more or less moisture as required. Some of the more complicated ones contain essentially two thermostats, one working on a dry-bulb temperature and the other on a wet-bulb temperature; by proper interconnection of the parts they operate to maintain a definite relation between these two temperatures. Other devices use elements, directly sensitive to humidity, made of special wooden blocks, human hair, fiber, membranes, or strips of prepared paper. Hygrostats are available for use with both electric and pneumatic control systems.

Apparatus Which Operates Valves

Apparatus which is so mechanically or electrically equipped that it will open and close valves, and possibly give them intermediate positions in any pipe line of a heating, ventilating, or air conditioning system, is termed a *valve operator*. The function of a valve operator is, essentially, to move the plunger of a valve in a manner required by its type and construction. For instance, in a single-seat valve, the disc is moved against the seat and held there with sufficient pressure to prevent flow. A three-way valve, however, requires a valve operator that will place the double disc, as required, between the two seats. Each type of valve has special characteristics to which a valve operator must be adapted.

When a valve is used in shut-off service the valve operator simply opens the valve or closes it completely, as required. When the valve is to provide throttling service, a different type of valve operator is used so that the valve may be held at any intermediate position between open and closed. Valve operators use as their power source either compressed air (pneumatic system), electricity (motor-driven type or solenoid type), or a volatile liquid (direct-connected type).

Apparatus Which Operates Dampers

Apparatus which is so mechanically or electrically equipped as to open and close dampers, and possibly give them intermediate positions, in accordance with the purposes of the system using the dampers is termed

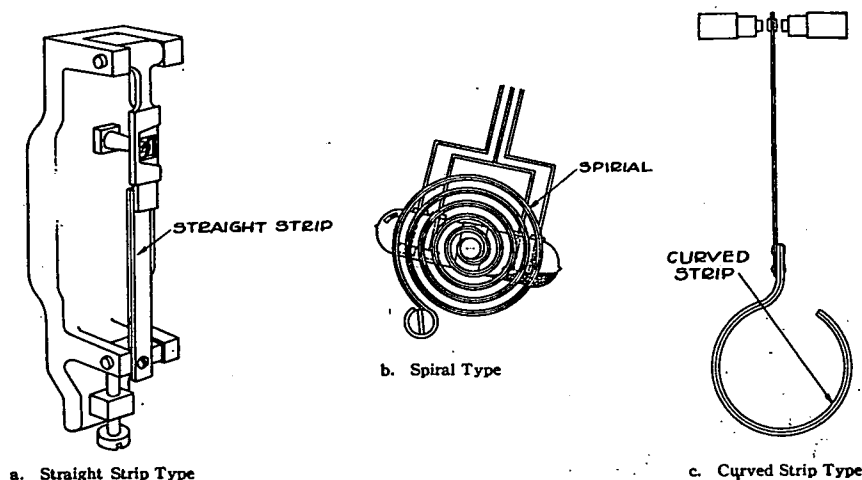


FIG. 1. TYPICAL BI-METALLIC THERMOSTATIC ELEMENTS

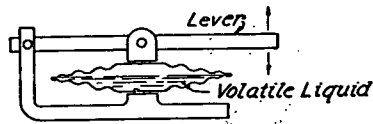


FIG. 2. DIAPHRAGM TYPE THERMOSTAT

is to be lower than that maintained during the daytime hours. Switching the control from one element to the other is accomplished by an external or an internal switch, which can be operated manually or by a time device.

Duct type thermostats are used in systems where the equipment must respond to changes in the temperature of the air passing through a duct. In their usual form, these thermostats are so constructed that their switching mechanism is outside the duct, while the temperature-sensitive element projects inside into the air stream.

Thermostats which operate in liquids have the same general construction as duct thermostats except that the sensitive element is usually

a *damper operator*. Damper operators are made for opening, closing, and positioning the dampers in the ducts of heating, ventilating, or air conditioning systems in the same way that valve operators regulate the valves. They are controlled from thermostatic or manual switches.

The sources of power used are compressed air, electricity, or volatile liquids. The damper operator is connected to its damper by direct connection or by a linkage, according to conditions, and it can usually be mounted either outside or inside the duct in which the damper is located.

Accessory Apparatus

Accessory apparatus is that additional equipment at the terminals of a control system necessary to make it operative. Every temperature control system requires a number of accessories, which will vary with the different types of systems. For instance, pneumatic systems require a compressor and a storage tank for the air which operates the units, and low-voltage electric systems require a transformer or generator to provide the required current.

Most of the larger control systems will have some sort of central switch-board which may include indicating and recording devices as well as control switches. Thermostat guards are generally used in gymnasiums, schools, and places of assemblage for protective purposes. Time switches and similar devices are often important parts of certain types of control systems. Couplings, mountings, and indicators are often parts of a system.

Connecting Apparatus

Connecting apparatus is that equipment used to connect the various parts of a control system. Because the parts of the system are often some distance apart, the connecting means are important, and the connections must be properly planned and made.

The connecting elements are fairly obvious. The pneumatic system uses compressed air carried in small pipes and tubing. Electric systems are wired for low-voltage or high-voltage power supply. Systems employing volatile liquids generally use flexible tubing if there is distance between the sensitive bulb and the operating unit. Each form has certain limitations which the designer of the system must consider.

Since few control installations are alike, the manufacturers of control apparatus usually maintain engineering departments staffed by experienced men whose advice may be had on control problems. Progress in automatic control has been rapid in the past few years and the field has become highly specialized.

TEMPERATURE CONTROL SYSTEMS

Control of Radiators or Convectors

The control of direct radiation is simple. Each radiator has a valve on its steam or water supply, with a thermostat to govern the opening and closing of the valve to maintain the desired uniform temperature as shown in Fig. 3. One thermostat may control the valves on all the radiators in a room, or, if the room is large, more than one thermostat may be used, with

each one governing one radiator or a group of them. Unit type thermostatic valves may be used, one on each radiator.

The location of wall thermostats is important. They must be on inside walls where they will not be affected by drafts of either warm or cold air, but where they will be exposed to general room conditions. If vibration is present, they must be mounted on shock-absorbing bases. If the walls are abnormally hot or cold, the thermostats must be mounted on heat-insulating bases. The connecting means can be concealed in the wall, under the floor or ceiling, or behind baseboards or moldings.

Modulating type valves cannot be used successfully on one-pipe steam systems because the partial opening of valves will not allow the condensate to escape against the incoming steam.

A discussion of steam heating systems is given in Chapter 31, and

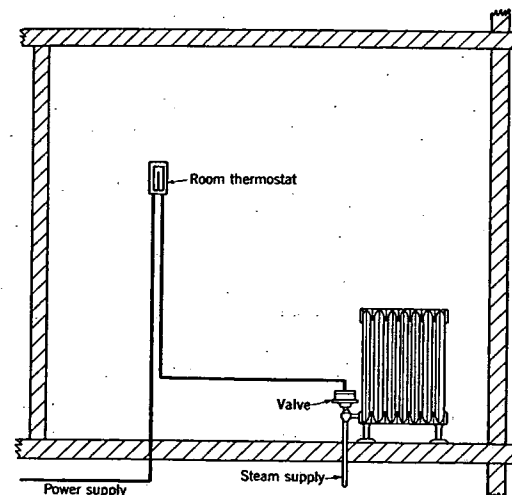


FIG. 3. CONTROL OF DIRECT RADIATOR WITH ROOM THERMOSTAT

further information on control requirements of direct radiation may be obtained therefrom.

Control of Unit Heaters

Unit heaters are commonly ceiling-hung or floor-mounted units consisting of a steam or hot water coil with a fan behind it to force air past the coil and into the room. Vanes direct the warm air flow. The simplest and commonest way to control a unit heater is to have in the heated space a thermostat which will turn on the fan when heat is required and shut it off when the demand is satisfied. However, where there is natural circulation through the unit, it is advisable to put a valve on the steam or hot water supply line and arrange it so the steam will be turned on only when the fan is running.

As a precaution against allowing the unit heater motors to continue to run if the steam supply fails or is for some reason shut off, a thermostat

on the return line may be installed to stop the motor when the temperature in the return line is below a predetermined point as shown in Fig. 4. When the fan and the steam are controlled simultaneously, such thermostat will also prevent the blowing of cold drafts. The net result in any case will be that the fan will run only when there is heat in the coil.

Control of Unit Ventilators

The unit ventilator presents a different control problem than the unit heater. Generally this type of unit draws its supply of air from the outside, heats it, and introduces this air into the room under control as shown in Fig. 5. There are many types of unit ventilators available. Some have a mixing damper by which the temperature of the air entering

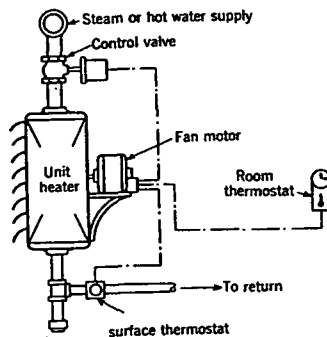


FIG. 4. UNIT HEATER CONTROL

the room may be varied, others have valves for this purpose, and still others use a combination of the two. Regardless of the construction of the machine, the essential requirement is that the temperature of the air delivered to the room should change slowly and remain as near room temperature as possible. Frequently direct radiators are used in conjunction with the unit ventilators to supply additional heat in extremely cold weather or for quickly heating up the room.

The four general types of control for unit ventilators are as follows:

1. A damper operator, which is controlled by a room thermostat, is attached to the mixing damper. When the thermostat calls for heat, the damper is moved to a position which forces more air through the heating unit and thus increases the amount of heat supplied to the room. This action must be gradual so that the air temperature may be changed slowly to prevent the drafty condition caused by supplying first hot and then cold air. This simplest arrangement is often condemned because it frequently results in drafts.
2. In mild weather the heating unit frequently supplies sufficient heat to cause overheating of the room, even though all of the air is by-passed around the heating unit. To avoid this fault a valve is placed on the heating unit to close the steam supply when the damper is by-passing all of the air. This valve is used in addition to the damper operator explained in the foregoing paragraph, but though giving better results, it may fail to prevent drafts.
3. In some unit ventilators one or more heating units are used without a mixing damper. A gradual-acting valve on each heating unit controls the supply of steam to the

unit to give the proper amount of heat required to maintain the desired room temperature. A thermostat to govern each valve may be installed in the room, or one thermostat may be used for all valves, but unless a thermostat is placed directly in the air stream of each unit, drafts may be encountered.

4. Another type of unit ventilator is arranged so that all recirculated air passes through the heating unit, and the outside air is introduced into the room for cooling purposes only. The outside air damper and the recirculated air damper are interlocked so that one damper operator will control them. In addition a valve operator is placed on the heating unit. Both of the operators should move gradually to avoid drafty conditions. When the thermostat calls for heat, the damper operator slowly closes the outside air damper and simultaneously opens the recirculating damper; if this does not meet the demand, the valve on the heating unit opens until the room temperature reaches the desired point.

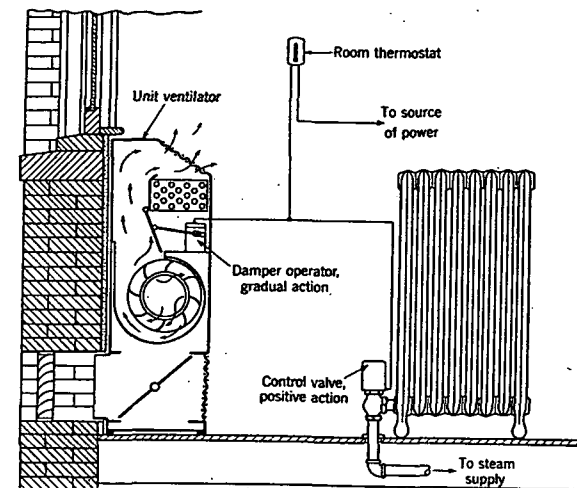


FIG. 5. CONTROL OF UNIT VENTILATOR AND RADIATOR

For additional information on the control of unit ventilators, refer to Chapter 12.

Central Fan Heating and Ventilating Systems

The numerous types of central fan systems present many control problems. In general they all have one point in common, namely, that the temperature change may be very fast because of rapid circulation.

System for Ventilating Only (Split System). Fig. 6 shows an accepted control for ventilating systems. Thermostat *A* located in the outside air duct is set just above freezing, and controls a valve *C* on the first heating coil. This valve is either completely open or completely closed. The bypass damper *B* and the other two valves *D* and *E* are controlled by a duct thermostat *F* located in the discharge duct from the fan. If the temperature of the air surrounding the thermostat *F* increases, the damper is moved automatically to admit more cold air. Should this not reduce the temperature sufficiently, the valves on the heating coil will be closed

gradually and in sequence until the correct temperature is reached. The opening or closing of the damper *B* and the valves *D* and *E* must be gradual or there will be a wide fluctuation in air temperature.

In ventilating systems it is customary to supply air to the ventilated spaces at an inlet temperature approximately equal to the temperature maintained in the rooms. The radiators therefore are designed to take care of all the heat losses from the room. Hence, in order to maintain controlled room temperatures it is necessary to use room thermostats governing control valves placed on the radiators. With this type of central fan system it is possible to ventilate a large number of rooms by means of one fan.

In some installations, such as in theaters or auditoriums, it is difficult to install sufficient direct heating surface to offset the heat losses from the room. Also there are installations where a short heating-up period is allowed before occupancy of the room, and it is advisable to use the

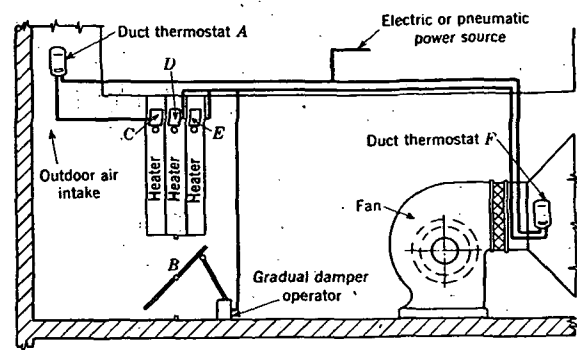


FIG. 6. CONTROL OF A SPLIT SYSTEM OF VENTILATION

entire heating capacity of the ventilating system for this purpose. An additional thermostat may be installed in the room to disable the duct thermostat and utilize the full heating capacity when the room is below a certain temperature, or a switch may be provided in the room or elsewhere for this purpose.

In central fan systems, air washers are often used and in such cases, due to the effect of temperatures on humidity, additional control is required. Fig. 7 shows such an arrangement with control of the second tempering heating unit by the air washer temperature and with the usual control of the first tempering heating unit by the outside temperature. This permits the air to be kept cool while passing through the washer so that too much moisture will not be absorbed. Fig. 7 also shows control of the reheating units by a duct thermostat in the fan discharge, and the application of a pilot thermostat to a system of this sort.

Combined Systems. There are various central fan systems which are used for both heating and ventilating. They are usually arranged with tempering heating units, automatically controlled to provide a minimum temperature for ventilating only, and additional heating units to supply

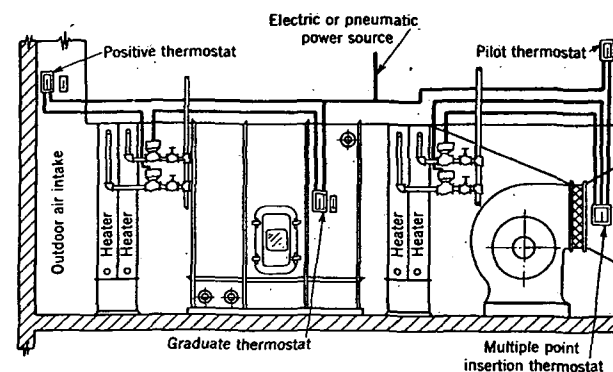


FIG. 7. USE OF PILOT THERMOSTAT ON VENTILATING SYSTEM WITH AIR WASHER

the heating requirements. Fig. 8 shows a type of system which has the reheating units located in the fan room. Tempered air at about 70 F is supplied to the fan. It may be further heated by the reheating units, or it may pass into the tempered air chamber. A room thermostat controls a gradual-acting damper operator on the double mixing damper in the warm and tempered air chambers. When the thermostat calls for heat, the damper operator moves the dampers so that more air is taken from the warm air chamber. It is essential that the double mixing damper be moved slowly to prevent alternate blasts of hot and cold air from being supplied to the room.

Outside Air, Recirculating, and Vent Dampers. In all types of plenum systems, the outside air damper is usually opened and closed by a damper operator. This operator may be controlled from a switch in the engineer's room or it may be operated by a relay in the fan motor circuit.

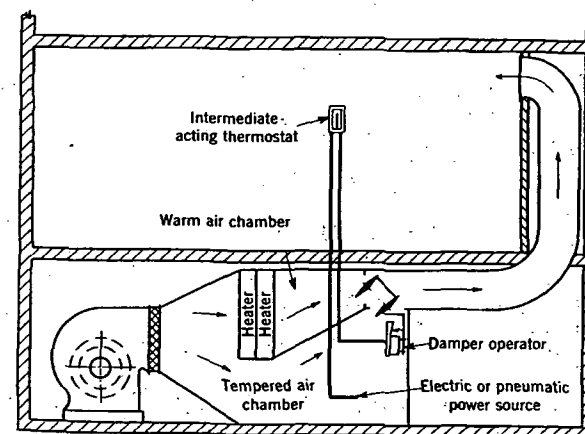


FIG. 8. CONTROL OF MIXING DAMPERS WITH INTERMEDIATE-ACTING THERMOSTAT

When the ventilating fan is started, the relay causes the damper operator to open the outside air damper.

Recirculating dampers and vent dampers may also be opened and closed by means of damper operators controlled from remote locations. Generally these damper operators are positive acting and are either completely opened or closed. However, in some cases where part outside air and part recirculated air is used, it is advantageous to use damper operators which have a certain number of definite positions. With this type of operator it would be possible to use 75 per cent outside air and 25 per cent recirculated air, or any other proportions which might be predetermined. These damper operators are controlled from switches generally mechanically interlocked so that the total opening of the two dampers is 100 per cent.

Hand-Fired Coal Systems

In small buildings the heating plant may be controlled by a single thermostat located in a key room in the building, instead of each room having its own control.

The most common control for a hand-fired furnace or boiler consists of a room thermostat and a furnace regulator of some type. The thermostat should be located in a representative room; never, of course, near the chimney or heat flue, too close to a radiator, or in a drafty hallway, and preferably on an inside wall. The regulator is attached to the draft and check dampers of the furnace. When the temperature of the air surrounding the thermostat drops, the thermostat causes the furnace regulator to open the draft and close the check damper. As soon as the room comes up to temperature, the draft is closed and the check damper opened. With this arrangement on hot water heating systems it is advisable to install an immersion thermostat in the boiler. This thermostat should be connected with the room thermostat so that both must call for heat before the draft is opened, but either one may cause the draft to be closed. On warm air systems it is advisable to use a bonnet thermostat and on steam heating systems a pressure limiting device, in series in each case with the room thermostat. If the temperature of the heating medium becomes too high, the drafts will be closed even though the room thermostat continues to call for heat.

There have been some recent improvements in controls of this type, involving the use of special types of thermostats and auxiliary apparatus which will give closer control and prevent overheating in mild weather.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic temperature control be used with oil burners, gas burners, and stokers to aid economical operation. There are many types of burners and many types of control, but there are some points common to all. First, a room thermostat is located in a key position in the building to maintain a given temperature at that point. Safety devices are installed in connection with this thermostat so that a failure of the ignition, power, or fuel supply will shut the system down. The same limit controls as recommended for coal burning should be used.

Oil Burners

Fig. 9 illustrates diagrammatically the essentials of an oil burner control circuit. Three thermostats are employed as shown in the illustration. Thermostat No. 1 will stop the burner when the room temperature is too high and No. 2 will stop the burner when the temperature of the heating medium exceeds the setting of thermostat No. 2. Both temperatures must be below their respective thermostat settings to start the burner. Thermostat No. 3 responds to the flame temperatures and in conjunction with the control switch acts as a safety to stop the burner if the latter fails to ignite or burn properly as demanded by thermostats Nos. 1 and 2.

Domestic Applications

Steam and hot water heating plants are often used to provide heat for the domestic hot water supply as well as for heating the building. Fig. 10

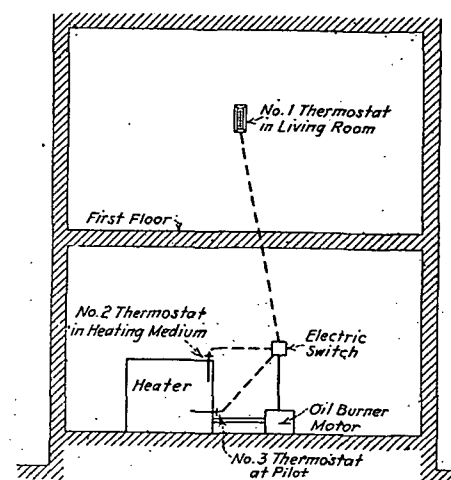


FIG. 9. ELECTRIC THERMOSTAT APPLIED TO OIL-FIRED HEATING SYSTEM

illustrates one such system. The burner control is similar to that shown in Fig. 9 except that either the room thermostat or the tank thermostat may start the burner. If the house is warm enough, the house temperature control valve will remain closed, and the boiler, through the coil heater, will warm the water in the storage tank until the tank thermostat is satisfied when the latter will close its valve and stop the burner. If the house is not warm enough when the tank thermostat is satisfied, the latter will merely close its control valve and the burner will continue to run. The burner will stop only when both thermostats are satisfied, or when the steam pressure shall have reached that allowed by the pressurestat. Much the same control is applied to gas burners and automatic coal stokers.

Gas Heating Appliances

On account of the ease and effectiveness with which the fuel can be controlled, gas-burning appliances are particularly adaptable to full

automatic control. Standard equipment on a steam boiler generally includes provision for control through a room temperature thermostat, a steam pressure regulator, and a device which shuts off the gas in the event that the water level becomes too low. Practically all gas boilers are or may be equipped with automatic safety pilots which shut off the gas if the pilot flame is too low.

Water boilers are adapted to operation under thermostatic room temperature control and are also provided with water temperature control equipment. Warm air furnaces can be under the control of thermostats

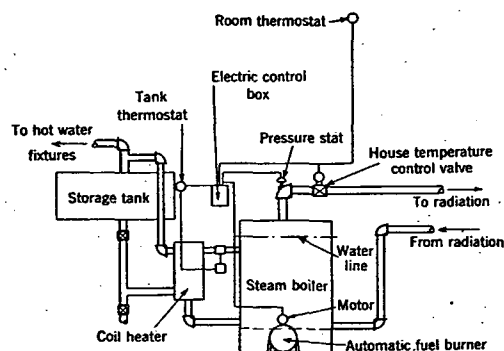


FIG. 10. TYPICAL ARRANGEMENT OF STEAM OR VAPOR SYSTEM WITH TWO THERMOSTATS CONTROLLING AUTOMATIC FUEL BURNER USED FOR HOUSE HEATING AND WATER HEATING

in the spaces being heated, as well as thermostats located in the heat ducts for the purpose of preventing unpleasantly hot air reaching the heated spaces. Variations in the pressure under which the gas is supplied to the appliance are controlled by means of a gas-pressure regulator. This is an essential part of practically all makes of gas-burning heating appliances; in fact, a gas-pressure regulator is required by the *American Gas Association* on all approved gas boilers, warm air furnaces (except floor furnaces), and unit heaters.

INDIVIDUAL ROOM CONTROL

The most complete type of automatic control is that by which the temperature in each room or in a group of rooms can be controlled. A thermostat in each room governs the valves on the radiators in that room, opening them as heat is called for and shutting them when the room is warm enough. The thermostats are all connected in relay so when any thermostat is calling for heat, an automatic burner will supply steam, hot water, or warm air, to the system; and when all the thermostats are satisfied, the burner will shut off. This is an excellent arrangement for larger residences, and it may be applied, in modified form, in houses which have one room or a section that is difficult to heat.

ZONE CONTROL

Zone control is a step between a single thermostat and individual room temperature control. The building is first divided into sections or zones which may have quite different heat requirements. With this method of control:

First: The zoning should be done with reference to the compass, since the north and west quarters in most localities require considerably more heat during the heating season than do the south and east quarters.

Second: Most large office buildings have more or less space occupied by merchants, and some by clubs, or restaurants, which have short hours of occupancy. Much can be accomplished in zoning with reference to the kind of occupancy of space. For additional information on this subject, refer to Chapter 31.

Variations of the usual zone control methods by the use of recently developed special devices have been quite successful in obtaining greater economy from heating systems. Frequently these use an outside thermostat or group of thermostats which adjust the operation of the controls to conform to variations in weather conditions.

COOLING UNITS

Cooling units are readily adaptable to thermostatic control. Several arrangements are as follows:

1. Room thermostat in conjunction with a magnetic or motor-operated valve to regulate the flow of refrigerant to coil. Usually the fans operate continuously.
2. Room thermostat to control the operation of the compressor. The fans operate continuously.
3. Room thermostat to control the operation of the fan motors.
4. Room thermostat to control the operation of the fan motor and the compressor motor simultaneously.
5. Room thermostat to control the operation of the compressor with back pressure control to regulate the fans.

INDUSTRIAL PROCESSES

There are many industrial processes requiring automatic temperature and humidity regulation. The control equipment operates on the same principles that have been described, but it is often especially designed for each particular process. Each installation, or the installation for each process, is likely to be a problem peculiar to that process.

AIR CONDITIONING SYSTEMS

The following fundamental principles should be borne in mind in the solution of problems involving the control of air conditioning systems:

1. Dew-point temperatures vary only with the amount of moisture. That is, no matter how much a given mixture of air and water vapor is heated or cooled, the dew-point temperature remains the same, as long as there is no addition or subtraction of water. Cooling below the dew-point temperature will, of course, cause condensation of the water vapor. Also, at the same temperature, there is always the same proportion of

water vapor in the saturated mixture, provided sufficient water and time are furnished for saturation.

Table 5, Chapter 1, shows the amount of moisture required to saturate a space at various temperatures. When the proper amount of moisture is determined, it is only necessary to set the air washer (dew-point) thermostat for the corresponding temperature of saturation; then if the air entering the washer has more humidity than desired, the excess will be condensed; and if it has less, the deficiency will be absorbed from the sprays.

For example, the dew-point temperature at 70 F and 40 per cent relative humidity is 45 F. Therefore, if the air temperature is maintained at 45 F as it leaves an air washer (assuming it is fully saturated) and then is heated to 70 F, it will have a relative humidity of 40 per cent. If it is desired to maintain these conditions in a given space, the air temperature can be raised to any necessary point, say 120 F (at which the relative humidity will be only 9 per cent). When the heat in the air has been dissipated, the space temperature being maintained at 70 F, the relative humidity will be 40 per cent.

2. Within ordinary operating ranges, saturated air will have a relative humidity of approximately 50 per cent when its temperature is raised 20 deg. For example, saturated air at 40 F raised to 60 F has a relative humidity of 48 per cent; 60 F saturated air raised to 80 F has a relative humidity of 50 per cent. (See Table 4, Chapter 1.) Thus a differential thermostat can be used to maintain a nearly constant relative humidity of 50 per cent by holding the dew-point temperature 20 deg below the dry-bulb temperature.

3. The total heat of the air and the water vapor mixed with it varies directly with the wet-bulb temperature. For example, the occupants of an auditorium give off sensible heat which tends to raise both the dry-bulb and the wet-bulb temperatures of the space; but the occupants also give off moisture which increases the absolute humidity and tends to further raise the wet-bulb temperature by an amount which is a direct indication of the heat expended by each occupant in evaporating this water. This relationship is useful in regulating the total heat, as wet-bulb temperatures can be controlled directly by means of a thermostat having a sensitive element covered with water-fed wicking, similar to a wet-bulb thermometer.

For example, the total heat of air at 80 F and 60 per cent relative humidity is the same as for air saturated at 70 F, i.e., 33.96 Btu per pound, both having a wet-bulb temperature of 70 F. Air at 80 F and 60 per cent relative humidity (70 F wet-bulb = 33.96 Btu per pound) reduced to 70 F and 50 per cent relative humidity (58½ F wet-bulb = 25.37 Btu per pound, total heat) must give up 8.59 Btu per pound. If the sensible heat and moisture pick-up in an auditorium is 8.59 Btu per pound of air handled in the conditioning system, the wet-bulb temperature of the air entering the space must be maintained at 58½ F to secure a final condition of 80 F and 60 per cent relative humidity.

Control of Relative Humidity

The following are the most commonly used methods of controlling relative humidity:

1. A thermostat is located in or at the outlet of a spray-type air conditioner which maintains a constant saturation temperature of the air leaving the conditioner by varying the temperature of water entering the suction of the pump supplying the spray nozzles, or by varying the temperature of the air entering the conditioner, or both. The temperature of the air entering the conditioner may be varied by use of tempering heaters, or by the proper proportioning of supply and return air entering the conditioner. This thermostat is known as a dew-point thermostat, as it determines the dew-point temperature of the air introduced into the conditioned spaces. A second thermostat in the room, or in the path of the air leaving the room, maintains a constant dry-bulb temperature by varying the amount of sensible heat added to the air leaving the conditioner, or by varying the volume of air introduced into the conditioned spaces. These two thermostats, in combination, control the dry-bulb and dew-point temperatures, which accordingly fix the relative humidity.

2. A wet-bulb thermostat is located in the room, or in the path of the air leaving the room, to maintain a constant wet-bulb temperature by varying the saturation temperature at the air conditioner outlet. A dry-bulb thermostat is located in the room to maintain a constant dry-bulb temperature, which in combination with a constant wet-bulb temperature fixes the relative humidity.

3. A differential thermostat may be used to control relative humidity. This instrument consists of two thermostatic elements, one of which is in the path of the air leaving

the conditioner, and the other under the influence of the dry-bulb temperature in the room. Instruments of this kind maintain a constant relative humidity by maintaining a constant difference between the dew-point temperature and the dry-bulb temperature in the room. (See Item 2 under Air Conditioning Systems.) One thermostatic element may be equipped with a moistening device to permit it to operate on wet-bulb temperatures. Such an instrument can be used to control the wet-bulb depression and thus the relative humidity.

4. A humidistat which responds directly to changes in humidity may be used to maintain a predetermined relative humidity with constant or with varying temperature. It may do this by varying the dew-point temperature of air leaving a conditioner; by varying, with dampers, the proportion of moist and dry air; by varying the amount of moisture otherwise added to the air; or by varying the dry-bulb temperature.

CENTRAL FAN AIR CONDITIONING SYSTEMS

In central fan air conditioning systems as described in Chapters 9 and 22, varying amounts of outside and recirculated air are used, except where contamination prevents re-use, and in general for obtaining humidity

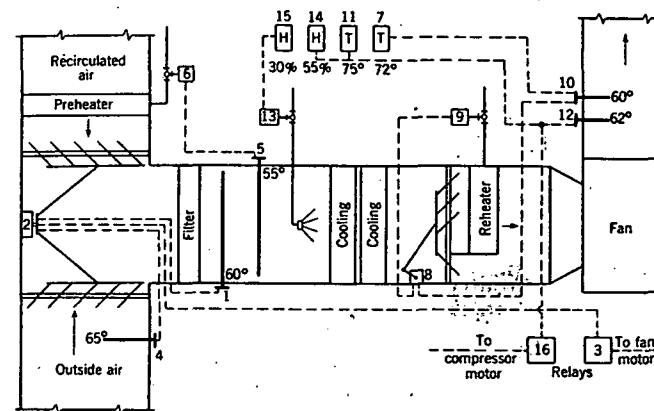


FIG. 11. ALL-SEASONS AIR CONDITIONING SYSTEM WITH COMPLETE AUTOMATIC CONTROLS

control under winter conditions heat is supplied to the air after it has passed the cooling coils. There are many control variations in use, and it is impractical to attempt a description of all the more usual ones in the limited space available. The following described system is, however, representative of many which are now in use.

Fig. 11 is a diagrammatic view of a completely automatic control system for heating and cooling, humidifying and dehumidifying. No manual switching is required between summer and winter operations. The control is fully automatic even in the spring and fall and under conditions where cooling and dehumidification is required in relatively cold weather.

Description of Control Units

Thermostat 1 controls damper 2 so that as the temperature at the thermostat rises above its setting, more outside air and less recirculating air is used. Relay 3 in the fan motor circuit is so arranged that when the

motor is stopped, the outside air damper is completely closed. Thermostat 4 in the incoming air is connected to controller 2 in such a manner that, regardless of thermostat 1, when the outside air is above the setting of this thermostat, the outside air damper will be closed to whatever minimum is set by the requirements of a code or the particular use of the system. Thermostat 5 is located at approximately the same point as 1, but controls valve 6 on the preheater so that if the temperature at thermostat 5 falls below its setting (which should be lower than the setting of thermostat 1), the recirculated air will be reheated so that the mixture will rise above this setting. Both controller 2 and valve 6 should be modulating type. The preheater may be placed in the outside air instead of in the recirculated air, but if this is done an additional thermostat will ordinarily be required in the outside air in order to prevent the possibility of freezing the preheater. Room thermostat 7 governs controller 8, which operates a face and by-pass damper as shown, and valve 9 on the reheater coil. The connection is such that if the room temperature is below the setting of thermostat 7, the by-pass damper will be closed and the face damper opened and, if necessary, more steam admitted to the reheater coil. Duct thermostat 10 inserted in the fan discharge is connected with thermostat 7 so that if the fan discharge temperature falls below the setting of thermostat 10, controller 8 and valve 9 will be moved in the direction to deliver more heat. Both controller 8 and valve 9 should be modulating type. Room thermostat 11 and duct thermostat 12 together control the cooling equipment which is shown as a relay to start the compressor. These thermostats must be respectively set for higher temperatures than thermostats 7 and 10. They are arranged so that if the temperature of the room exceeds the setting of thermostat 11, the cooling will be initiated unless the temperature of the fan discharge is below the setting of thermostat 12. Hygrostat 14 is also arranged to control relay 16 but in such a manner that if the relative humidity is above the setting of hygrostat 14, the cooling will be initiated regardless of the temperature in the fan discharge. In this case, the fan discharge temperature will be controlled by thermostat 10. Hygrostat 15, set much lower than hygrostat 14, is shown controlling a valve 13 for a spray type humidifier.

Heating Season Operation

Temperatures indicated in Fig. 11 will be used in the following description of operation as they are fairly indicative of an average installation of this type. The temperature at thermostats 1 and 5 will be maintained at 60 F if the weather is not too severe (below about 15 F if a minimum of 25 per cent outside air is required) by movement of the outside air and recirculated air dampers through the medium of controller 2. If the weather is extremely cold, the temperature at this point will be maintained at 55 F by means of thermostat 5. If the room temperature is below 72 F (the setting of thermostat 7), controller 8 and valve 9 will move in the direction to increase the heat until thermostat 7 is satisfied, after which they will move in the direction to decrease the heat until the duct temperature falls to 60 F. Hygrostat 15 will introduce moisture if the relative humidity in the room falls below 30 per cent.

Even in rather cold weather cooling is sometimes required, but will be

supplied without artificial refrigeration since the thermostat 7 will be above its setting and the fan discharge will be held at 60 F. The starting of the compressor will be prevented by thermostat 12 even if the room temperature goes above 75 F.

Cooling Season Operation

If the weather is very warm, thermostat 4 will close the outside air damper to its minimum position, and thermostat 1 will no longer be in control. If the compressor is stopped, the temperature at thermostat 12 will exceed 62 F which will cause the compressor to start provided the room temperature is above 75 F (the setting of thermostat 11). However, if the relative humidity exceeds 55 per cent, the temperature of the air leaving the cooling coils should be reduced as much as possible, and under this condition hygrostat 14 (as previously described) causes the compressor to run continuously. If sufficient cooling capacity is provided, the temperature leaving the cooling coils will fall below 60 F (the setting of thermostat 10) which will cause controller 8 and valve 9 to reheat the air to 60 F, thus providing maximum dehumidification and a satisfactory entering air temperature.

If cooling is required in mild weather (for example, 55 F outside), the control functions as before except that thermostat 1 is in control of the outside air and recirculated air damper, and sufficient cooling and dehumidification may be provided by the outside air alone. However, if insufficient cooling or insufficient dehumidification (or both) can be obtained in this manner, the fan discharge temperature will exceed 62 F with the room thermostat 11 above its setting, or the relative humidity will exceed the setting of hygrostat 14, and the artificial refrigerating system will again come into operation.

PROBLEMS IN PRACTICE

1 ● What important functions of heating, ventilating, and air conditioning systems do automatic controls fulfill?

Controls are applied to maintain adequate requirements for human comfort and efficiency; to maintain requirements for industrial processes; to obtain economy in operation; and to provide necessary safety measures.

2 ● How may temperature control be obtained in a room heated by a unit heater?

With constant steam supply, the unit heater motor may be started or stopped by a thermostat, either directly or through a relay. With intermittent steam supply, operation of the motor by thermostat can be limited to the time that steam is available, by using a reverse-acting temperature or pressure limit switch.

3 ● How may temperature control be provided in a room heated and ventilated by a unit ventilator which includes two extended-surface units?

Operation of the unit for service during occupancy of the room may be manual, by switch, or by time clock. When the desired temperature level is reached, the outside air intake may be controlled by a damper motor coupled with the fan motor circuit by means of a thermostat. The outside air damper will operate to a given position in either case.

Air passing through the unit may be preheated through the first heating coil to a definite temperature by a control valve on the steam supply governed by a temperature controller reacting to the temperature of the air on the outlet side of the convactor. The second

heating coil may provide the necessary heating capacity, and the steam supply to this coil may be modulated, either manually or automatically, in accordance with the temperature required in the room.

4 ● How may temperature control be obtained in a room heated by a duct system?

Air may enter the room from the central fan system at a predetermined minimum temperature. Heaters placed in the duct to bring the air up to this temperature should be equipped with face and by-pass dampers which may be adjusted by a positioning damper motor to give temperature control.

5 ● How may temperature control be obtained in a room cooled by a self-contained mechanical unit?

The fan operation may be controlled by a manual switch, while a room thermostat in conjunction with a solenoid valve may regulate the flow of the refrigerant to the coil. The thermostatic circuit might be operative only when the fans are running; and the compressor might be controlled by refrigerant pressure.

6 ● How may temperature control be obtained in a room heated by an automatically-fired warm air furnace?

A room thermostat might control the combustion unit; and a limit switch in the top of the furnace unit, when at a low setting of its control might operate the fan whenever there is a rise of temperature, and when at a high setting of its control it might shut off the combustion unit. A room humidity control operating a solenoid valve on the water supply to the humidifier, or operating a relay on the recirculating pump motor to the humidifier, may be connected in parallel with the fan motor. Humidification may be supplied only when heat is supplied and when the humidity control acts in conjunction with a time switch.

7 ● How may humidity be controlled in a unit humidifier for a steam or hot water heating plant?

Since heat is required for evaporation, a temperature limit switch, preferably of the immersion type, may be placed in the heating supply riser to cause the unit to be inoperative when heat is not available. A room humidity control will operate a solenoid valve on the water supply to the sprays. Both the solenoid valve and the humidity control may be electrically wired in parallel with a fan motor, and be subject to the temperature limit switch.

8 ● Discuss a control system, including control of humidity, for the heating cycle of a central fan system of air conditioning.

During the heating cycle it is necessary to vary the amount of outdoor air drawn into the system in accordance with the temperature of that air. It is also advisable to adjust the volume of return air when mixing it with the outdoor air so that the resultant mixture will be of constant volume delivered to the preheater coils at some predetermined constant temperature.

The reheating coil determines the dry-bulb temperature of the delivered air, so if the conditioner is equipped with both face and by-pass dampers on this coil it is obvious that these dampers should be controlled by a thermostat located at some representative position in the space being supplied with the conditioned air. If this thermostat is in turn connected with auxiliary apparatus which will vary the damper settings, it will be possible to pass more or less air through the reheater as the temperature falls or rises.

A low-limit temperature control might also be mounted in the discharge duct as a precaution against blowing cold air into the space. Such control would actuate the dampers of the reheater when the duct temperature fell below a predetermined minimum regardless of the demands of the master controller.

The amount of steam supplied to the reheater coils should be a function of the position of the dampers. If the face dampers are closed no heat is required, and to conserve steam suitable interconnection between the damper motor and the control valve should be made in order that this valve will close whenever the damper valve is closed. By adding modulating auxiliary apparatus to the steam valve, it may be made to operate proportionately to the setting of the dampers.

Chapter 15

AIR POLLUTION

Sources of Air Pollution, Effects of Air Pollution on Health, Pulmonary Effects, Occlusion of Solar Radiation, Industrial Air Pollution, Abatement of Atmospheric Pollution, Smoke Abatement, Dust and Cinder Abatement

THIS chapter considers the hygienic aspects of atmospheric pollution and the methods by which this pollution may be lessened. Information concerning the cleaning of air brought into buildings for ventilating purposes will be found in Chapter 16, and a discussion of the exhausting of dusts and toxic gases from factories and industrial plants is considered in Chapter 21.

The impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silk, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon. (See Fig. 1, Chapter 16, for size of impurities in air.)

Dust, Fumes, Smoke

The most conspicuous sources of atmospheric pollution may be arbitrarily classified according to the size of the particles as dusts, fumes, and smoke. *Dusts* are particles of solid matter varying from 1.0 to 150 microns in size. *Fumes* include particles resulting from chemical processing, combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size. *Smoke* is composed of fine soot or carbon particles, less than 0.1 micron in size, which result from incomplete combustion of carbonaceous materials, such as coal, oil, tar, and tobacco. In addition to carbon and soot, smoke contains unconsumed hydrocarbon gases, sulphur dioxide, sulphuric acid, carbon monoxide, and other industrial gases capable of injuring property, vegetation, and health.

The lines of demarcation in these three classifications are neither sharp nor positive, but the distinction is descriptive of the nature and origin of the particles, and their physical action. Dusts settle without appreciable agglomeration, fumes tend to aggregate, smoke to diffuse. Particles larger than one micron will eventually settle out by gravitation; particles smaller will remain in suspension as permanent impurities unless they agglomerate to sizes larger than one micron.

Fly-Ash, Cinders

The term *fly-ash* is usually applied to the extremely small particles of ash, and the term *cinder* to the larger particles of coke and ash which are discharged with the gases of combustion from burning coal.

AIR POLLUTION AND HEALTH

Many kinds of dusts and gases are capable of producing pathological changes which may cause ill health. The harmful effects depend largely upon the chemical and physical nature of the impurities, and the concentration, length of time, and conditions under which they are breathed. Dust particles must be minute in size to be inhaled at all, although fairly large particles may gain access to the upper air passages.

The human body possesses remarkable filtering media for protecting the lungs. Small hairs which line the nasal passages, and a multitude of microscopic hairs, called *cilia*, in the epithelial lining in the bronchial tubes intercept many of the dust particles before they reach the lungs.

The constant inhalation of dusts in city air irritates the mucous membranes of the nose, throat, and lungs, and eventually may produce discomfort and a series of minor respiratory disorders. The pigmented lung of the city dweller is an example of the pathological change produced over a period of years. This condition may be of no clinical importance, but an exaggeration of it in the coal miner results in anthracosis or dark spots on the lung due to the presence of pigment in the lymph channels which impairs the functioning of the lung cells under stress.

Effects of Solids

Bronchitis is the chief condition associated with exposure to thick dust, and follows upon inhalation of practically any kind of insoluble and non-colloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it appears to have a marked influence in aggravating the disease once it has started. There is, however, quite reliable evidence that carbon pigment, one of the atmospheric dusts, tends to wall off local tuberculosis rather than to further its spread.

The sulphurous fumes and tarry matter in smoke are probably more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages, leading to asthmatic breathing and bronchitis and, in extreme cases, to death. The Meuse Valley fog disaster will probably 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 sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons. Physical examination showed difficult breathing, rapid pulse, cyanosis, cardiac dilation, and a redness and inflammation of the mucosa of the nose, mouth, throat, trachea, and bronchi.

Carbon monoxide from automobiles and from chimney gases constitutes another important source of aerial pollution in busy cities. During heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets is found to contain enough CO

to menace the health of those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of CO in city air is believed to be insufficient to affect the average city dweller or pedestrian.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore¹ by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City² a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

The effect of air pollution on the health of city dwellers is difficult to determine, owing to the slowness of its manifestations. The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution have seemed justifiable to the public.

AIR POLLUTION IN INDUSTRY

In many industrial processes, sufficient amounts of dusts, fumes, and vapors are liberated to be injurious to the health of workers. Some dusts are poisonous (lead, mercury, arsenic, manganese, and cadmium) and some act as irritants (silica, steel, iron, and granite). Certain dusts may produce catarrhal conditions and increase susceptibility to such diseases as bronchitis, pneumonia, and tuberculosis. Silicious dust is especially harmful because it has a direct damaging action upon the tissue of the lungs, but organic dusts, both animal and vegetable (hair, pollen, textile, and fiber), do not seem to affect the lungs at all, although they may cause considerable discomfort in the upper respiratory passages to persons sensitive to them.

Industrial gases and fumes act specifically upon the mucous membranes, the lungs, blood, skin, and eyes. Some extremely poisonous gases act after very short exposures. Among these are carbon monoxide, hydrogen sulphide, ammonia, chlorine, bromine, arsine, and cyanogen.

The industrial processes which liberate harmful substances are too manifold and the effects too diverse to be considered here, where discussion is limited to the commonest and most serious with which the ventilating engineer may be confronted, namely, carbon monoxide, lead, and silica. For a more thorough treatise on the subject reference should be made to books by Hamilton³, Rosenau⁴, and Henderson and Haggard⁵.

Carbon Monoxide Poisoning

Carbon monoxide is a common form of poisonous industrial gas, met with in mines, foundries, coke-oven sheds, garages, and houses. Its action

¹Effects of Atmospheric Pollution upon Incidence of Solar Ultra-Violet Light; by J. H. Shrader, M. H. Coblenz and F. A. Korff (*American Journal of Public Health*, p. 7, Vol. 19, 1929).

²Studies in Illumination, by J. E. Ives (*U. S. Public Health Service Bulletin No. 197*, 1930).

³Industrial Poisons in the United States, by Alice Hamilton.

⁴Preventive Medicine and Hygiene, by Milton J. Rosenau.

⁵Noxious Gases, by Y. Henderson and H. Haggard.

is due to the fact that the combining power of carbon monoxide with the haemoglobin of the red blood corpuscles is about 300 times greater than that of oxygen. Since the resulting stable combination destroys the power of the haemoglobin to unite with oxygen in the lungs and to supply it to the tissues, the effects are due to lack of oxygen, and the symptoms are those of anoxemia, namely, dizziness, headaches, sleepiness, fatigue, and, in extreme cases, paralysis and death. The dangerous saturation level of the blood with carbon monoxide is about 50 per cent. Even as little as 0.07 per cent in the air will render, in half an hour, one quarter of the red corpuscles incapable of uniting with oxygen. One to two parts per 10,000 parts of air is set as a safe limit of pollution which may be breathed for a long time without producing perceptible symptoms.

Silicosis

Silicosis is a chronic disease of the lungs which results from the local physio-chemical action of hydrated silica upon the pulmonary tissue, causing progressive lymphatic fibrosis, and rendering the tissue susceptible to tuberculosis. The disease is slow in evolution, requiring usually a number of years of exposure. It occurs principally among granite workers, sand blasters, metal miners, metal polishers, potters, and mill-stone workers.

Lead Poisoning

Lead poisoning is the most insidious and most common of all industrial diseases. It occurs principally among lead workers and smelters, lead miners, potters, painters, typesetters, stereotypers, plumbers, and workers with glass, gold and silver. Lead, in practically all forms, is a cumulative poison which is absorbed by way of the blood stream, chiefly from the respiratory tract, but also from the digestive tract and from the skin. The effect may be either an acute or chronic poisoning. The principal symptoms are colic, constipation, anemia, headache, anorexia, a bluish line along the edges of the gums, rheumatic pains, and, in extreme conditions, paralysis, blindness, insanity, and death.

It has been found⁶ that 2 mg per day is the smallest dose, by inhalation, which in the course of years may result in lead poisoning. Regular inhalation during the usual working hours of air containing less than 0.2 mg of lead per cubic meter does not seem to produce serious lead poisoning in individuals of representative industrial groups⁷.

Prevention

The prevention of industrial hazards from dusts and poisonous gases is largely a ventilation problem consisting of keeping the impurities in air down to a safe concentration. As yet there are no generally accepted standards on which to base the design of the ventilation equipment. Approximate data on the toxicity of various gases and fumes met with in industrial establishments are given in Table 1. Column 5, giving the maximum allowable concentrations for prolonged exposures, was compiled from experiments in which most exposures lasted not more than a

⁶Lead Poisoning, by Thomas Morrison Legge (*Journal Royal Society Arts*, 1929, Vol. 77, p. 1023).

⁷What is a Dangerous Quantity of Lead Dust in Air, by C. M. Salls (*Industrial Hygiene Bulletin*, New York State Department of Labor, 1925).

week, and it is reasonable to assume that over more prolonged exposures such concentrations would cause pernicious effects.

Much is known concerning the physiological and pathological effects induced by various types and concentrations of atmospheric pollutants. In the absence of an accepted standard for safe breathing, and because of the slow, cumulative effects of certain kinds of air contaminants, the best procedure is the periodic medical examination of individuals, and the

TABLE 1. TOXICITY OF GASES AND FUMES IN PARTS PER 10,000 PARTS OF AIR^a

VAPOR OR GAS	RAPIDLY FATAL	MAXIMUM CONCENTRATION FOR FROM 1/2 TO 1 HOUR	MAXIMUM CONCENTRATION FOR 1 HOUR	MAXIMUM ALLOWABLE FOR PROLONGED EXPOSURE
Carbon monoxide.....	40	15-20	10	1
Carbon dioxide.....	800-1000	-----	-----	-----
Hydrocyanic acid.....	30	1 1/2	1/2	1/2
Ammonia.....	50-100	25	3	1
Hydrochloric acid gas.....	10-20	1/2	-----	1/10
Chlorine.....	10	1/2	-----	1/100
Hydrofluoric acid gas.....	2	1/10	-----	1/100
Sulphur dioxide.....	4-5	1/2-1	-----	1/10
Hydrogen sulphide.....	10-30	5-7	2-3	1
Carbon bisulphide.....	-----	11	5	1/2
Phosphene.....	20	4-6	1-2	-----
Arsine.....	2 1/2	1/2	1/2	-----
Phosgene.....	Over 1/4	1/4	-----	1/100
Nitrous fumes.....	2 1/2-7 1/2	1-1 1/2	-----	1/10
Benzene.....	190	-----	31-47	1 1/2-3
Toluene and xylene.....	190	-----	31-47	-----
Aniline.....	-----	-----	1-1 1/2	1/10
Nitrobenzene.....	-----	-----	1/100	1/100
Petrol.....	243	100-220	-----	-----
Carbon tetrachloride.....	480	240	40	16
Chloroform.....	250	140	50	2
Tetrachlorethane.....	73	-----	-----	1 1/2
Trichlorethylene.....	370	-----	-----	-----
Methyl chloride.....	1500-3000	200-400	70	5-10
Methyl bromide.....	200-400	20-40	10	2
Lead vapor.....	-----	-----	-----	5-6

^aOriginal data compiled by Y. Henderson and H. Haggard. (See *Noxious Gases*, 1927.) Data revised by T. M. Legge. (See *Lessons Learned from Industrial Gases and Fumes*, Institute of Chemistry of Great Britain and Ireland, London, 1930.)

routine measurement and study of the concentration and the physical and chemical characteristics of the dusts to which those individuals are exposed.

ABATEMENT OF SMOKE AND AIR POLLUTION

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is

the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of buildings that operate their own boiler plants.

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. (See Chapter 27.)

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 gravity oil always produces less smoke than a high volatile coal or low 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 necessary.

Legislative measures at the present time are largely concerned with the smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 43), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the

smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious gases, such as sulphur dioxide and sulphuric acid mist, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDERS

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in later paragraphs.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

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 coal 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.

Dust and Cinder Catchers⁸

The various types of dust and cinder catchers available today can be divided into six general classes:

1. Settling chambers.
2. Dust and cinder traps.
3. Centrifugal separators.

⁸See Smoke and Dust Abatement, by M. D. Engle (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

4. Electrostatic precipitators.
5. Gas scrubbers.
6. Fabric filters.

The selection of the proper type of catcher calls for a careful study of the material to be caught and the draft and space available. After installation, constant vigilance is necessary to keep the catchers in proper working condition if satisfactory operation is to be obtained.

If possible, the dust or cinder catcher should be installed on the inlet side of the induced draft fans because the dust and cinders in the gases seriously erode the wheels of the fans, the inlet connections and the scrolls. Where the induced draft fans operate at high tip speeds and no catchers are installed, it is not uncommon for the fans to require major repairs within one year and complete replacement within five years.

Settling Chambers

Probably the oldest form of dust catcher is the settling chamber, which generally consists of a large-sized, gas-tight space into which the dust-laden gases are discharged before being delivered to the chimney. The velocity of the gas should be reduced to a point where the larger and heavier particles will be precipitated by gravity. For good operation, the velocity of the gas should be reduced to a maximum of 2 fps. The bottoms of the chambers should be provided with dump plates through which the collected dust can be removed. Because these chambers are not effective in removing the finer dust particles they have been practically superseded by smaller and less costly devices.

Traps, Catchers, Precipitators

Various types of traps have been devised. In general they all depend upon breaking the gas up into thin strata and subjecting those thin strata to several abrupt changes in direction. The dust is thrown out of the gas stream into specially shaped pockets, or impinged against a roughened surface. The trapping pockets are drained into a hopper below with a small quantity of gas and the dust settles out by gravity due to the low velocity in the hopper. In the roughened surface type, various sections of the trap are closed off at intervals by means of dampers and the dust is shaken off the roughened surface into a hopper below.

These devices work very well in catching large size dust and cinders and trap much of the fine dust. They have been used most extensively on stoker-fired installations. They have the advantages of low pressure drop, relatively small space requirements, and low first cost.

Centrifugal catchers obtain separation by projecting the particles tangentially out of the gas stream. The effectiveness of this type of catcher varies directly as the specific weight of the dust and as the square of the tangential velocity, and inversely as the radius of rotation.

Electrostatic precipitators are used for catching fine dust. These precipitators consist of dust-tight chambers in which are suspended reinforced concrete slabs on about 10-in. centers. Between the slabs are suspended bare metal rods. High-voltage unidirectional current is applied to the reinforcing rods in the concrete slabs acting as positive electrodes, the bare rods acting as negative electrodes. The dust-laden

gas flows horizontally through the precipitator and the dust particles migrate toward the concrete slabs to which they adhere and then fall or are scraped off into the dust hoppers below.

Gas Scrubbers

Wet scrubbers have been used for many years for removing dust from gases. A number of different types of scrubbers are now being built for removing dust from boiler flue gases. One type depends upon saturating the gas and washing the dust out of suspension by a spray of water. For best results with this type, the water should be atomized into as fine a spray as possible.

Another type depends upon splitting the gas into thin strata and subjecting these strata to a number of abrupt changes in direction, throwing the dust against the wet surfaces. The main problem in developing a satisfactory wet dust catcher is to find suitable materials of construction that will resist the corrosive action of the wash water for a reasonable length of time.

Fabric Filters

Filters of many kinds have been used with variable success. The filter bags are made of cotton, wool or asbestos fabric. The fabrics used in these filters do not withstand the temperatures at which gases are usually discharged from the boilers, and hence the gases must be cooled by some means. Surface coolers or water sprays can be used for reducing the gas temperatures.

One of the serious objections to all of these dust catchers is the relatively high cost of installation and maintenance, and the space required for installation.

Disposal of Dust and Cinders

Even after the dust and cinders have been caught, the disposal of the material caught presents a serious problem. The cinders discharged with the gases from stoker-fired boilers are usually very high in carbon and contain from 50 to 80 per cent as much heat per pound as the coal which is being burned. It is possible, and usually economical, to burn these cinders. They cannot be satisfactorily mixed with the coal in the stoker hopper but they can be blown into the furnace over the stoker fuel bed and burned satisfactorily. If a sufficient quantity of cinders is caught, a small unit pulverizer can be installed to prepare them for burning over the stoker fuel bed. The same pulverizer can be used for coal at times of peak load and will materially increase the capacity of the fuel-burning equipment for the boiler to which it is connected.

No satisfactory market has been developed for the dust caught from pulverized coal installations, but the possibilities are being investigated and it seems likely that in the future this material will have a market value that will go a long way toward paying the fixed charges on the cost of catching it.

The distribution of dust in the gas entering and leaving the dust and cinder catchers is not uniform and is different in practically every in-

stallation, and varies widely with changes in furnace conditions. In order to obtain a representative sample it is necessary to traverse the inlet and outlet of the catcher with a sampling tube which faces into the gas flow. The velocity of the gas into the sampling tube must be the same as the velocity of the gas in the duct at the instant the sample is taken. The swirls and eddy currents in the ducts make it difficult to obtain consistent readings, but if the test is conducted by some one of experience, an indication of the approximate efficiency can be obtained.

Nature's Dust Catcher

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for each rain drop and the rain picks up dust as it falls from the clouds to the earth. In fact, without dust in the air to form the nuclei for rain drops it would never rain, and the earth would be continually enveloped in a cloud of vapor.

PROBLEMS IN PRACTICE

1 ● Classify the detrimental aspects of air pollution as it effects large industrial communities.

Air pollution may be classified (a) medical, as it affects the physiological functions of people; (b) botanical, as it affects vegetation, trees, plants, shrubs and flowers; and (c) physical, as it affects the discolorization and deterioration of buildings, and the nuisance of soiled interior furnishings, clothes, merchandise, etc.

2 ● Distinguish between dusts, fumes, and smokes.

Solid particles ranging in size from 1.0 micron to 150 microns are called *dusts* (micron = $\frac{1}{25,000}$ in.).

Particles resulting from sundry chemical reactions and ranging from 0.1 to 1.0 micron in size are called *fumes*.

Carbon particles less than 0.1 micron in size which generally arise from the incomplete combustion of such materials as coal, oil, or tobacco are called *smokes*.

3 ● What are some of the more important physical properties of these various groups of foreign bodies which are of importance in ventilation?

In slowly moving air, dusts tend to settle out by gravity without agglomerating to form larger particles; fumes have the tendency to form larger particles which will settle when they attain the size of approximately 1.0 micron; while smokes tend to diffuse and remain in the air as permanent impurities.

4 ● Why is atmospheric pollution an important engineering problem?

- Certain impurities, when present in too great concentrations, cause ill health or even death.
- High concentrations of solids occlude solar radiations.
- Some materials cause permanent injury to parts of buildings, as sulphur fumes corrode exposed metal.
- Extra cleaning expense is incurred in dusty localities.
- Internal combustion engines are damaged by abrasive dusts.

5 ● How may the hazards of dust-producing industrial operations best be curtailed?

By providing mechanical exhaust ventilation sufficient to keep dust concentration at a safe level (see Table 1) and then removing foreign bodies to reduce the pollution of outside air.

6 ● How may the pollution of the atmosphere be lessened?

By compelling industrial plants to install dust catching and smoke controlling devices. In many cities the domestic heating plant is one of the most serious offenders, but these plants are too small to justify the installation of dust catchers. Public education in improved firing methods would be of considerable help in this field.

7 ● Compare the dry and wet types of dust catchers.

The dry types are very effective in removing the larger dust particles but the smaller particles generally pass through other kinds than the electric precipitator. The dry types also require considerable space and therefore sometimes introduce resistance to the flow of air. The wet types are effective in removing some of the smaller dusts and the water-soluble gases. The principal disadvantage of the washer is its short life caused by the corrosive action of the wash water.

8 ● What size particles are detrimental to health?

While fairly large particles may enter the upper air passages, those found in the lungs are seldom more than 10 microns in size, and comparatively few of them are more than 5 microns. It is agreed that particles between $\frac{1}{2}$ and 2 microns may be harmful; some authorities place the upper limit at about 5 microns, and some incline to extend the lower limit to 0.1 of a micron.

9 ● Is the shape of the particle of any significance?

Hard particles with sharp corners or edges have a cutting effect on the delicate mucous membranes of the upper respiratory tract which may lower the resistance of the nose and throat to acute infections. This is aggravated by the irritating effects of some chemical compounds which may be taken in with the air and which act to reduce resistance.

10 ● What are the principal meteorological effects of smoke and dust?

a. The reduction in the amount of light received. Measurements have shown that visible light may be as much as 50 per cent less intense in a smoky section of a city than in a section that is free from smoke. Ultra-violet light is reduced as much or more, and in some cases is cut out entirely for a time.

b. Smoke and dust aid in the formation and prolongation of fogs. City fogs accumulate smoke and become darker in color and very objectionable. The sun requires a longer time to disperse them, and when the water is evaporated, there is a rain of smoke and soot particles that have been entrained.

11 ● Why has not smoke abatement been more effective?

Because communities have not been made sufficiently aware of the possibilities of burning high volatile fuels smokelessly and of separating cinder and ash from the stack gases to a degree that will prevent a nuisance.

12 ● Is the abatement of dust and cinders important?

Yes. Only a small percentage of the solid emission from stacks is smoke, in the accepted popular sense; the remainder is fly-ash and cinders. While black smoke is disagreeable and its tarry matter and carbon particles soil anything with which they come in contact, the cinders and some of the ash are hard and destructive. They also, together with dusts from industrial processes, make up the hard, sharp, irritating, air-borne solids that are breathed by individuals not working in a dusty mill or factory.

13 ● Are air-borne impurities causative factors in hay fever, bronchial asthma, and allergic disorders?

Yes. Recent medical investigations indicate that 90 per cent of seasonal hay fever and 40 per cent of bronchial asthma are caused by air-borne pollens, tree dusts, and other allergic irritants.

14 ● Name some essential requirements for the smokeless combustion of fuels.

Time, temperature, and turbulence. A study of these factors is usually of value in overcoming a smoke nuisance.

15 • What is the Ringelmann Chart Method of comparing smoke densities?

See Chapter 43. The Ringelmann Chart consists of four cards ruled with lines having different degrees of blackness. These cards, together with a white card and a black one, are hung in a horizontal row 50 ft from the observer. At this distance the lines become invisible and the cards appear to be different shades of gray, ranging from white to black. The observer, by matching the cards against the shades of smoke coming from a stack, is able to estimate the blackness of the smoke as compared with the chart.

Chapter 16

AIR CLEANING DEVICES

Requirements of an Air Cleaner, Types, Air Washers and Scrubbers, Viscous Type Filters, Dry Air Filters, Air Filter Installations

THE removal of impurities from air brought into a building for ventilating or air conditioning purposes is the function of any air cleaning or filtering device. These impurities include carbon (soot) from the incomplete combustion of fuels burned in furnaces and automobile engines, particles of earth, sand, ash, automobile tires, leather, animal excretion, stone, wood, rust and paper, threads of cotton, wool and silk, bits of animal and vegetable matter, bacteria and pollen. Microscopic examination shows that the character of the impurities varies with the locality, but as a rule carbon forms the greater part of them while the total is somewhat proportional to the state of industrial activity and the wind intensity. Additional information on sources of air pollution will be found in Chapter 15.

Observations have shown that practically all atmospheric impurities are less than 5 microns in size. (One micron equals 0.001 millimeter or approximately 0.00004 in.) The size and composition of each individual particle determines its buoyancy and consequently the length of time it will remain in suspension. The chart, Fig. 1, shows graphically the sizes of impurities found in the air, and other related data.

To estimate the probable dust load for air filter installations, the following approximate averages of atmospheric dust concentration may be used (7000 grains equal 1 lb):

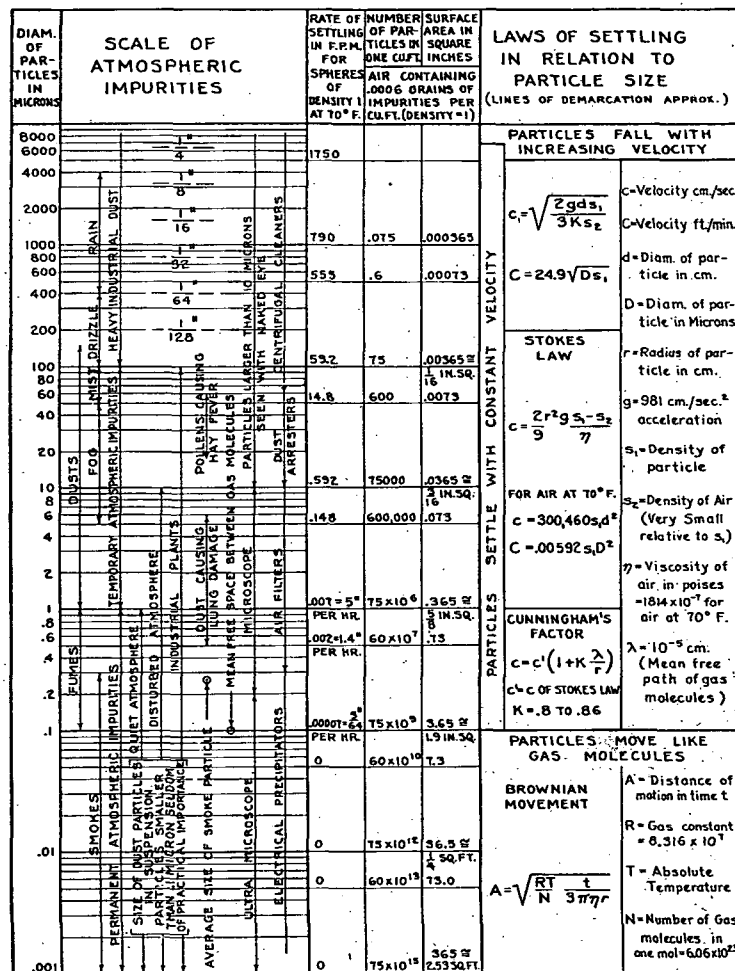
Rural and suburban districts.....	0.2 to 0.4 grains per 1000 cu ft
Metropolitan districts.....	0.4 to 0.8 grains per 1000 cu ft
Industrial districts.....	0.8 to 1.5 grains per 1000 cu ft

REQUIREMENTS OF AN AIR CLEANER

To fulfill the essential requirements of clean air, an air cleaner should:

1. Be efficient in the removal of harmful and objectionable impurities in the air, such as dust, dirt, pollens, bacteria.
2. Be efficient over a considerable range of air velocities.
3. Have a low frictional resistance to air flow; that is, the pressure drop across the filter, measured in inches of water, should be as low as possible.
4. Have a large dust-holding capacity without excessive increase of resistance, or have ability to operate so as to keep the resistance constant automatically.
5. Be easy to clean and handle, or clean itself automatically.
6. Leave the air passing through the cleaner free from entrained moisture or charging liquids used in the cleaner.

The A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilating Work¹ explains how such devices are rated by (1) capacity in cubic feet of air handled per minute, (2) resistance



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FIG. 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

in inches of water at rated capacity, (3) dust arrestance, the percentage relationship expressing dust removal efficiency at rated capacity, (4) reconditioning power, the energy necessary to operate the mechanism of

¹Adopted 1934 by A.S.H.V.E. See Chapter 44.

an automatic air cleaning device, and (5) dust holding capacity, the amount by weight of standard dust which a non-automatic air cleaning device will retain before reconditioning is necessary.

TYPES OF AIR CLEANERS

According to the Code, the following four classifications are given the devices:

Class A. Automatic Type: In general all air cleaning devices which use power to automatically recondition the filter medium and maintain a non-varying resistance to air flow.

Class B. Low Resistance Non-Automatic Type: Air cleaning devices for warm air furnaces, unit ventilating machines and similar apparatus and installations in which a maximum of not more than 0.18 in. water gage is available to move air through the air cleaning device.

Class C. Medium Resistance Non-Automatic Type: Air cleaning devices for systems in which a maximum of not more than 0.5 in. water gage is available to move air through the air cleaning device.

Class D. High Resistance Non-Automatic Type: Air cleaning devices for the air intake of compressors, internal combustion engines, and the like, where a pressure of 1.0 in. or more water gage is available to move air through the air cleaning device.

Air cleaners may be also classified as follows:

1. According to principle of air cleaning.

- a. Air washers.
- b. Viscous air filters.
 - (1) Unit type.
 - (2) Automatic type.
- c. Dry air filters.

2. According to application.

- a. For central fan systems of ventilation and air conditioning. Filters of the automatic or semi-automatic type are usually recommended and are installed in a central plenum chamber.
- b. For unit ventilators. Filters of viscous unit or dry type, installed at inlet of individual units.
- c. For window installations. Self-contained units consisting of fan and filter, usually dry type, adapted to be placed in the ordinary window.
- d. For warm-air furnaces. Unit type viscous or dry filters placed in small plenum chamber of warm-air house heating systems.
- e. For compressors and Diesel engines. Unit type viscous or dry filters, installed at air intake of compressors and Diesel engines.
- f. For compressed air lines. Unit type viscous or dry filters.

With the growing congestion of large cities and an industrial growth throughout the entire country, the percentages of foreign material in the air, such as soot or carbon, which are unaffected by an air washer type of air cleaner, have increased. This has brought about the development of the viscous and dry type air filters which are part of many ventilating and air conditioning systems.

AIR WASHERS AND SCRUBBERS

Information on air washers will be found in Chapter 11.

Scrubbers have not been used very extensively in the past for cleaning

air for ventilating purposes. However, new types have been developed which appear to have possibilities for cases where the air to be cleaned is extremely dirty or where a higher degree of cleanliness is desired than can be obtained with an air washer.

VISCOUS TYPE FILTERS

The principle of air cleaning used in viscous filters is that of *adhesive impingement*. Dust and dirt in the air, especially soot and carbons, are trapped and retained by successive impingements on coated surfaces. While the arrangements of filtering media and the kind of materials used are almost unlimited, there are certain rather definite requirements for a practical commercial filter.

Investigations in this country and abroad demonstrate that the first impingement of dust laden air on a viscous coated surface removes about 60 per cent of the dust, the next impingement takes 60 per cent of what then remains—that is, 24 per cent—and the next impingement removes 9.6 per cent. To secure maximum efficiency, it is necessary to divide the air into innumerable fine streams, as the more intimately and freely the air is brought into contact with the viscous-coated media the better will be the cleaning.

The binding liquid used with viscous filters should have the following properties:

1. Its surface tension should be such as to produce a homogeneous film-like coating on the filter medium.
2. The viscosity should vary only slightly with normal changes of temperature.
3. It should be germicidal in its action to prevent the development of mold spores and bacteria on the filter media.
4. The liquid should flow freely at low temperatures.
5. Evaporation should not exceed 1 per cent.
6. It should be fireproof.
7. It should be odorless.

Viscous Unit Filters

In the unit type viscous filter, the filtering media are arranged in units of convenient size to facilitate installation, maintenance, and cleaning. Each unit consists of an interchangeable cell or replaceable filter pad and a substantial frame which may be bolted to the frames of other like units to form a partition between the source of dusty air and the fan inlet. The necessary washing, draining, and recharging equipment should be installed near each group of unit filters, with hot water and sewer connections provided.

To secure greater dust holding capacity and a practically constant resistance and air volume, the filter media are usually placed in the direction of air flow, with progressively finer filter densities determined by the percentage of dust impinged. This arrangement provides relatively large spaces for the collection of dirt in the front of the filter where the bulk of the dust is taken out without undue increase in resistance, while at the back of the filter the openings are smaller to secure high efficiency in the removal of the finer dust particles.

The resistance of a well-designed unit filter of the adhesive impinge-

ment type usually depends upon the velocity at which the air is handled and upon whether the unit is clean or dirty. The cleaning efficiency of the unit is usually highest after it has accumulated a certain portion of its maximum load of dirt because some dust collected in the cell acts as an efficient medium for the further seizing of solids from the air. By periodically cleaning a predetermined number of cells, the resistance and capacity of a built-up filter may be held at any desired figure. The frequency of cleaning any unit filter installation depends upon the dust concentration

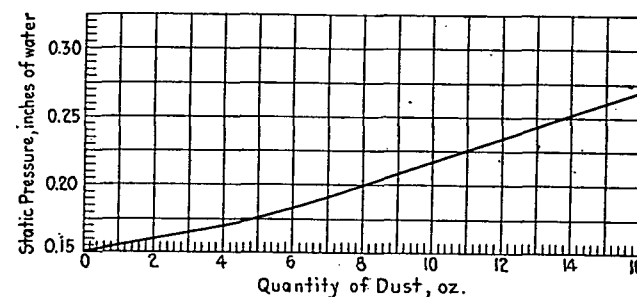


FIG. 2. CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION

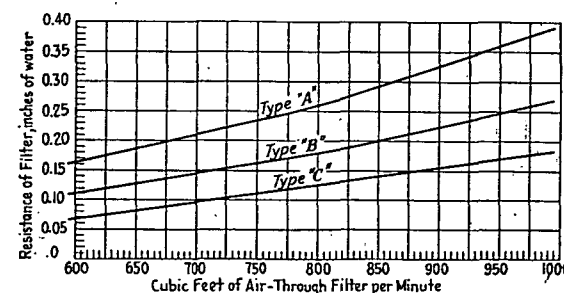


FIG. 3. RESISTANCE TO AIR-FLOW OF A TYPICAL UNIT AIR FILTER

of air being cleaned, and on the amount of dirt which can be accumulated in the filter medium without causing excessive resistance. (Figs. 2, 3 and 4.)

Filters consisting of inexpensive frames of cardboard or similar material filled with viscous-coated glass wool or steel wool are available. Because of their construction these units may be discarded when dirty and replaced with new units at relatively little expense. They are used in general ventilation work and with warm air furnaces and other installations where first cost and low resistance to air flow are essential. The operating characteristics of these units conform in general with those of the rigid frame type.

Viscous Automatic Filters

The principle of air cleaning used in the viscous automatic filters is the same as in the unit filters. The removal of the accumulated dust,

however, is done automatically instead of by hand. The automatic cleaning and recoating of these filters is based on the principle that the viscous fluid itself will perform the cleaning function, thereby eliminating a separate washing agent. The dust collected by the filter thus is deposited finally in the bottom of the viscous fluid reservoir from which it may be removed by different methods, depending on the design of the filter.

There are three general types of automatic filters. They are differentiated from each other according to the process of self-cleaning and renewing of the viscous coating used by each type, as follows:

1. The filter medium has the form of an endless curtain suspended vertically, with its lower portion submerged in a viscous fluid reservoir. The curtain rotates slowly through this bath, thus performing the cleaning and recoating of the filter medium.
2. The filter screen is arranged in the form of shelves or cylinders, and the viscous fluid is flushed through all parts of the medium in a direction opposite to the air flow.
3. The filter medium is arranged vertically and is stationary. The viscous fluid is flushed from above over the medium, while the air flow is stopped.

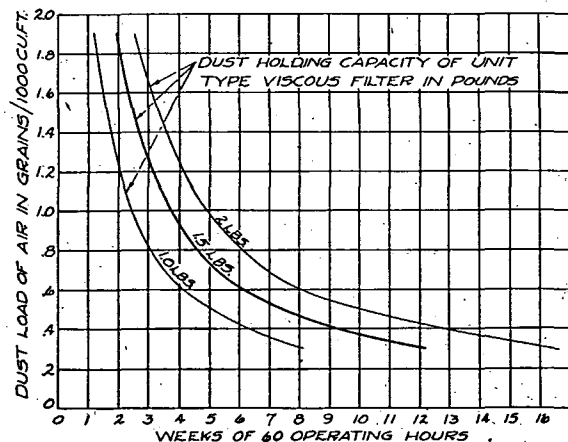


FIG. 4. MAINTENANCE CHART FOR UNIT TYPE VISCIOUS FILTERS

The washing and renewing process in automatic filters usually is intermittent. It is accomplished by an electric motor or by other motive power and is controlled by manual or by automatic timing devices. The operating cycle is of a predetermined frequency and should be so timed as to insure a constant static pressure drop across the filter. The customary resistance to air flow is $\frac{3}{8}$ -in. water gage at an air velocity of 500 fpm, measured at the filter entrance. Automatic viscous filters are made up in units which are delivered either fully assembled or in parts to be assembled at the point of installation.

DRY AIR FILTERS

Dry air filters, in which dust is impinged upon or filtered through screens made of felt, cloth, or cellulose, are available in various types. These filters require no adhesive liquid, but depend on the straining or screening action of the filtering medium. Because of the close texture

of the filtering media used in most of the dry filters, the surface velocity, or velocity of the air entering the media, ranges between 10 and 50 fpm, depending on the nature and texture of the fabric. This necessitates a relatively large screen surface, and the filter media are usually arranged in the form of pockets to bring the frontal area within customary space requirements.

As in viscous unit filters, an average constant resistance and air volume may be obtained by periodic reconditioning or renewal of the filter screens. Since some materials suitable for dry filtering media are affected considerably by moisture which tends to cause a rapid increase in resistance, they should be treated or processed to minimize the effect of changes in humidity.

Filters using felt and similar materials as filter media depend upon vacuum cleaning for reconditioning. A special nozzle, operated from a portable or stationary vacuum cleaner, is shaped to reach all parts of the filter pockets. Permanent filter media should be capable of withstanding repeated vacuum cleanings without loss in dust removal efficiency. While most dry filters are cleaned by replacing an inexpensive filter sheet, the useful life of these sheets often may be lengthened by vibrating or vacuum cleaning.

INSTALLATION METHODS

The published performance data for all air filters are based on *straight through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents and dead air spaces should be avoided and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle. An overload of 10 to 15 per cent is regarded as the maximum allowable. When air volume is subject to increase, a larger filter should be installed.
2. The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.
3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters and washers installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.
2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.
3. Access doors of convenient size should be provided in the sheet metal connections leading to and from the filters.
4. All doors on the clean air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet metal ducts on the clean air side should be as air-tight as possible.

5. Electric lights should be installed in the chamber in front of and behind the air filter.
6. Air washers should, whenever possible, be installed between the tempering and heating coils to protect them from extreme cold in winter time.
7. Filters installed close to air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.
8. Filters should have permanent indicators to give a warning when the filter resistance reaches too high a value.

REFERENCES

- Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).
- 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).
- Operation and Maintenance of Air Filters, by W. G. Frank (*Heating, Piping and Air Conditioning*, May, 1931).
- Size and Characteristics of Air-Borne Impurities, by W. G. Frank (*Heating, Piping and Air Conditioning*, January, 1932).
- Determining the Quantity of Dust in Air by Impingement, by F. B. Rowley and John Beal (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).
- A Study of Dust Determinators, by F. B. Rowley and John Beal (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).
- Design and Application of Oil-Coated Air Filters, by H. C. Murphy (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927).
- Determining the Efficiency of Air Cleaners, by A. M. Goodloe (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924).

PROBLEMS IN PRACTICE

1 ● Assume a fan and duct system which handled 10,000 cfm through clean filters with a system resistance of 0.8 in. of water and that after the filters have become dirty the system resistance increases to 1.0 in. of water, and that the fan speed remains unchanged. Is there any way of predicting the volume of air delivered after the filter becomes dirty?

Yes. If the performance curves for the particular make of fan are available, the new volume may be determined from the resistance pressure curve. (Figs. 1, 2, 3 and 4, Chapter 17.)

2 ● What are the advantages of viscous filters?

The principal advantage of the viscous filter is its large dust holding capacity. The dust accumulation is distributed through the depth of the filtering medium rather than upon the surface as in the dry types, which makes it possible for viscous filters to handle heavy dust concentrations without excessive resistance. Since its efficiency and resistance are based on maximum air velocities of from 300 to 500 fpm through the filter, the viscous filter consumes the minimum amount of space for a given air volume.

3 ● What are the advantages of dry filters?

Dry filters are more efficient in the removal of fine dust particles from the air, and some types will eliminate even as much as 60 per cent of the smoke particles. Dry filters also are easily and conveniently maintained by vacuum cleaning, vibrating, or renewing the filtering medium.

4 ● If an air washer is used for cooling and humidity control in an air conditioning system, is a filter needed?

An air filter is desirable in conjunction with an air washer because of the large amount of soot in the air which, due to its greasy and amorphous nature, is not readily trapped in

an air washer. Filters should be placed between the washer and the air intake so that all the dirt will be collected at one point to simplify maintenance and to protect all the equipment in the system.

5 ● Is an air filter needed with an extended surface type heat exchanger?

An air filter is essential with an extended surface heat exchanger in order to maintain its efficiency, for without this protection dust particles will adhere to the exposed surfaces, and gradually build up a deposit to the point where the efficiency will be impaired and the resistance increased by restricting the air passage.

6 ● What is the proper location of a filter in relation to the fan?

A filter will operate equally well whether placed on the suction or discharge side of the fan. It has become standard practice, however, to locate the filter on the fan inlet side because there it has: (1) simpler duct connections, (2) reduced static pressure losses, (3) more even air distribution over the entire filter area. Where an exceptionally high efficiency in dust removal must be maintained, it is often advisable to place the filter on the discharge side of the fan so there can be no infiltration of unclean air.

7 ● What instruments and apparatus are required for determining the pollen concentration in air by means of the settling method?

A microscope with a field of known area and a glass slide coated with a viscous material.

8 ● Describe the procedure for determining the pollen concentration in air by means of the settling method.

A glass slide coated with a viscous material is placed for a period of 24 hours in a horizontal position in the atmosphere to be tested. The slide is then removed and placed under the microscope, and pollen counts are made of approximately 25 fields over the area of the glass slide. Having determined the count over a definite area, as for example, 1 sq cm, and finding the settling rate of the average particles from the chart, Fig. 1, the concentration in parts per cubic yard can be calculated.

9 ● The resistance to air flow of a unit air filter is found to be 0.4 in. of water. The volume of air passing through the filter is 1000 cfm at a velocity of 200 fpm. What would be the filter area required in order to reduce the pressure drop across the filter from 0.4 in. of water to 0.16 in. of water?

Referring to Fig. 3: The resistance is substantially proportional to the square of the velocity, or

$$\begin{aligned} \frac{R_1}{R_2} &= \frac{V_1^2}{V_2^2} \\ \frac{0.4}{0.16} &= \frac{200^2}{V_2^2} \\ V_2^2 &= 16,000 \\ V_2 &= 126.5 \text{ fpm} \\ Q &= AV \\ 1000 &= 126.5 A \\ A &= \frac{1000}{126.5} = 7.91 \text{ sq ft} \end{aligned}$$

The filter area would be increased from 5 sq ft to 7.91 sq ft.

10 ● A ventilating system complete with filters has a fan which, when operating at 400 rpm and delivering air at 1 in. of water total static pressure, requires an input of 3 horsepower. After the system operates for a time, the pressure drop across the filter caused by the clogging action of the collected dust and dirt increases from 0.1 in. of water to 0.4 in. of water. To maintain the original

rate of air delivery with the increased static pressure, at what speed must the fan be run and what horsepower will be required?

Static pressure after clogging of filter = $1 + (0.4 - 0.1) = 1.3$ in. of water.

The static pressure varies as the square of the fan speed. Therefore, if X is the fan speed after the static pressure increases:

$$\frac{1.3}{1} = \left(\frac{X}{400} \right)^2$$

$$X = 456 \text{ rpm.}$$

The horsepower varies as the cube of the fan speed. Therefore, if Y is the horsepower after the static pressure increases:

$$\frac{Y}{3} = \left(\frac{456}{400} \right)^3$$

$$Y = 4.44 \text{ horsepower.}$$

To maintain the original rate of air delivery with the increased static pressure, the fan speed must be increased from 400 to 456 rpm, and the horsepower from 3 to 4.44.

Chapter 17

FANS

Performance, Fan Efficiency, Characteristic Curves, Selection of Fans, Controls, Designation of Fans

IN heating and ventilating practice, fans are used to produce air flow except where positive displacement is required, in which case compressors or rotary blowers are used. Fans are classified according to the direction of air flow as (1) *axial flow* or *propeller* type if the flow is parallel with the axis, and (2) *radial flow* or *centrifugal* type if the flow is parallel with the radius of rotation.

Axial flow fans are made with various numbers of blades of a variety of forms. The blades may be of uniform thickness (sheet metal), either flat or cambered, or may be of varying thickness of so-called aerofoil section (airplane propeller type). Where an axial flow fan is intended for operation at comparatively high pressures the hub sometimes is enlarged in the form of a disc and the fan is known as a *disc fan*.

Radial flow or centrifugal fans include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades. The angularity of the blades determines the operating characteristics of a fan: a forward curved blade is found in a fan having slow speed operating characteristics, while a backward curved blade is found in a fan having high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types in each general class can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type. The most common field of service for fans of the propeller type is in moving air against moderate resistances, especially where no long ducts or heavy friction must be overcome and where noise is not objectionable, whereas centrifugal fans are commonly employed for operation at the comparatively higher pressures and where extreme quietness is necessary.

PERFORMANCE OF FANS

Fans of all types follow certain laws of performance which are useful in determining the effect of changes in the conditions of operation. These

laws apply to installations comprising any type of fan, any given piping system and constant air density, and are as follows:

1. The air capacity varies directly as the fan speed.
2. The pressure (static, velocity, and total) varies as the square of the fan speed.
3. The power demand varies as the cube of the fan speed.

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?

$$\text{Speed} = 400 \times \frac{15,000}{12,000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

When the density of the air varies the following laws apply:

4. At constant speed and capacity the pressure and power vary directly as the density.

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.07495 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 0.06018 lb) and the speed of the fan remains the same, what will be the static pressure and power?

$$\text{Static pressure} = 1 \times \frac{0.06018}{0.07495} = 0.80 \text{ in.}$$

$$\text{Power} = 4 \times \frac{0.06018}{0.07495} = 3.20 \text{ hp}$$

5. At constant pressure the speed, capacity and power vary inversely as the square root of the density.

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?

$$\text{Speed} = 400 \times \sqrt{\frac{0.07495}{0.06018}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12,000 \times \sqrt{\frac{0.07495}{0.06018}} = 13,392 \text{ cfm (measured at 200 F)}$$

$$\text{Power} = 4 \times \sqrt{\frac{0.07495}{0.06018}} = 4.46 \text{ hp}$$

6. For a constant weight of air:

- (a) The speed, capacity, and pressure vary inversely as the density.
- (b) The horsepower varies inversely as the square of the density.

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?

$$\text{Speed} = 400 \times \frac{0.07495}{0.06018} = 498 \text{ rpm}$$

$$\text{Capacity} = 12,000 \times \frac{0.07495}{0.06018} = 14,945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.07495}{0.06018} = 1.25 \text{ in.}$$

$$\text{Power} = 4 \times \left(\frac{0.07495}{0.06018}\right)^2 = 6.20 \text{ hp}$$

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 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.

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 or dynamic pressure. This efficiency is variously known as the total, dynamic, 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 per-

¹See Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, Edition of 1932.

centages 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 are determined by tests performed in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers² as adopted by the AMERICAN SOCIETY OF HEATING and VENTILATING ENGINEERS and the *National Association of Fan Manufacturers*. The results of tests are plotted in different ways: the

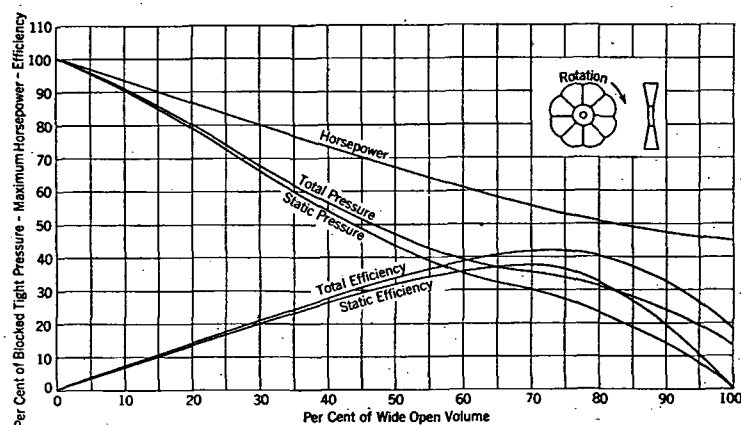


FIG. 1. OPERATING CHARACTERISTICS OF AN AXIAL FLOW FAN

abscissae may be the ratio of delivery, assuming full open discharge as 100 per cent, and the ordinates may be static pressure, dynamic pressure, horsepower and efficiency. Pressures may be expressed in per cent of the maximum pressure in the manner shown in the illustrations in this chapter, but in engineering calculations they are sometimes expressed in proportion to the pressures due to the peripheral velocity.

It should be noted that characteristic curves of fan performance are plotted for a constant speed. Some variation in values of efficiency may occur at different speeds but such variation is usually slight within a wide range of speeds. Fans of similar design but of different size will also show some difference in efficiency. Figs. 1 to 4 show characteristic curves for different types of fans using blades of various shapes, but without reference to the design of housing employed. The efficiency curves are therefore not serviceable for making rigid comparisons of efficiencies obtainable with blades of the various shapes but are intended merely to show reasonable values and more particularly to show the manner in which variations occur with changes in fan capacity.

²A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923. Amended June, 1931.

Axial flow fan characteristics are indicated by Figs. 1 and 2. These fans, when properly designed, have a satisfactory efficiency at low resistance, comparing favorably in this respect with centrifugal fans. They are low in cost and economical in operation and occupy relatively little space. Although this type of fan can operate against considerable resistance, the noise often becomes objectionable, so that it does not always compare favorably with centrifugal fans for such service. With most of the designs which employ blades of uniform thickness the power increases rapidly with an increase in resistance.

The curves (Fig. 1) show the rapid reduction in capacity and increase in power as the resistance increases. The low efficiency when overcoming

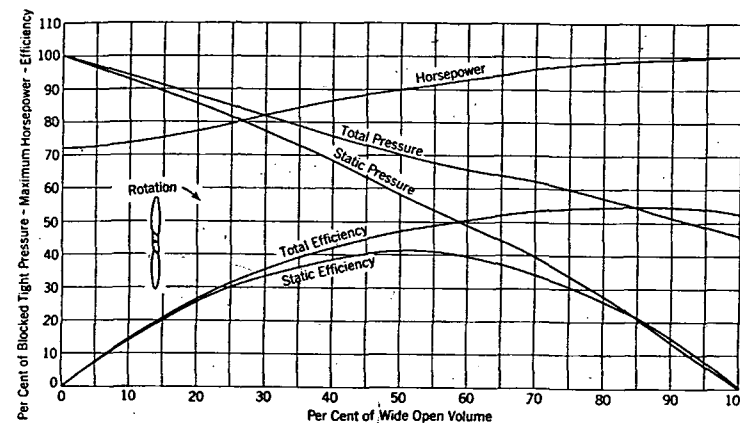


FIG. 2. OPERATING CHARACTERISTICS OF AN AIRPLANE PROPELLER FAN

heavy resistance is due to the low speed of the blades near the hub as compared to the relatively high peripheral or tip speed. The air driven by the blade area near the rim can pass back through the less effective blade area at the hub more easily than it can overcome the duct resistance.

Fig. 2 shows the performance of the *airplane propeller fan* in which the blades are similar in shape to those of an airplane propeller but of varying number according to the pressure to be developed. This fan usually operates at a higher speed than does the former type of propeller fan, and with a different power characteristic, the power remaining fairly constant throughout the range of pressures, being somewhat less at the higher than at the lower pressures. The flatness of the horsepower curve indicates the advantage of this type of fan in preventing overloading of motors where fluctuations in pressure occur. Variations in the diameter, width, pitch, camber, and the thickness of the blades provide a considerable degree of flexibility in design, so that the peak of total efficiency may be made to occur at wide-open volume or at various percentages of that volume.

Another advantage of this type of axial flow fan is its low resistance to air passage when standing still. There are some installations in which such a characteristic is desirable.

The straight blade (paddle-wheel) or partially backward curved blade type of fan is practically obsolete for ventilation. Its use is largely confined to such applications as conveyors for material, or for gases containing foreign material, fumes and vapors. The open construction and the few large flat blades of these wheels render them resistant to corrosion and tend to prevent material from collecting on the blades. This type of fan has a good efficiency, but the power steadily increases as the static

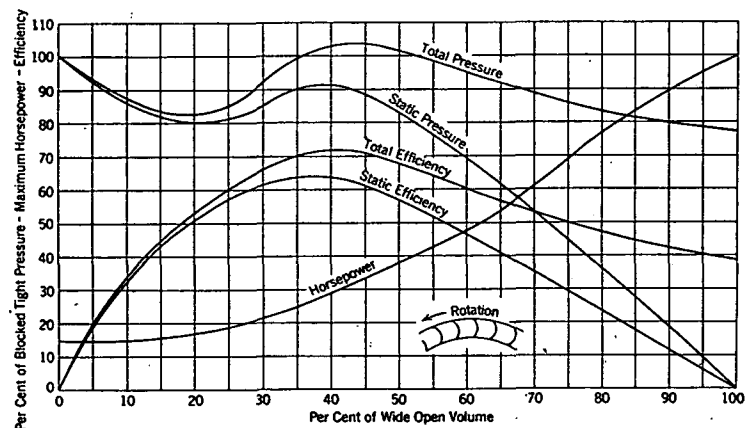


FIG. 3. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

pressure falls off, which requires that the motor be selected with a moderate reserve in power to take care of possible error in calculation of duct resistance.

The *forward curved multiblade fan* is the type most commonly used in heating and ventilating work, as it has a low peripheral speed, a large capacity, and is quiet in operation. The point of maximum efficiency for this fan occurs near the point of maximum static pressure. The static pressure drops consistently from the point of maximum efficiency to full open operation. The power curve rises continually from low to peak capacity, but if reasonable care is exercised in figuring resistance there is no danger of overloading the motor.

The outstanding characteristics of the *full backward curve multiblade type fan* are the steep pressure curves, the non-overloading power curve, and the high speed. (See Fig. 4.) This fan operates at a peripheral speed of approximately 250 per cent of the forward curve multiblade type for like results. The pressure curves begin to drop at very low capacity and continue to fall rapidly to full outlet opening. The steep pressure curves tend to produce constant capacity under changing pressures. Where wide fluctuations in demand occur, this type of fan is desirable to prevent overloading of motors. 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 high speed may necessitate somewhat heavier construction and more operating attention or service. The dimensional bulk for a given duty often is 150 per cent of that of a forward curve multiblade type fan.

Between the extremes of the forward and the full backward curve blade type centrifugal fans a number of modified designs exist, differing in the

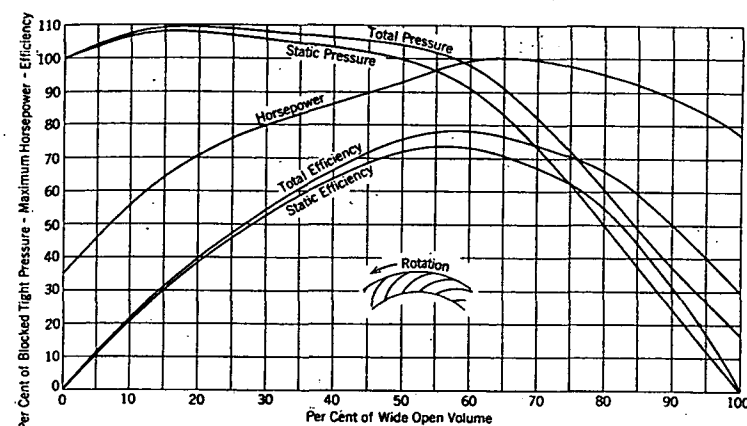


FIG. 4. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

angularity or in the shape of the blades. Common among these designs are the straight radial blade type, the radial tip type, and the double curve blade type with a forward angle at the heel and a slight backward angle at the tip of the blade. Characteristic curves of these types show varying degrees of resemblance to the curves of Figs. 3 and 4, according to the degree of similarity to one or the other of the two designs of fan considered.

SYSTEM CHARACTERISTICS

A given fan performs as determined by the real characteristic of the system to which it is attached. When a different performance of a fan is desired, it is necessary to either change the speed of the fan (as *A* to *B* or *C* to *D* in Fig. 5), or to change the system (as by moving a damper from *A* to *C* in Fig. 5). If the speed of the fan is changed, the new point of operation is the intersection of the constant speed static pressure—cubic feet per minute curve for the new speed with the system characteristic. If the system is changed, the new point of operation is the intersection of the constant speed static pressure—cubic feet per minute curve with the new system characteristic.

Heating and ventilating systems follow the simple parabolic law quite closely but other types of systems follow some other more or less complex relation. The more complex systems can be separated into their component parts whose individual characteristics are known and the summation of the characteristics of the several parts of a system will give the composite characteristic of the system.

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.

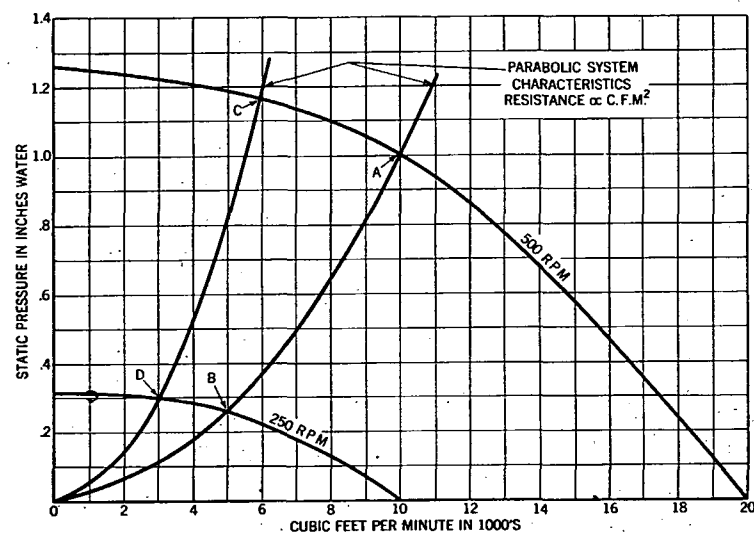


FIG. 5. ILLUSTRATION OF OPERATING POINTS OF A GIVEN FAN AT TWO SPEEDS ON THE SAME AND DIFFERENT SYSTEMS

Knowing the requirements of the system, the main points to be considered for fan selection are (1) efficiency, (2) speed, (3) noise, (4) size and weight, and (5) cost.

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, 0.07488 lb per cubic foot).
2. Outlet velocity.
3. Revolutions per minute.
4. Brake power.
5. Tip or peripheral speed.
6. Static pressure.

CHAPTER 17—FANS

The most efficient operating point of the fan is usually shown by either bold-face or italicized figures in the capacity tables.

Fans for Ventilation and for Cooling Systems

Two important factors in selecting fans for ventilating systems are efficiency (which affects the cost of operation) and noise. First cost and space available are secondary. The fans should be selected to operate at maximum efficiency without noise. Because noise in a ventilating system is irritating and a cause for complaint, fans must be selected of proper size in order to reduce it to a minimum. Noise may be caused by other factors than the fan, namely, high velocity in the duct work, unsatisfactory location of the fan room, improper construction of floors and walls, and poor installation. Where noise is chargeable directly to the fan, it is caused either by excessive peripheral speeds, or the fan is of insufficient size. It should be remembered, however, that the tip speed

TABLE 1. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR FORWARD CURVED MULTIBLADE VENTILATING FANS

STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
$\frac{1}{4}$	1000-1100	1520-1700
$\frac{3}{8}$	1000-1100	1760-1900
$\frac{1}{2}$	1000-1200	1970-2150
$\frac{5}{8}$	1100-1300	2225-2450
$\frac{3}{4}$	1200-1400	2480-2700
$\frac{7}{8}$	1300-1600	2660-2910
1	1500-1800	2820-3120
$1\frac{1}{4}$	1600-1900	3162-3450
$1\frac{1}{2}$	1800-2100	3480-3810
$1\frac{3}{4}$	1900-2200	3760-4205
2	2000-2400	4000-4500
$2\frac{1}{4}$	2200-2600	4250-4740
$2\frac{1}{2}$	2300-2600	4475-4970
3	2500-2800	4900-5365

required for a specified capacity and pressure varies with the type of blade, and that a tip speed which may be excessive for the forward curved type is not necessarily so for the backward or slightly backward type. A noisy fan usually is one which is operated at a point considerably beyond maximum efficiency.

For a given static pressure there is a corresponding outlet velocity and peripheral speed wherein maximum efficiency is obtained. If a fan is selected to operate at this point, the cost of operation and the noise can be held within control.

To aid in selecting fans as near as possible to the point of maximum efficiency, there are listed in Tables 1 and 2 for each static pressure corresponding outlet velocities and tip speeds which will give satisfactory results. The proper tip speed for a given static pressure varies with the design of wheel and with the number of blades or vanes in the wheel.

Lower outlet velocities than those listed in Table 1 may be employed, but care must be exercised to avoid selecting a fan for operation below its useful range. The useful range of the fans of Table 2 extends over the full length of the performance curve.

In exhaust ventilating systems where the air column moves toward the fan, noise due to the higher tip speeds and outlet velocities will not be so readily transmitted back through the air column to the building as when the air column is moving toward the rooms. Therefore higher outlet velocities may be used, but this will be at the expense of increased horsepower.

Ample large fans should always be used for both exhaust and supply systems, as there may be and usually is leakage despite the most careful workmanship, necessitating the delivery of more air at the fans than is exhausted from or supplied through the openings in the various rooms.

Long runs of distributing ducts, heaters, and air washers require definite increments of the total pressure which a supply fan in a ventilating system must overcome. These static pressures should be considered when selecting the fan characteristics, speed, and power.

TABLE 2. GOOD OPERATING VELOCITIES AND TIP SPEEDS FOR MULTIBLADE VENTILATING FANS WITH BACKWARD TIPPED AND DOUBLE CURVED BLADES

STATIC PRESSURE INCHES OF WATER	OUTLET VELOCITY FEET PER MINUTE	TIP SPEED FEET PER MINUTE
$\frac{1}{4}$	800-1100	2600-3100
$\frac{3}{8}$	800-1150	3000-3500
$\frac{1}{2}$	900-1300	3400-4000
$\frac{5}{8}$	1000-1500	3800-4500
$\frac{3}{4}$	1100-1650	4200-5000
$\frac{7}{8}$	1200-1750	4500-5300
1	1200-1900	4800-5750
$1\frac{1}{4}$	1300-2100	5300-6350
$1\frac{1}{2}$	1400-2300	5750-6950
$1\frac{3}{4}$	1500-2500	6200-7550
2	1600-2700	6650-8050
$2\frac{1}{4}$	1700-2800	7050-8550
$2\frac{1}{2}$	1800-2950	7450-9000
3	2000-3200	8200-9850

Fans picked within the limits of Table 1 will operate close to the point of maximum efficiency. No attempt has been made to select these limits for quiet operation, since this is a relative term and varies with the type and location of the installation.

The connection of a fan to a metallic duct system should be made by canvas or a similar flexible material so as to prevent the transmission of fan vibration or noises. Where noise prevention is a factor the fan and its driver should have floating foundations.

Fans for Drying

Both axial flow and centrifugal types of fans are used for drying work. Propeller fans are well adapted to the removal of moisture-laden air when operating against low resistance and when handling air at low temperatures. Motors on these fans usually are of the fully-enclosed moisture-proof types so that saturated air or air containing foreign material will not injure the motors.

Unit heaters employing axial flow fans are widely used in the drying

field. In drying, these fans may be used with unit heaters where not too much duct work is required and where air is to be delivered against pressure, since the noise developed from the high peripheral speed of these fans is not ordinarily objectionable in process work.

Centrifugal fans of the multiblade type generally are selected to supply air for drying, as they are capable of delivering large volumes of air against all pressures likely to be encountered.

Belt driven fans usually are to be preferred to direct-connected fans since efficient motor speeds do not usually coincide with efficient fan speeds. Replacement of a standard motor is quick and easy if it is belted.

Wherever drying is done throughout the year and where air requirements change as the drying conditions change, the drying can be speeded up or reduced through control of the fan capacity. This may be done by changing the fan speed or by varying the outlet area with dampers. A throttled outlet reduces the volume and reduces the power.

Due to the low speeds of forward curved multiblade or paddle-wheel type fans, these can be direct-connected to reciprocating steam engines, and the exhaust steam from the engines may be used in the heating apparatus. In selecting engine driven fans for drying processes, where a large quantity of exhaust steam is used in the heaters, a smaller fan and greater power consumption may be used, because power economy is not essential under this condition.

Where static pressure in a dryer varies, and where several fans must operate in parallel, fans are to be preferred which have a continuously rising pressure characteristic, such as is given by backward-curved or double-curved blades. This type of fan is well adapted for direct-connected motors of the higher speeds. (See Chapter 41 on Drying.)

Fans for Dust Collecting and Conveying

The application of fans for handling refuse, dust, and fumes generated by machine equipment is covered in Chapter 21. Information is given regarding the methods for determining air quantities, the velocity required for carrying various materials and the method of determining maintained resistance or total static pressure at which the fan is to operate. The selection of a proper size fan is at times governed by the future requirements of the plant. In many instances, additional future capacity is anticipated and should be provided for.

Having determined the necessary volume of air and the maintained resistance or static pressure required, the proper size fan may be selected from the fan manufacturers' performance charts or capacity tables. The fan chosen should be the size that will provide the required ultimate quantities with the minimum power consumption.

FAN CONTROL

Some method of volume control of fans usually is desirable. This may be done by varying the peripheral velocity or by interposing resistance, as by throttling-dampers. Both methods, since they reduce the volume of air, reduce the power required. In many installations adjustments of volume are desirable during varying hours of the day. In others an

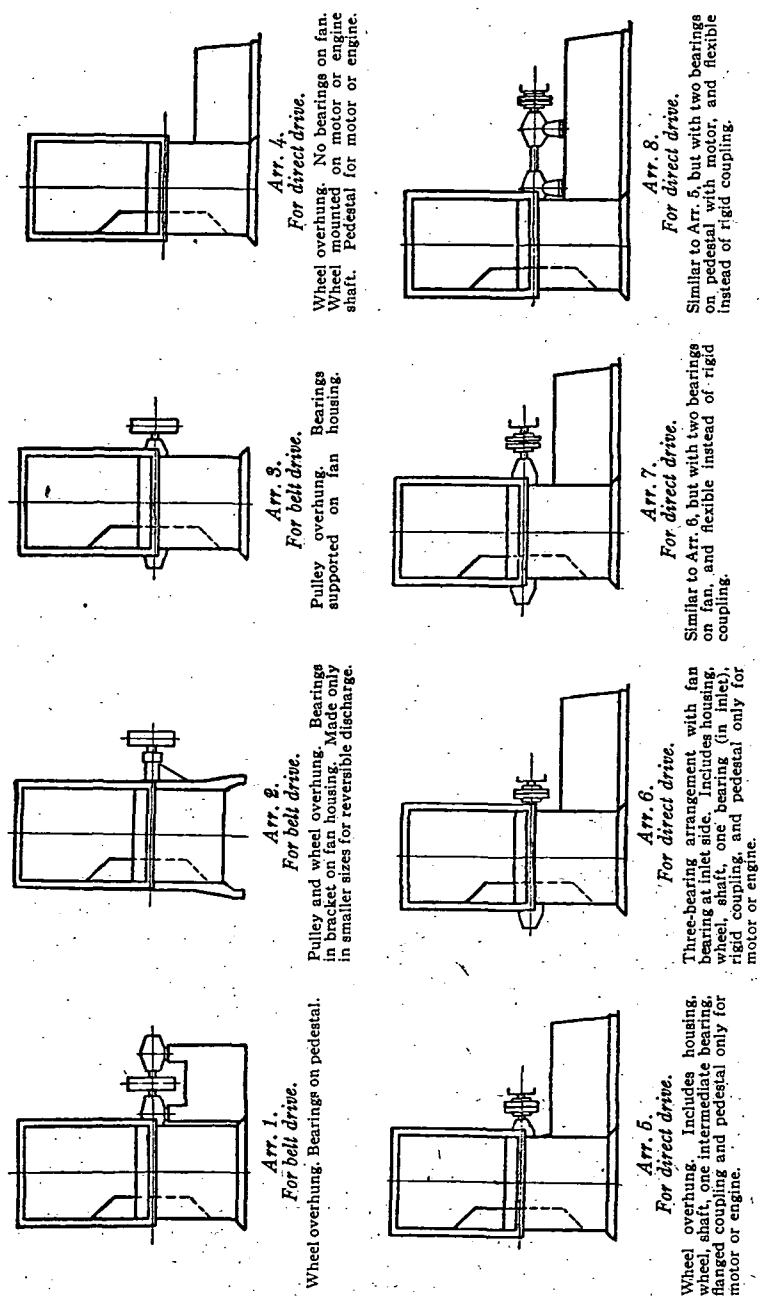


FIG. 6. ARRANGEMENT OF FAN DRIVES

increased supply of air in summer over that needed for winter is demanded. Experience is required in deciding whether speed-control or damper-control shall be used for specific cases. Where noise is a factor, it may be exceedingly desirable to reduce the speed at times, while on the other hand, any fan which has its normal speed reduced as much as 50 per cent without *change in resistance* will move only 50 per cent of the air.

DESIGNATION OF FANS

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-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.)^{*}

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.

In order to prevent misunderstandings, which cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 6 are suggested.

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. The *backward curved* and *double curved* types with backward tip operate satisfactorily in double or in parallel operation.

MOTIVE POWER

It is no easy matter to predetermine the exact resistance to be encountered by a fan or, having determined this resistance, to insure that no changes in construction or operation shall ensue which may increase air resistance, thus requiring more fan speed and power to deliver the required volume, or which may reduce air resistance, thus causing delivery of more air and a consequent increase of power even at constant speed.

It is recommended, therefore, for centrifugal type fans that the rated power to be supplied shall exceed the rated fan power by a liberal margin, when *forward curved* types are used. When *backward* or *double curved* blade types are used, motors with ratings very close to that of the fan horsepower demand can be employed, provided the fan has a limiting horsepower characteristic.

^{*}Recommendations adopted by the *National Association of Fan Manufacturers*.

Justification for liberal power provision exists also in the possibility of varying demand due to changes in ventilation requirements, intensity of occupation, and weather conditions.

The motive power of fans should be determined in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers*.

Fans may be driven by electric motors, steam engines (either horizontal or vertical), gasoline or oil engines, and turbines, but as previously stated the drive commonly used is the electric motor.

REFERENCES

- Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932.
- Fan Engineering, Buffalo Forge Company.
- Theories and Practices of Centrifugal Ventilating Machines, by D. Murgue, translated by A. L. Stevenson.
- Mechanical Engineers' Handbook, by Kent.
- Mechanical Engineers' Handbook, by Lionel S. Marks.
- Constructive Mechanism and the Centrifugal Fan, by George D. Beals.
- Coal Miners Pocket Book.
- The Fan, by Charles H. Innes.
- Mine Ventilation, by J. J. Walsh (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917).
- Fan Blower Design, by H. F. Hagen (A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922).
- The Centrifugal Fan, by Frank L. Busey.
- Section X, A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

PROBLEMS IN PRACTICE

- 1 ● What information must be supplied to the manufacturer when ordering a centrifugal fan?
 - a. Size of fan (catalog number).
 - b. Type of fan.
 - c. Width of fan (single or double).
 - d. Number of inlets (single or double).
 - e. Fan performance and kind of application.
 - f. Direction of rotation (clockwise or counter-clockwise).
 - g. Direction of discharge (top horizontal, down blast, etc.).
 - h. Drive arrangement (see Fig. 6).
 - i. Style of housing (full, three-quarters, etc.).
- 2 ● In selecting fans for quiet operation in public buildings:
 - a. Should the outlet velocity of the fan be limited?
 - b. Should the tip speed of the fan be limited?

a. Because all commercial fans operating at pressures suitable for this class of work would be considered noisy if the fan were to discharge directly into the room; and

because the duct system on the fan discharge is depended upon to absorb a reasonable amount of fan noise, it is desirable to have a moderate run of duct work with some bends and elbows included as sound deadeners. Where this duct is of necessity very short, the outlet velocity must be kept down to the lower limits recommended in this chapter or else an efficient sound absorber must be used. The experience of the engineer must be his guide in determining the allowable outlet velocity in each individual case.

b. Tip speed should not ordinarily be limited, because different types of fan blades have entirely different allowable tip speeds for quiet operation. A fan having a backward blade at the tip can run at much higher tip speed than can a forward curved or a straight blade fan, with the same degree of quietness.

3 ● Is a direct connected or a belted fan preferable in public building work?

Where space is at a premium, direct connection is best. Next in space economy is the short V-belt drive. The flat belt drive fan requires the greatest floor space. In this class of work, pressures are usually so low that even with the high speed fans the motor cost is greater for direct connected units than for belt drive fans.

4 ● a. What type fans are used in industrial work?

b. What outlet velocity is suitable?

- a. All of the centrifugal types are suitable; the disc and propeller types are suitable for low pressure work, or they are often used as exhausters.
- b. The outlet velocities on fans for industrial work can be much higher than can those in public building work, where quietness is essential. Fans should be selected with outlet velocities as recommended in this chapter, using the upper limit of velocities.

5 ● Are direct connected or belted fans preferred in industrial work?

In industrial applications, fans are often advantageously direct connected to motors. The pressures are usually high enough to use standard motor speeds. The high speed types of fans have limiting horsepower characteristics so that little margin in power must be provided in the driving motor. Belted fans may be used, but where high power is required a special arrangement is often necessary for shaft and bearings on account of the weight of the sheave and the belt pull.

6 ● A forward curved multiblade fan which requires 5.4 bhp is delivering 22,800 cfm at 70 F against a resistance pressure of 1 in. of water at an outlet velocity of 1440 fpm:

- a. What is the static efficiency?
- b. What is the total efficiency?

- a. 66.3 per cent (see Equation 2).
- b. 74.5 per cent (see Equation 3).

7 ● If the above fan has a 54-in. diameter wheel and operates at 193 rpm, will it be suitable for a ventilating installation where a minimum of noise is desirable?

Yes. The tip speed will be 2720 fpm and this, together with the 1440 fpm outlet velocity, falls within the limits given in Table 1 for 1-in. resistance pressure.

8 ● What objectionable feature is inherent in the ordinary propeller fan when it is operating at high resistance pressures?

It must operate at a high speed with consequent noise.

9 ● At what point should a fan be selected for operation, and why?

At its point of maximum efficiency because the cost of operation and the noise produced will be least.

10 • In Fig. 3, a static pressure of 85 per cent of blocked tight pressure corresponds to three different volumes, namely 11 per cent, 30 per cent and 48 per cent of wide open volume. What will determine which volume the fan delivers?

The fan can operate only at the intersection of its pressure-volume curve and the system characteristic. The type of system, together with the specification of the volume at a certain static pressure, completely defines the system characteristic.

As illustrated in Fig. 5, a given system characteristic will intersect the fan curve in only one point.

If the 85 per cent value for static pressure is specified for the 48 per cent value of volume, it is at once obvious that the same system will not have the same resistance at any other volume.

Chapter 18

SOUND CONTROL

Measurement of Noise, Noise in Buildings, Coefficients of Absorption, Insulation of Air-Borne Sound, Location and Insulation of Equipment Room, Insulation of Machinery and Solid-Borne Vibration, Control of Noise Transmission Through Ducts, Effect of Humidity upon Acoustics

THE ventilating and air conditioning of any space affect its acoustics and become apparent when consideration is given to the requirements for good hearing in any architectural interior. The requirements which must be given careful study are:

1. The room should be free from noise, whether of inside or outside origin.
2. The useful sound, whether speech or music, should be sufficiently loud (with reference to any residual noise) to be heard easily and distinctly.
3. The useful sound should be distributed uniformly in all parts of the room, and the sound reaching the listeners should be free from long-delayed reflections which produce interference or echoes.
4. The room should be free from pronounced resonant tones which may result from either volume or panel resonance.
5. The room should contain sound-absorptive materials in such amounts, and of such qualities, as will provide a proper balance between the persistence and cessation of the articulated components of sound, that is, the reverberation in the room should be long enough to sustain harmony and impart tonal blending to music, and at the same time it must be short enough to prevent the overlapping and confusing of the separate sounds of speech.

Obviously, the first of these requirements is the one which imposes restrictions on the installation of air conditioning or ventilating equipment—the equipment noises must be unobjectionable in occupied rooms—although the fifth requirement is not entirely independent of the humidity and temperature of the air.

LOUDNESS

Loudness is the sensation of sound intensity. When it is said that one sound is louder than another a difference in intensity level is implied. Two identical whistles when sounded together do not make a sound twice as loud as one. It may take ten to make a sound 20 per cent louder than one. It has been found that loudness bears a logarithmic relationship to intensity of sound. On this basis a scale of loudness has been built and a unit, the decibel (db), has been established. This scale is illustrated in Fig. 1 which shows the loudness of some typical noises. The formula for relating loudness and intensity is:

$$L_1 - L_2 = 10 \log_{10} \frac{I_1}{I_2} \quad (1)$$

where

L = Loudness in db.

I = Intensity.

Thus the two whistles made a noise $10 \log_{10} 2 = 3$ db louder than one whistle and the ten whistles, $10 \log_{10} 10 = 10$ db louder than one. It would take a hundred whistles to make a noise 20 db louder than one and a thousand to make a noise 30 db louder.

MEASUREMENT OF NOISE

Since the chief acoustical problem in the ventilating or air conditioning of a building consists of reducing equipment noise, it is necessary to describe methods for measuring noise. The measurement of noise is a relatively new problem, and although there are several reliable methods, there are as yet no standardized units, scales, or instruments for measuring noise¹. However, the *decibel* (db) already described is widely used in this country and England as the standard unit for noise or sound intensity—a unit of the same size, but called a *phon*, is used in Germany—and the zero level of the scale is a barely audible sound. Since the relation between subjective loudness and sound intensity is dependent upon pitch, it is customary to refer loudness to a single frequency. A 1000-cycle tone is generally accepted as the reference frequency, that is, the loudness of any sound is rated in terms of an equally loud 1000-cycle tone. Thus, a noise of 50 db means that the noise would be judged to be of the same loudness as a 1000-cycle tone which is 50 db above the normal threshold of audibility for the 1000-cycle tone.

As the frequencies decrease below 1000 cycles, the ear becomes less sensitive, until at about 30 cycles, sounds are no longer audible regardless of their intensity. Similarly, for higher frequencies, the limit of audibility is reached around 7000 cycles. Thus, at frequencies below 1000 cycles, sounds of the same loudness must have a greater intensity than at 1000 cycles. This is particularly fortunate, as otherwise the low frequency sounds would mask all others.

Noise measurements are usually made by one of three methods. The first is the electrical instrument method, which uses a noise meter usually consisting of a microphone, an amplifier, and a galvanometer. Where such a meter is to measure the loudness of a noise without regard to the frequency distribution, it must contain a weighted network which electrically simulates the varying sensitivity of response of the ear to different frequencies. Where it is desired to analyze the character of the sound, filters which shut out all but certain bands of frequencies are used with the meter. A number of manufacturers make such meters.

The second method consists essentially of varying the intensity of an artificially generated sound until the noise generated is masked by the noise being measured. Obviously, this method is subject to human errors in observation to which the instrumental method is not, but in the hands of

¹See Proposed Tentative Standards for Noise Measurement, and Proposed American Tentative Standard Acoustical Terminology of the American Standards Association Sectional Committee on Acoustical Measurements and Terminology.

Also see How Sound is Controlled, by V. O. Knudsen (*Heating, Piping and Air Conditioning*, October, 1931), and Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (*A.S.H.V.E. TRANSACTIONS*, Vol. 38, 1932).

a careful observer quite satisfactory results may be obtained. One instrument used is the audiometer, which consists of a buzzer, an ear phone, and a rheostat. The phone is held a fixed distance from the ear while the resistance of the rheostat is varied until the sound of the buzzer, as transmitted electrically to the phone, can no longer be heard. Audiometers are available either for covering all frequencies, as in the noise meter, or for covering certain frequency bands only.

A third method of measuring noise, simple, yet sufficiently accurate for most field measurements, employs only three tuning forks and a stop watch. Forks having frequencies of 128, 512, and 2048 are recommended. The forks must be calibrated. That is, it is necessary to know for each fork (1) the initial intensity, in number of decibels above its threshold, immediately after it has received a standard hit or excitation, and (2) the damping rate, in decibels per second. These calibrations can be made in any well-equipped acoustical laboratory. A *standard hit or excitation* can be imparted to the fork by a felt-covered spring hammer, or simply by letting the fork fall from a vertical position through an arc of 90 deg, hitting a suitable pad (such as soft rubber or felt for the 128 and 512 forks and hard rubber for the 2048 fork). The average 512 steel fork will have an initial intensity, when held $\frac{1}{4}$ in. from the ear with the broad side of the prong facing the ear canal, of about 80 db, and will decay at a rate of about 1.0 db per second. Such a fork will remain audible about 80 sec in a perfectly quiet place, provided the listener has normal hearing. In the presence of a noise, it will remain audible until its tone is just *masked* by the noise. Thus, if a 512 fork, having an initial intensity of 80 db and a damping rate of 1.0 db per second, should be found to remain audible 35 sec in the presence of a certain noise, the masking effect of the noise is 80 - 35, or 45 db.

Procedure

The method of measuring any noise is as follows: The observer, in the presence of the noise, strikes the 128 fork a standard blow. At the same instant he starts a stop watch. The fork is then held in front of the ear canal, and moved back and forth slightly, until the tone of the fork is just completely masked by the noise, at which instant the watch is stopped. This measurement is repeated at least two times. The average time is subtracted from the time the 128 fork remains audible in a quiet place. This difference multiplied by the damping rate of the fork gives the masking effect of the noise at 128 cycles. Similar measurements are made with the 512 and 2048 forks. Measurements of this type give a satisfactory description of both the intensity and the frequency distribution of the noise. The average masking effect of the noise at 128, 512, and 2048 cycles will usually be about 5 to 10 db less than the reading given by a noise meter.

NOISE IN BUILDINGS

Measurements of the intensity of speech, music and noise in many buildings, with special consideration of the noise produced by ventilating equipment, have given the results indicated by Fig. 1. The equivalent loudness of sounds in buildings varies from less than 10 db near the outlet of an air duct in a very quiet sound studio to nearly 100 db in a noisy boiler factory. It will be noted that the noise from the ventilating

fan in a certain high school auditorium was nearly as loud as average speech in a large auditorium. Such an amount of noise is devastating to good acoustics; in fact, it is impossible to hear speech in the presence of such a noise.

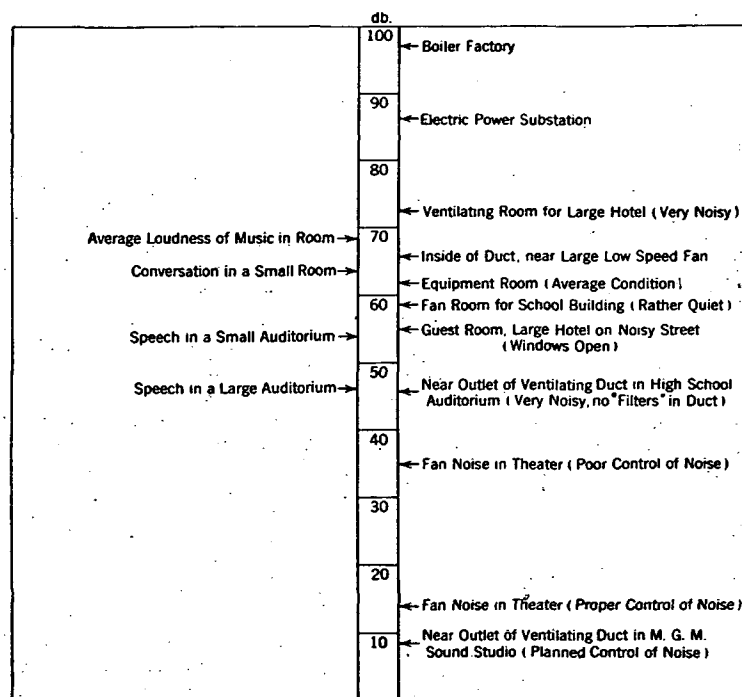


FIG. 1. CHART SHOWING THE EQUIVALENT LOUDNESS (IN DECIBELS) OF SPEECH, MUSIC, AND A NUMBER OF NOISES INCIDENT TO THE VENTILATING OF BUILDINGS*

*Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

In every problem of noise reduction in buildings it is necessary to know how much noise can be tolerated. The noise levels given in Table 1 may be regarded as completely inoffensive. They represent what might be termed ideal conditions, not often realized in existing buildings. However, they represent conditions which can be attained by proper control of noise, and heating, ventilating and air conditioning engineers should aim to provide the degree of quiet specified in the table.

In considering the tolerable room noise level due to heating, ventilating, or air conditioning apparatus, not only must the absolute value of the noise be considered but also its relation to the room noise level without the apparatus running. This is necessary since a large increase of noise subjects the apparatus to serious criticism even though the level may be low. It must also be borne in mind that the noise produced by the ap-

paratus is additive to that of the room without apparatus. Thus if the two are equal, when combined the noise level will be 3 db higher. For these reasons the room noise caused by the apparatus should not exceed the other room noise.

Noise Control

Essential to the design of a satisfactory system are: *first*, a knowledge of the nature and intensity of the noise generated by the various parts of the equipment; *second*, a knowledge of how to vary the noise level between the apparatus and the conditioned room if need be; *third*, a knowledge of the acceptable level of apparatus noise in the conditioned room. Besides these, the engineer must be able to deal with other noises which might enter the room when openings are made into it, such as *cross talk* between rooms connected with common ducts, and noise

TABLE 1. ACCEPTABLE NOISE LEVELS

Talking picture studios	6 to 8 db
Radio broadcasting studios	8 to 10 db
Hospitals	8 to 12 db
Music studios	10 to 15 db
Apartments, hotels, homes, small private offices	10 to 20 db
Theaters, churches, auditoriums, classrooms, libraries	12 to 24 db
Talking picture theaters, small clothing stores	15 to 25 db
General offices	20 to 30 db
Large public offices, banking rooms, upper stories of department stores, restaurants, barber shops	25 to 35 db
Grocery stores, drug stores	30 to 50 db
Accounting and typewriting offices	35 to 45 db
Main floor of department stores	40 to 50 db

transmitted to portions of duct systems outside the conditioned room and thence to its interior.

The problem of apparatus noise is receiving the study of equipment manufacturers who are aiming at both noise reduction and standardization. Some manufacturers now have noise ratings available for their equipment, while some pass each unit of equipment of certain types through sound tests during the course of manufacture.

ROOM NOISE LEVEL, COEFFICIENTS OF ABSORPTION

The problem of noise reduction from apparatus to room must take into consideration and treat separately the three modes of travel of noise to the room: *first*, from the apparatus through the air to the walls of the room and thence to its interior; *second*, through the building structure to the room; *third*, through ducts or openings to the room. Because the noise entering by each of these three channels is susceptible to quantitative analysis, solutions are available. Along with the transmission of sound through the building structure, the engineer must also consider the transmission of vibration, which may also be objectionable. The solution is not complete, however, until the effect of the noise entering the room on the room noise level is determined.

One of the most effective means of reducing noises in ventilating equipment is accomplished by the proper covering of the interior walls and

ceiling of the equipment room, or the inner walls of the ducts, with sound-absorptive materials. The intensity I of a continuous sound in a room is

$$I = \frac{E}{a} \text{ or } \frac{I' S'}{a} \quad (2)$$

where

E = the rate of emission of the noise source = $I' S'$. (The intensities of noises entering the room times the areas through which they enter.)

a = the total amount of absorption supplied by the boundaries and contents of the room.

= $a_1 S_1 + a_2 S_2 + a_3 S_3 + \dots$, where $S_1, S_2, S_3, \dots$ are the areas of the boundary materials for the room, and $a_1, a_2, a_3, \dots$ are the corresponding coefficients of absorption.

Hence, by increasing tenfold the absorptivity of the boundaries of a room it is possible to reduce tenfold the average intensity of sound in the room; that is, the intensity level would be reduced 10 db.

Thus it is possible to compute the noise level in the room if the intensity of noises entering the room or generated in it are known.

It will be seen that the noise intensity reduction is dependent upon the amount of sound absorption in the room, and that the first units of absorption are more effective than succeeding units. In general, the room noise level will be from 10 to 20 db lower than the air inlet or outlet noise intensity, the 10 db being in the case of bare rooms having large ventilating or air conditioning openings in relation to their size, and the 20 db in the case of rooms having large amounts of absorptive material with small openings. In some cases, the noise level reduction may run up to as much as 30 db, but then the higher sound intensity adjacent to the openings tends to nullify the effects of the extra reduction. Where these openings are large, the local effect on the noise intensity extends some distance from the opening; for instance, a four-square-foot opening might have a local effect within ten feet, while a one-half-square-foot opening would have a local effect within only five feet.

The coefficients of sound-absorption for a number of standard absorptive materials used, or suitable for use, in equipment rooms are given in Table 2. Coefficients are given for frequencies of 128, 512, and 2048 cycles. Where the frequency of the noise is not known, the values for 512 or 128 cycles are usually used.

INSULATION OF AIR-BORNE SOUND

The transmission of air-borne sounds through rigid partitions is accomplished primarily by the diaphragm-like vibrations of the partition. The weight per square foot of the wall is the determining factor, and the insulation value of a wall, in terms of the transmission loss in decibels, is proportional to the logarithm of the weight per square foot. Other factors, such as size, stiffness, composition; manner of mounting, and the use of multiple structures separated by air spaces or flexible connectors, contribute to the effective insulation. If the coefficients of sound transmission of different types of structures and the noise intensity in the space adjoining a room are known, it is possible to calculate the noise intensity in a room by the use of formula 1 and the following formula:

$$I = I' \tau \quad (3)$$

where

I' = noise intensity in space adjacent to room.

τ = coefficient of sound transmission.

Coefficients of sound transmission for some common walls are shown in Table 3.

Example 1. Suppose the brick wall between an equipment room and an adjacent auditorium has an area of 200 sq ft and a coefficient of sound of 0.00001 (see Table 3); that the auditorium contains 2000 sabines^a of absorption; and that the noise level in the equipment room is 70 db above zero level.

$$70 - 0 = 10 \log_{10} \frac{I'}{I_0} \quad (\text{from Formula 1})$$

$$\frac{I'}{I_0} = 10^7$$

$$\frac{I'}{I_0} = 10^7 \times 0.00001 = 100 \quad (\text{from Formula 3})$$

$$\frac{I}{I_0} = 100 \times \frac{200}{2000} = 10 \quad (\text{from Formula 2})$$

TABLE 2. COEFFICIENTS OF SOUND ABSORPTION^a

MATERIAL	THICKNESS (INCHES)	COEFFICIENTS OF SOUND ABSORPTION		
		128 Cycles	512 Cycles	2048 Cycles
Acoustex 60, spray painted.....	1	0.16	0.51	0.72
Acousti-Celotex, Single B.....	5/8	0.11	0.45	0.68
Acousti-Celotex, Triple B.....	1 1/4	0.20	0.75	0.67
Acoustic Flexfelt.....	-----	0.27	0.56	0.68
Acoustone.....	1	-----	0.66	0.69
Akoustolith plaster.....	1/2	0.21	0.29	0.37
Akoustolith A, Tile.....	1	0.14	0.48	0.83
Brick wall, unpainted.....	18	0.024	0.031	0.049
Calicel.....	1	0.23	0.72	0.71
Corkoustic, Type C.....	1 1/2	0.08	0.61	0.64
Glass.....	-----	0.035	0.027	0.020
Insulite Acoustile, Type 44.....	1 3/8	0.26	0.50	0.61
Kalite, with three coats lacquer.....	3/4	0.35	0.43	0.45
Macoustic Plaster, stippled to depth of 1/2 in.....	1 1/2	0.13	0.31	0.58
Masonite.....	1 1/8	0.18	0.32	0.33
Plaster, gypsum on hollow tile.....	-----	0.013	0.020	0.040
Plaster, gypsum, scratch and brown coats on metal lath on wood studs.....	-----	0.020	0.040	0.058
Plaster, lime, sand finish, on metal lath.....	3/4	0.038	0.060	0.043
Poured concrete, unpainted.....	-----	0.010	0.016	0.023
Rockoustile.....	1	0.18	0.57	0.72
Sabnite.....	1 1/2	-----	0.34	0.49
Sanacoustic Tile.....	1 1/4	0.19	0.79	0.74
Stuccoustic Plaster, Type XB.....	3/4	0.29	0.59	0.72
Transite Tile.....	1	0.19	0.81	0.72
Trutone Tile.....	1 1/8	0.31	0.57	0.64
Wood sheathing, pine.....	3/4	0.098	0.10	0.082
Wood, varnished.....	-----	0.05	0.03	0.03

^aArchitectural Acoustics, by V. O. Knudsen, pp. 219, 220, 240-251.

^AA sabine is 1 sq ft of totally absorptive surface.

Room loudness = $10 \log_{10} 10 = 10$ db

If the sound absorption in the auditorium had been as small as 200 sabines, the sound intensity in the auditorium would have been 10 times as great and the noise level in the auditorium would have been 20 db.

If the rest of the auditorium has an area of 20,000 sq ft with a surrounding noise intensity of 50 db ($I^n = 10^5$) the noise level due to all of the noise entering through the wall would be found as follows:

$$\frac{I}{I_0} = 10^5 \times 0.00001 = 1$$

$$\frac{I}{I_0} = 10 \text{ (Through equipment wall)} + 1 \times \frac{20,000}{2000} = 20$$

Room loudness $\times 10 \log_{10} 20 = 13$ db

Now suppose that there is also a duct having 20 sq ft outlet connecting the room with apparatus having a noise level of 70 db ($I^n = 10^7$) and suppose that there is an assumed attenuation in the duct equivalent to a transmission factor of 0.0002. Then,

$$\frac{I}{I_0} = 10^7 \times 0.0002 = 2 \times 10^3$$

$$\frac{I}{I_0} = 20 \text{ (from above)} + 2 \times 10^3 \times \frac{20}{2000} = 40$$

Room loudness = $10 \log_{10} 40 = 16$ db

It may be seen how the energies of noises entering a room are added to obtain the final room noise intensity.

The average coefficients of sound transmission (128 to 4096 cycles) for a number of walls and of floor and ceiling partitions are listed in Table 3.

TABLE 3. AVERAGE COEFFICIENTS OF SOUND TRANSMISSION FOR BUILDING PARTITIONS*

DESCRIPTION OF PARTITION	AVERAGE COEFFICIENT
Brick panel, Mississippi, 8 in.; plastered both sides gypsum brown coat, smooth white finish; good workmanship	0.000010
Brick wall, 2½ in. plaster both sides	0.000032
Brick wall, 2½ in., 2-in. furring strips, ½ in. rigid insulation lath plastered both sides	0.000016
Brick wall, 4 in., 2-in. furring strips and ½ in. rigid insulation lath, plaster, on one side; other side plastered directly on brick	0.000040
Concrete flat slab floor construction, reinforced; floating floor consisting of nailing strips, rough and finish flooring; ½ in. rigid insulation furred out and applied as ceiling	0.000020
Glass, plate ¼ in.	0.0010
Glass, plate ¼ in. double glazed, 1½ in. separation	0.0001
Metal lath, double, on 1½ in. channels, ¾ in. gypsum plaster; without cross bracing clips; 4 in., connected at edges only	0.000016
Tile, hollow clay partition, three cells, 4 in. x 12 in. x 12 in., wood furring strips, ½ in. rigid insulation, gypsum brown coat, smooth white finish	0.000050
Wood joists, lower side plastered on wood lath; floating floor consisting of nailing strips, rough and finish flooring	0.000050
Wood studs, two ½ in. sheets rigid insulation both sides, joints filled, gypsum scratch and brown coats, smooth white finish	0.000013
Wood studs, 2 in. x 4 in., staggered, metal lath ½ in. gypsum plaster; 7½ in.; connected at edges only	0.000040

*Architectural Acoustics, by V. O. Knudsen, pp. 308-322.

LOCATION AND INSULATION OF EQUIPMENT ROOM

The equipment room, if possible, should be located at a considerable distance from all rooms in which quiet is required. If this is not possible, it is necessary to provide a high degree of insulation against the noise which may be transmitted through the walls of the equipment room, and also against the noise which almost certainly will be communicated through the short ducts. (See discussion of Control of Noise Transmission through Ducts, p. 337). Three wall sections and two floor and ceiling sections which are satisfactory for the wall insulation of the equipment room are shown in Fig. 2. Other partitions, with their sound insulating values, are listed in Table 3. The addition of absorptive materials (such as are described in Table 2) to the inner walls and ceiling

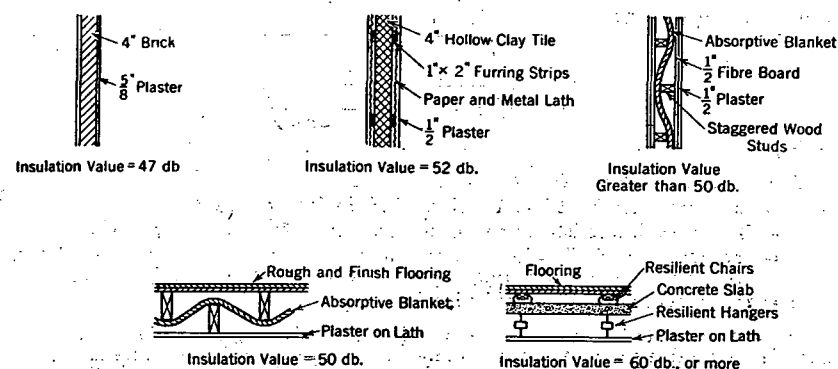


FIG. 2. THREE WALL SECTIONS AND TWO FLOOR AND CEILING SECTIONS WHICH ARE SUITABLE FOR THE INSULATION OF EQUIPMENT ROOMS*

*Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

of the equipment room will not only increase the insulation through the walls, but will also reduce the intensity of the noise in the room. The equipment room noise intensity may be figured in the same way as that of the conditioned space, taking the equipment as the source of noise. In case the equipment is subject to considerable vibration it is advisable to provide a separate or *floating* floor.

INSULATION OF MACHINERY AND SOLID-BORNE VIBRATION

Since mechanical vibrations are readily transmitted through the solid structure of a building, it is extremely important in air conditioning that all mechanical equipment in which vibrations are generated be thoroughly insulated from the solid structure of the building. An almost universal notion prevails that the vibrations generated by machinery can be insulated from a building simply by placing a slab of cork or a layer of hair felt between the machinery and the floor of the room. If the machinery is sufficiently heavy, and the cork or felt sufficiently resilient, this expedient may suffice. On the other hand, if the machinery is not sufficiently heavy to *load* the cork or felt support to the extent that the

natural frequency of the machinery on the cork or felt is low in comparison with the frequency generated by the equipment, the cork or felt may be of little avail. The insulation of vibration can be accomplished by means of suitable elastic supports or suspensions, but the design of these elastic supports should be based upon calculation rather than guess-work.

The theory of the insulation of vibration was first worked out by Soderberg³. If a machine of mass m be supported by an elastic pad the amount of vibratory force communicated by the machine to the floor or foundation upon which it rests will be determined by the elastic and viscous properties of the pad. The ratio of the vibratory force communicated to the floor or foundation with the machine resting upon the pad, and with the machine resting directly upon the floor, is given by the following equation:

$$\tau^1 = \sqrt{\frac{r^2 + \frac{1}{4\pi^2 n^2 c^2}}{r^2 + \left(2\pi n m - \frac{1}{2\pi n c}\right)^2}} \quad (4)$$

where

τ^1 = the so-called *transmissibility* of the support.

c = the compliance (that is, the reciprocal of the force constant).

r = the mechanical resistance owing to the viscous forces within the support.

n = the frequency of vibration generated by the machine which is to be insulated, such as the commutation frequency of a motor or the blade frequency of a fan.

m = the mass of the machine to be insulated.

It should be noted that not only must vibrations within the audible range of frequencies be considered, but those in the sub-audible range as well, since these may cause objectionable vibrations. All the possible frequencies should be considered in the calculation. Sometimes beat effects are introduced by slight irregularities of belts or pulleys that have much lower frequencies than those of the rotating elements.

If the pad is to be of any value in the prevention of solid-borne vibrations, the value of τ^1 must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be established, and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. It is necessary, therefore, that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 50 vibrations per second, and for best results preferably 20.

When the forced frequency is low, it is frequently impossible to insulate for the fundamental forced frequency due to connecting pipe work and other relevant factors. In cases of this kind an effective installation of

sound insulation may be obtained with a mounting which functions far above the fundamental forced frequency. For example, a compressor operating at 500 rpm has a forced frequency of 8.3 vibrations per second. By designing a mounting having a natural frequency of 20 to 25 vibrations per second, it is possible to isolate practically all of the noise.

If a slab of insulating material be placed under the entire foundation of a machine, as is often done in practice, it may happen that the natural frequency of the machine on its elastic support will be nearly the same as the frequencies which are to be insulated, in which case the elastic support will be worse than nothing. In general, as Equation 4 shows, both m and c should be as large as possible if the vibrations of the machine are to be effectively insulated from the solid structure of the building. Furthermore, the machine should rest upon a rigid floor so that the elastic yielding of the floor is prevented from communicating the machinery vibrations to the solid structure of the building.

The elastic support under the machine acts as a low-pass filter which passes all frequencies below about two times the natural frequency of the machine mounted on its elastic support, but presents all frequencies above about $\sqrt{\frac{mc}{\pi}}$ from reaching the solid structure of the building. The

principal influence of the internal mechanical resistance r is to limit the vibration at the resonant frequency. It is generally advisable, therefore, to use materials which have an appreciable internal resistance.

The values of c and r can be determined for any specimen of flexible material and, when known, can be used to determine the insulation value of any particular set-up. The value of c can be obtained by making static measurements of the amount of displacement of the compressed support for each additional unit of the compressing force. If this be done for a specimen of the flexible material of a certain thickness and area of cross section, the compliance can be determined for any other thickness or area from the relation that c will be directly proportional to the thickness and inversely proportional to the area of the flexible support. When the internal resistance r is not too large, it can be determined by observing the successive amplitudes of the free vibrations of a mass m which rests upon a specimen of the flexible material, and solving for r by the usual log-decrement method. Or, if the damping be so great that the free motion of m is non-oscillatory, r can be obtained from measurements on the experimentally-determined resonance curve of the forced vibrations of m , or from measurements of the rate of return of m when it is given an initial displacement.

If the resistance of a certain specimen of material, as cork, felt, or rubber, has been determined by any of these methods, the resistance for any other thickness or area of the material can be determined approximately because the resistance will be inversely proportional to the thickness and directly proportional to the area of cross-section of the flexible support. Thus, if the values of c and r for a flexible material be known, it is possible to calculate, by means of Equation 4, the amount of insulation that will be obtained from the use of this material as a flexible support for a piece of equipment having a mass m . For the routine calculations in practice, r may be neglected with only a slight sacrifice

³C. R. Soderberg, *The Electric Journal* (January, 1924), and succeeding articles. See also V. O. Knudsen, *Physical Review*, Vol. 32, 1928, p. 324, and A. L. Kimball, *Journal Acoustical Society of America*, Vol. 2, 1930, p. 297.

of accuracy. Table 4 gives the values of c and r for a number of commonly used flexible materials.

In general, there are two principal points to observe in the design of a flexible support for any piece of equipment, namely, the material should have a relatively large compliance and *it should be loaded to nearly the upper safe limit of loading*. Several flexible metallic supports have recently been developed.

Example 2. A machine weighing 1000 lb. has a base area of 20 sq ft. Assume that the principal vibration of the machine has a frequency of 100 cycles per second (most machinery vibrations are less than 150 vibrations per second; and the assumed frequency of 100 is quite representative of typical machines). Suppose that a 1-in. slab of corkboard weighing 1.10 lb per board foot be placed between the machine and the floor. The loading on the cork will then be only 50 lb per square foot, or slightly more than $\frac{1}{2}$ lb per square inch. (It is assumed that the compliance c in centimeters per dyne for a specimen 1 in. thick and 1 sq cm in cross-section is 0.25×10^{-6} and the resistance r in mechanical ohms is 0.15×10^5 .)

The *transmissibility* is calculated in the following manner:

$$\text{Mass of machine in grams} = 1000 \times 454 = 4.54 \times 10^5$$

$$\text{Area of base in square centimeters} = 20 \times 144 \times 2.54 \times 2.54 = 1.86 \times 10^4$$

Therefore, the compliance of the entire support, 1 in. thick and 20 sq ft in cross section, is $0.25 \times 10^{-6} \times \frac{1}{1.86 \times 10^4} = 0.134 \times 10^{-10}$ cm per dyne, and the resistance of

TABLE 4. COMPLIANCE AND RESISTANCE DATA FOR TYPICAL SPECIMENS OF FLEXIBLE MATERIALS^a

The compliances and resistances given in the table are for specimens 1 in. thick and 1 sq cm in cross-section.

MATERIAL	DESCRIPTION OF MATERIAL	APPROXIMATE UPPER SAFE LOADING IN POUNDS PER SQUARE INCH	COMPLIANCE c IN CENTIMETERS PER DYNE	RESISTANCE r IN ABSOLUTE UNITS
Corkboard	1.10 lb per board foot	12	0.25×10^{-6}	0.15×10^5
Corkboard	0.70 lb per board foot	8	0.50×10^{-6}	0.25×10^5
Fiber Board	1.35 lb per board foot	4 to 6	0.60×10^{-6}	0.50×10^5
Fiber Board	Carpet lining	10	0.40×10^{-6}	
Fiber Board	Insulating board	12	0.18×10^{-6}	
Fiber Board	Insulating board	15	0.16×10^{-6}	
Fiber Board	Insulating board	15	0.12×10^{-6}	
Anti-Vibro-Block		5	0.60×10^{-6}	1.5×10^5
Sponge Rubber	25 lb per cubic foot	1 to 3	3.0×10^{-6}	
Soft India Rubber	55 lb per cubic foot	3 to 6	1.2×10^{-6}	
Hairfelt	10 lb per cubic foot	1 to 2	1.5×10^{-6}	

^aFrom *Architectural Acoustics*, by V. O. Knudsen, p. 278.

the entire support is $0.15 \times 10^5 \times 1.86 \times 10^4 = 0.28 \times 10^9$ mechanical ohms (or absolute units). Therefore,

$$\tau' = \sqrt{\frac{(0.28 \times 10^9)^2 + \frac{10^{20}}{4\pi^2 \times 100 \times (0.134)^2}}{(0.28 \times 10^9)^2 + \left(2\pi \times 100 \times 4.54 \times 10^5 - \frac{10^{10}}{2\pi \times 100 \times 0.134}\right)^2}} = 0.93$$

Consequently, it is seen that the *transmissibility* is nearly equal to unity, and that the support therefore is not satisfactory for insulating 100 or fewer vibrations per second.

If the amount of cork be reduced so that it is loaded to 10 lb per square inch, the total area of the supporting cork will be only 100 sq in. or 645 sq cm. The compliance of the entire support will now be $0.25 \times 10^{-6} \times \frac{1}{645} = 0.39 \times 10^{-9}$ cm per dyne, and the resistance will be $0.15 \times 10^5 \times 645 = 0.97 \times 10^7$ mechanical ohms (or absolute units). Therefore

$$\tau' = \sqrt{\frac{(0.97 \times 10^7)^2 + \frac{10^{18}}{4\pi^2 \times 100 \times (0.39)^2}}{(0.97 \times 10^7)^2 + \left(2\pi \times 100 \times 4.54 \times 10^5 - \frac{10^9}{2\pi \times 100 \times 0.39}\right)^2}} = 0.037$$

It is seen, therefore, that with the bearing surface on the cork reduced to 100 sq in. (that is, with the cork loaded to 10 lb per square inch), the *transmissibility* is reduced to 0.037, or the amplitude of vibration transmitted to the floor will be only about 1/27 of what it would be if the machine were mounted directly upon the floor. These two numerical examples will serve to show not only the manner of making the calculations, but also the importance of selecting the proper type and design of flexible supports for insulating the vibrations of a machine from the rigid structure of a building.

CONTROL OF NOISE TRANSMISSION THROUGH DUCTS

The most troublesome sources of noise from ventilating and air conditioning equipment are fan and motor noises which are transmitted through the ducts. The reduction, in decibels, of noise transmitted through a duct, neglecting reflection from ends and bends, is proportional (1) directly to the length of the duct, (2) directly to the perimeter of the duct, (3) inversely to the area of cross-section of the duct, and (4) directly (or at least approximately so) to the coefficient of sound absorption of the material which comprises the interior surface of the duct. It is apparent therefore that long narrow ducts, lined with highly absorptive material, will provide a high degree of insulation against the transmission of noise through ducts. In fact, small ducts (4 in. x 6 in.), made of material having a coefficient of sound-absorption of 0.50, will provide a noise reduction of slightly more than 1 db per linear foot.

As can be seen from an inspection of Table 2, noises of low frequency are difficult to absorb; on the other hand, these frequencies are easily reflected by elbows, branches, and duct ends whereas higher frequencies are little affected. Furthermore, the reflection effects are more pronounced in small ducts than in large ducts. Hence, by introducing into a duct a sufficient length of small, absorptive channels together with a number of elbows or other reflecting elements it is possible to reduce the

transmitted noise to any required degree. This applies not only to ducts between the equipment room and other rooms in a building, but also to ducts connecting adjacent or nearly adjacent rooms. By the proper use of such *filters* it is possible to eliminate all of the difficulties which arise in connection with the transmission of sound through ventilating ducts. The problem is an engineering one which can be worked out prior to the installing of the equipment, and it can be calculated in such a way as to meet the most rigorous demands for silent operation. There is a need for quantitative data regarding the attenuation or noise-reduction provided by different types of ducts, but even with the meager data available it is possible to design filters which will suppress the ordinary noises incident to the ventilating or air conditioning of buildings⁴.

In general, the motion of air resulting from the ventilating of rooms is not sufficient to introduce any appreciable difficulty in auditoriums, except where noise may originate from the issuing of high-speed air from nozzles. However, by proper stream-lining of the nozzles, it is possible to work with speeds which are adequate for all practical purposes without producing any disturbing noises. Since sound is propagated with a velocity of more than 1100 fps, the velocity of the air would have to attain speeds of at least 20 to 30 fps before these wind velocities would have any appreciable influence upon the propagation of sound.

If there is to be any appreciable motion of air in an auditorium, it is advantageous to have the upper layers of air moving in a direction from the stage toward the audience, as this will tend to refract the sound waves down toward the audience. However, unless the speed of the air is as great as 20 or 30 fps, the amount of refraction will not be noticeable. Therefore, as a rule the motion of air in an auditorium does not have an appreciable effect upon the acoustical properties of the room.

EFFECT OF HUMIDITY UPON ACOUSTICS

Recent experiments⁵ have shown that both the humidity and the temperature of air have a marked influence upon the rate of absorption of high-pitched sounds. Perfectly dry air is more absorptive than air containing any amount of water vapor. At relative humidities of 5 to 25 per cent, the air is highly absorptive but becomes less and less absorptive as the humidity is increased. High-frequency sounds are propagated better in cold humid air than in hot dry air, and since high-frequency sounds are particularly important for the preservation of good quality in speech and music it is advantageous to maintain the air in a room at a relatively high humidity, not less than about 55 to 60 per cent. On the other hand, where it is desirable to absorb all frequency components of sound, as for the reduction of noise in offices, it is advantageous to maintain relatively dry air.

The time of reverberation in a room is given by the following equation:

$$t = \frac{0.049V}{AmV - S \log_e (1 - \alpha)} \quad (5)$$

⁴How Sound is Controlled, by V. O. Knudsen (*Heating, Piping and Air Conditioning*, October, 1931).

⁵Effect of Humidity upon the Absorption of Sound in a Room, by V. O. Knudsen (*Journal Acoustical Society of America*, July, 1931). Also see report presented at the May, 1933, meeting of A. S. of A.

where

V = volume of room in cubic feet.

S = interior surface of room.

α = average coefficient of sound-absorption of the interior surface of the room.

m = the absorption coefficient of the air in the room.

The coefficient m depends upon the frequency of the sound and the humidity (and probably the temperature) of the air. At a temperature of 70 F, and for sound waves having a frequency of 4096 vibrations per second, $m = 0.0027$ at 25 per cent relative humidity, 0.0018 at 54 per cent, and 0.0013 at 82 per cent. It will be seen, therefore, that the absorption of sound in the air is twice as great at a relative humidity of 25 per cent as it is at a relative humidity of 82 per cent. This explains why sounds in the open travel so much better on humid days than they do on dry days. Although this dependence of absorption upon humidity is characteristic of low-frequency as well as high-frequency sound, the actual amount of absorption in the air is negligible for frequencies below about 1024 vibrations per second. However, the absorption of the higher frequencies in the air is a significant factor, and its dependence upon humidity calls for careful consideration in planning the air-conditioning equipment for buildings.

PROBLEMS IN PRACTICE

1 ● What are the characteristics of a good duct lining material for the absorption of sound?

A good duct lining should have the following characteristics: (1) high absorption qualities, (2) proper physical strength, (3) fire resistant, (4) clean with an absence of loose fibers or pieces, (5) smooth surface to reduce air friction, and (6) simple and easy application to existing or new installations.

2 ● Why do modern improvements in the acoustics and air conditioning of buildings present new acoustical problems to the heating and ventilating engineer?

In acoustically treated rooms, both outside and inside noise are reduced, and consequently the noise of ventilating equipment becomes more noticeable. The closed windows in air conditioned buildings exclude outside noise, which makes all inside noise from mechanical equipment seem louder.

3 ● Name the acoustical problems which should be solved in connection with the installation of heating or air conditioning equipment.

Selection of quietly operating equipment; adequate insulation of walls surrounding the equipment room; mounting of all vibrating equipment on flexible supports which will eliminate solid-borne vibrations; design of suitable sound-filters to reduce the transmission of noise through ventilating ducts; the use of suitably low air speeds and stream-lining, where necessary, to prevent eddy noises.

4 ● Are good heat insulators also good sound insulators?

As a rule, no. Blankets and felted materials offer considerable insulation for sounds of high frequency, but very little for sounds of low frequency.

5 ● What is the principal consideration in the selection of elastic supports for the insulation of machinery vibration?

The support should be so compliant that the natural frequency of the mass of the machinery on its elastic support will be low in comparison with the vibrational frequencies which are to be insulated.

6 • What means should be utilized for preventing air-borne noise from the ventilating equipment from being transmitted through the walls, ceiling, or floor of the equipment room?

Treat the interior walls and ceiling of the equipment room with absorptive material; see that all doors and windows to the equipment room fit tightly in their frames; and use wall, and floor and ceiling partitions which have an insulation value of not less than 50 db.

7 • Name effective methods for reducing the transmission of sound through ventilating ducts.

Line the ducts with sound absorptive material, or use suitable sound filters made up of long channels of small cross-sectional area, lined with sound absorptive material.

8 • What are the effects of humidity and temperature on the absorption of sound in air?

The absorption increases with a rise in temperature, and decreases for relative humidities above about 20 per cent. A relative humidity of 55 to 60 per cent is advantageous acoustically in large auditoriums.

9 • How may sound be measured and what are the advantages of the methods available?

Three practical methods are now available to the heating and ventilating engineer, namely:

- The noise meter method.
- The audiometer and ear method.
- The tuning fork and ear method.

Except for instrument adjustments and the use of the eye in reading a meter, the human element does not enter into measurements made with the noise meter, so it is to be preferred, if available. The tuning fork method is relatively cheap and simple and sufficiently accurate for most field work. The audiometer and ear method ranks between these two in preference.

10 • What are some of the more important sources of noise in buildings, for which the heating and ventilating engineer may be held responsible?

- Furnace room equipment.
- Radiators and piping.
- Uncalked openings in walls around pipes and ducts.
- Ventilating fans, if noisy in operation and not isolated from the building structure by properly designed vibration damping foundations.
- High air velocity in ducts.
- Ventilation fan rooms not insulated acoustically from parts of the building where noise would be objectionable.
- Ventilating ducts without flexible non-metallic sleeves in them to break metallic sound conducting paths.
- Cross connection of rooms acoustically through ducts.
- Ventilating ducts without sound absorbing lining, if required.
- Unit heaters and ventilators.
- Unit air conditioners.

11 • The noise level in the fan room directly under the main floor of a theater is 70 db. The floor is constructed as described in Item 5, Table 3. What is the fan noise level in the theater?

According to Table 3, the average coefficient of sound transmission, t , of such a floor construction is 0.0000020. The transmission loss through the floor, expressed in db, is:

$$TL = 10 \log_{10} \frac{1}{t}$$

$$= 10 \log_{10} \frac{1}{0.0000020}$$

$$= 57$$

The fan noise level in the theater would, therefore, be 70 db less 57 db, or 13 db, which, according to Table 1, is an acceptable level.

Another way of arriving at the same result is by use of Formula 3, in which I' is the intensity of fan noise as measured in the theater, and I'' its intensity as measured in the fan room, I_0 being the reference intensity in both cases, while τ is 0.000002 or 2×10^{-6} .

$$\frac{I'}{I_0} = 10^7$$

$$\frac{I''}{I_0} = 10^7 \times 2 \times 10^{-6} = 20$$

$$\text{Noise level} = 10 \log_{10} 20 = 13 \text{ db.}$$

12 • Measurements made separately of the noises from different sources prevailing in a large, noisy banking room revealed the following average noise levels:

- From the street through windows, doors, and walls, 40 db.
- From adding machines, typewriters, human movements and conversation, 60 db.
- From the ventilating system, 50 db.

What was the total noise level of the room?

Calling I_s , I_b , and I_v the intensities of the street, banking room, and ventilation noises, respectively, and I_0 the reference level, we have:

$$\frac{I_s}{I_0} = 10^4 \qquad \frac{I_b}{I_0} = 10^6 \qquad \frac{I_v}{I_0} = 10^5$$

The total intensity, I , will be $I_s + I_b + I_v$

$$\text{The intensity level is } 10 \log_{10} \frac{I}{I_0}$$

$$= 10 \log_{10} \frac{(I_s + I_b + I_v)}{I_0}$$

$$= 10 \log_{10} (10^4 + 10^6 + 10^5)$$

$$= 60.4 \text{ db}$$

Note that the total loudness level is not much above the level of the loudest noise. While noise intensities may be added arithmetically, noise levels expressed in decibels cannot be so added.

13 • A ventilating fan room 30 ft by 30 ft by 12 ft has brick walls, a concrete floor, and a concrete ceiling. How much will the noise level of this room, expressed in decibels, be reduced by applying sound insulating material (coefficient of absorption 0.6 at 512 cycles) to two walls and the ceiling?

Use Formula 2:

$$I = \frac{I'S'}{a} \text{ before applying material}$$

$$I_1 = \frac{I'S'_1}{a_1} \text{ after applying material}$$

$$\frac{I}{I_1} = \frac{\frac{I'S'}{a}}{\frac{I'S'_1}{a_1}} = \frac{a_1}{a}$$

Referring to Table 2:

$$a = (4 \times 12 \times 30 \times 0.031) + (2 \times 30 \times 30 \times 0.016) = 73.4$$

$$a' = (2 \times 12 \times 30 \times 0.031) + (30 \times 30 \times 0.016) + (2 \times 12 \times 30 \times 0.6) + (30 \times 30 \times 0.6) = 1008.7$$

$$\frac{I}{I_1} = \frac{a'}{a} = \frac{1008.7}{73.4} = 13.7$$

$$\text{Noise level reduction} = 10 \log_{10} \frac{I}{I_1} = 10 \log_{10} 13.7 = 11.4 \text{ db.}$$

14 • What relation does the movement for the suppression of noise bear to the trend toward air conditioning of offices and other places in cities where people work or congregate?

Very important sources of disturbing sounds are the various street noises that gain entrance, not only through open windows but to a certain extent even through closed windows. If windows are to be kept closed to exclude noise, air conditioning is a practical necessity, especially in summertime. Summertime air conditioning makes use of awnings, which are not only desirable but economical in that they keep down cooling loads. To obviate condensation and frost on windows, wintertime air conditioning calls for storm sash or double glazing which in turn reduces the transmission of street noises.

Chapter 19

AIR DISTRIBUTION

Opening Characteristics, Residences, Stores and Offices, Auditoriums, Railroad Cars

TO produce proper air distribution in an enclosed space to be ventilated, heated, or cooled by air, the design and location of the air inlet and exhaust openings must be carefully considered. A system may fail, though it handles the proper amount of air at the required temperature, if such important design principles of air distribution are ignored. Since air temperature and quantity are now moderately easy to control and vary, air distribution becomes the prime factor, particularly when comfort air conditioning is involved.

Owing to the lack of practical and satisfactory instruments for measuring accurately air movement all air distribution information available may be justly criticized; but for practical design purposes this information is sufficiently accurate.

OPENING CHARACTERISTICS

Any temperature differential between an occupied space air and inlet air causes the air to rise or fall a given vertical distance in a given time. This vertical rise or fall of the air stream can be moved horizontally by velocity, and if the mass of supply air is not diffused, the horizontal movement may be accomplished without changing the given vertical distance or the given time. This horizontal movement at a constant velocity may be reduced by expanding the air stream either by deflection, changing the dimensional ratio of the stream, or breaking the air into a series of streams in close proximity.

With a few exceptions a high velocity should be used, with a large compact mass of air for a long air throw (Fig. 1), so that the periphery of the air stream will be reduced as much as is consistent with good practice.

A grille used for a long air throw should have a large free area, and should have a number of fins, bars, or tubes with sufficient depth so that the width of each slot is approximately one-half the depth of the fins or bars. The grille opening should be as near square as possible and should be expanded only slightly between the riser or duct and opening. When using grilles with a large free area, if the supply air makes an abrupt turn before being introduced into the occupied space, turning blades (Fig. 2) may be used to equalize the velocity and volume over the entire grille area and assist in controlling the direction of the air stream.

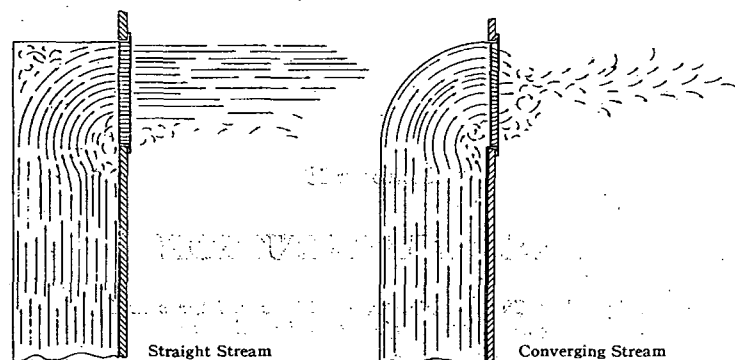


FIG. 1. HIGH VELOCITY LONG THROW INLET OPENINGS

For a short air throw, and when the opposing wall is a short distance from the inlet opening, the average velocity should be high, with the periphery of the air stream increased as much as possible to increase the scrubbing action between the incoming air stream and the stationary air. This causes turbulence or air mixing and readily reduces the vertical rise or fall of the air by equalizing the temperature differential. This method of air introduction may be accomplished by several satisfactory methods. It allows the use of a grille larger than the duct, even though the grille has a large open or free area, providing the dimensional ratio (Fig. 3) is high enough to increase the periphery of the air stream. With the same general type of grille this same result is obtained by expanding the air stream (Fig. 3) as it leaves the opening. The oldest method for reducing the projection of the air stream is by the use of stream turbulence. Introducing heated air with a simple inlet opening located in or near the floor is satisfactory for low air velocities and volumes, but for high air velocities and volumes larger openings are desirable. Cooled air should be intro-

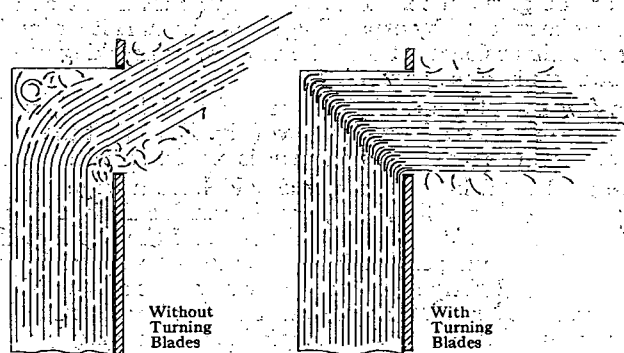


FIG. 2. EFFECT OF TURNING BLADES

duced as far above the floor line as possible, allowing 12 in. or more between the top of grille and the ceiling for air diffusion at this point so that the incoming air will become tempered before dropping into the occupied zone. The supply air should not be directed against the ceiling or wall except as the restrictions of the problem limit the proper applica-

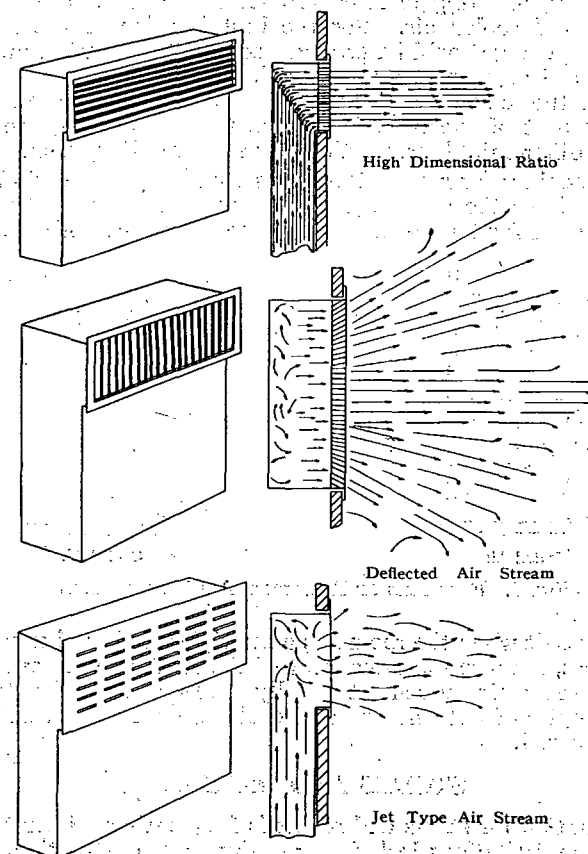


FIG. 3. PERSPECTIVES AND CROSS SECTIONS OF HIGH VELOCITY SHORT THROW INLET OPENINGS

tion, because even though the striking of the air breaks the stream and assists in reducing the effect of temperature differential, it eventually spots and discolors the ceiling or wall area.

RESIDENCES

With gravity warm air systems the inlet openings are usually located in the floor or base board of each room and the exhaust openings in the floor near the exposed walls. (See Chapter 24).

With a mechanical warm air system the inlet openings may be located in the floor or base board with satisfactory results but may also be located above the breathing line, which is equally satisfactory when proper inlet velocities are used. (See Chapter 23). Wall openings located above the breathing line permit higher inlet velocities with reduced sizes of the openings, resulting in improved air distribution satisfactory for both heated and cooled air. For the best location of the high inlet opening, the opening should be arranged so that the air will flow parallel and in close proximity to an exposed wall. If a low inlet opening is used for cooled air the temperature differential causes the cooled air to settle in the lower part of the occupied space, thereby causing too great a temperature differential between the floor and breathing level.

Exhaust openings for mechanical warm air systems located in or near the floor on either the exposed or interior wall function satisfactorily. The wall selected for the location of this opening does not appear to be

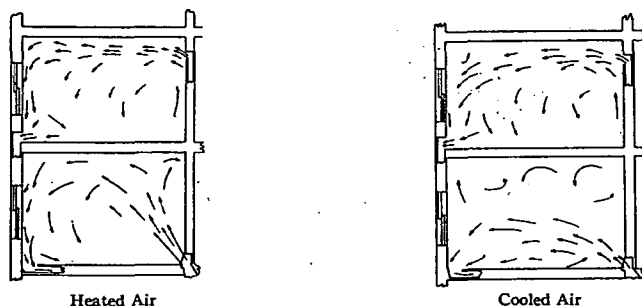


FIG. 4. EXAMPLES OF RESIDENCE AIR DISTRIBUTION

important, except when the room has a multiple number of exposures, or excessive glass, because the movement of the air above the opening and the vertical room air movement usually exceeds the horizontal movement. Examples of both heated and cooled air residential distribution are given in Fig. 4.

STORES AND OFFICES

Air distribution for stores and offices is frequently a difficult problem because of the limitations in building design and construction. With a duct distribution system it is advisable to locate the inlet opening in the side wall as high as possible so that the location will be applicable for cooling as well as for heating.

When an inlet opening is located near the ceiling on one of the walls, the velocity through the opening should be as high as possible so as to project the air stream. The air stream may be expanded by deflection for short throws in offices and across the width of narrow stores. If the length of the throw required should exceed 40 ft as in the case of inlet openings installed at one or both ends of a store, a long throw opening (Fig. 1) should be used with a multiple number of openings proportionately spaced in either or both ends. Since every grille has a different charac-

teristic, it is advisable to use reliable grille engineering information for the proper selection. See *Catalog Data Section*. It is generally impossible to supply too many inlet or exhaust openings for any of these conditions.

One exhaust opening should be supplied for each office whenever possible, using a long narrow opening so that the air will be drawn from as large an area as possible, although the location can be either under or opposite the supply openings.

Suspended unit heaters and coolers are occasionally used in stores and offices and where a single unit is used, care should be exercised to insure that the air flow will diffuse properly. Where a multiple number of units are used it is advisable to locate the units so that the air streams all

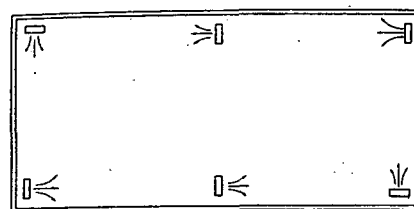


FIG. 5. INSTALLATION ARRANGEMENT OF MULTIPLE SUSPENDED UNIT HEATERS

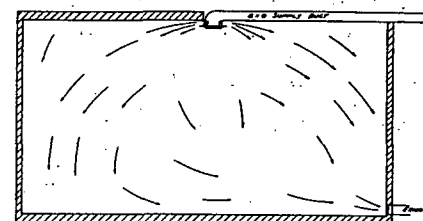


FIG. 6. AIR DISTRIBUTION WITH CEILING INLET OPENING

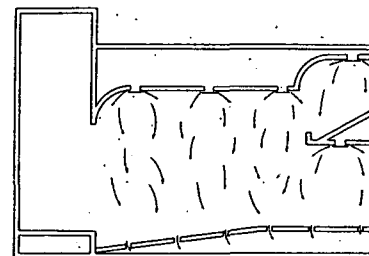


FIG. 7. SECTION THROUGH THEATRE SHOWING AIR DISTRIBUTION WITH COOLED AIR

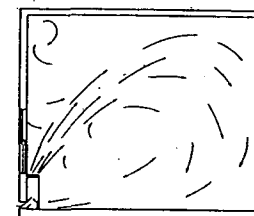


FIG. 8. SECTION THROUGH ROOM SHOWING AIR DISTRIBUTION WITH UNIT VENTILATOR

circulate in the same direction (Fig. 5) or in such a manner as to distribute the air uniformly throughout the space.

Another method of air introduction is a ceiling outlet (Fig. 6) with the air discharged downward on a deflector which spreads the air in all directions at right angles to the opening. With such an opening, a foot square and properly designed with a 4 in. opening completely around the outlet, the air may be discharged in an expanding circle with a low velocity for large and small volumes.

AUDITORIUMS

This type of building often presents the greatest problem in air distribution, mainly because winter conditions require air cooling, and for summer comfort air conditioning large volumes of air are required.

For all such buildings the downward system (Fig. 7) offers the best distribution without draft if a sufficient number of exhaust openings are placed under the seats. This method discharges the cooled air so that it filters through and mixes with the layer of heated air rising from the occupants.

The upward supply system allows the cooled air to remain partially undisturbed in the lower area and thus the heated air from occupants is not distributed until it rises near the exhaust openings.

Small theatres and similar buildings may be properly designed by distributing the air across the width of the building. The inlet opening locations should be as high as possible, with high velocities and multiple return openings in the floor under the seats, for best results.

Schools are often heated and cooled by unit ventilators. These units should be designed for high velocity and a deflection of approximately 40 deg from the vertical (Fig. 8), so that the air stream will not strike the ceiling and fall too rapidly, particularly when a large quantity of air is admitted from outdoors.

RAILROAD CARS

Although the general variation in the size of a railroad car is slight the construction often does not allow the best arrangement for air distribution, and consequently several alternate designs have been used. (See Chapter 13).

In early practice most cars were air conditioned with the bulkhead delivery system. This consisted of an inlet opening at each end of the car discharging the air towards the middle with the flow parallel to the long dimension of the car. This installation resulted in drafts in the middle of the car and was frequently unsatisfactory.

The side duct delivery system (Fig. 9) consists of a duct on each side of the car roof directing the air through about 12 inlet openings toward the middle of the car, where the two streams meet and deflect downward gradually filtering into the aisle. The inlet openings generally have a dimensional ratio of 3 and 4 to 1 with deflection to expand the air stream.

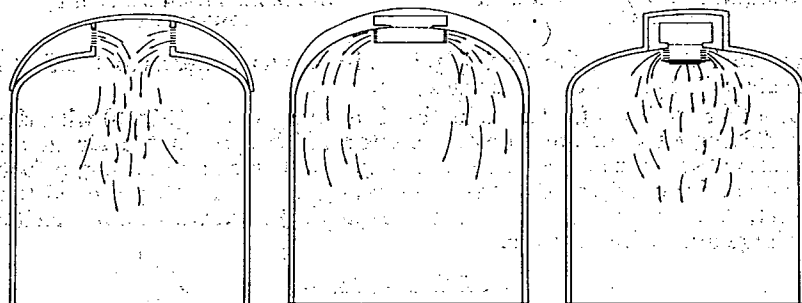


FIG. 9. SECTIONS THROUGH RAILROAD CAR AIR SUPPLY SYSTEMS

This system is highly satisfactory if the volume is properly proportioned for the multiple openings.

Several center duct air distribution systems are used in connection with railroad car air distribution. One method consists of a continuous slot in the bottom of the duct to which is attached a flat plate so the air is deflected along the car ceiling. This is satisfactory from an air distribution stand point, but it is difficult to equalize the volume of air over the entire length of this slot.

Two other center duct installations are shown in Fig. 9. One shows the air distributed from the side of the duct through about 12 inlet openings which have a high dimensional ratio and an expanding deflection. This method of distribution causes the air to deflect downward on the outside wall of the car with the result that many passengers raise objections to the drafts. The other method, shown in Fig. 9, is to use a standard square or circular ceiling outlet in the bottom of the duct which allows the inlet air to be discharged on a deflector plate, which distributes the air along the ceiling in a widening circle at right angles to the inlet opening. Since the velocity is very low, most of the air drops into the aisle of the car.

Recirculated air from a railroad car usually consists of from 60 to 75 per cent of the air supplied and is picked up in the passageway ceiling of the car at one end. This results in all of the air being drawn across the floor of the car toward the exhaust opening. This opening should be large enough so that the velocity is not in excess of 400 fpm, and arranged so as to eliminate objectional drafts near this opening.

PROBLEMS IN PRACTICE

1 • What are the essential differences between a high velocity long throw and short throw inlet grille opening?

Generally a high velocity long throw grille is used where a large compact mass of air is projected with a reduction in the periphery of the air stream, whereas with a short throw grille design the periphery of the air stream is expanded as much as possible to increase the scrubbing action between the incoming air stream and the stationary air.

2 • Is the conventional warm air system, employing floor or baseboard supply registers, suitable for heating and cooling?

Floor or baseboard supply registers are suitable for heating service because the natural tendency of warm air is to rise. They are not suitable for cooling because the natural tendency of cool air is to stay near the floor and gradually work its way to the return registers, thus not cooling the air in the upper part of the room. See Fig. 4.

3 • What type of air distribution system is suitable for heating and cooling a home?

In order to provide satisfactory cooling without drafts it is necessary to discharge the air at relatively high velocity toward the ceiling from a high point, as shown in Fig. 4. If the register is properly designed and the air capacity is limited to approximately 400 cfm, the cool air will mix with the air in the room before it drops to the occupied zone. However, care must be taken that discharged air does not impinge on beams which would cause the cool air to be deflected downward. This arrangement is also satisfactory for heating.

4 • Why is the conventional low velocity side wall inlet unsatisfactory for cooling purposes?

With the conventional side wall inlet using velocities of 300 to 400 fpm the discharged air quickly loses its velocity and drops, causing drafts in the occupied zone.

5 • What method of side wall introduction is satisfactory for cooling purposes with a 12-ft ceiling height?

The methods shown in Figs. 1 and 3 can satisfactorily circulate air as much as 10 to 15 F below room temperature, provided (1) each jet is limited to 400 cfm, (2) the outlet velocity is high, (3) the air is directed toward but not striking the ceiling, and (4) there are no beams on the ceiling. In order to employ this method in a classroom it is usually necessary to have at least three inlets, but even with three inlets the cooling capacity is limited to that obtained by circulating air at 10 to 15 F below room temperature.

6 • Should unit ventilators be considered as heating units or as cooling units?

Experience has shown that approximately 75 per cent of the time a classroom is occupied the problem is one of cooling rather than one of heating. For this reason unit ventilators should be considered as cooling units.

7 • What method of air distribution is usually employed with unit ventilators?

Most unit ventilators employ a unique method of air distribution in which the air is discharged at a high velocity toward the ceiling. The air stream is usually inclined toward the room.

8 • How should a unit ventilator be located in a room that has ceiling beams?

When there are ceiling beams the unit ventilator should be so located that the beams will be parallel with the direction of the air discharge in order that the beams will not deflect the air downward.

9 • What is the minimum temperature at which unit ventilators can distribute air in a classroom without causing drafts?

Some manufacturers recommend a minimum outlet temperature of 60 F. Others recommend no limitation of the outlet temperature, providing the unit is properly located and installed.

10 • What is the usual method of ventilating school auditoriums and gymnasiums when unit ventilators are used in the classrooms?

If unit ventilators are used in classrooms the usual method of ventilating the auditorium or gymnasium is to use one or more large units located above and on either side of the stage.

11 • What is the maximum amount of air which should be discharged from one point in a school auditorium or gymnasium?

The maximum amount of air which should be discharged from one point is 5000 cfm. This limitation applies whether the air is supplied by units or by a central fan from a distant point.

12 • Are vents required in school classrooms, auditoriums, and gymnasiums?

Many states require vents in school classrooms, auditoriums and gymnasiums. Most engineers recommend them for central fan systems. Many engineers recommend the elimination of all vents when unit ventilators are used. If vents are installed they should be provided with dampers in order that they may be throttled and with back draft dampers to prevent cool air from entering the building at night and during holidays.

13 • What type of system is generally used in large continuously operated theaters?

Most large continuously operated theaters are provided with complete downward systems of air distribution similar to the one shown in Fig. 7. With this system a large number of inlet openings is provided, each of which discharges air in a thin horizontal stream at high velocity in order that the cool air will be mixed with the air in the theater before it reaches the patrons.

Chapter 20

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Friction Loss Chart, Proportioning the Losses, Sizes of Ducts, General Rules, Procedure for Duct Design, Air Velocities, Proportioning the Size for Friction, Main Trunk Ducts with Branches for Public Buildings, Equal Friction Method, Details of Duct Construction

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.

In the development of the formulae, diagrams, and tables for the flow of air, use is made of the following basic equation for the flow of fluids:

If H_v be the velocity head in feet of a fluid, and the velocity, V , be expressed in feet per minute, the fundamental equation is

$$V = 60 \sqrt{2g H_v}$$

The factor g is the acceleration due to gravity, or 32.16 ft per second per second.

It is usual to express the head in inches of water for ventilating work and, since the heads are inversely proportional to the densities of the fluids,

$$\frac{H_v}{h_v} = \frac{62.4}{d}$$

or

$$H_v = 5.2 \frac{h_v}{d}$$

therefore,

$$V = 1096.5 \sqrt{\frac{h_v}{d}} \quad (1)$$

where

V = velocity in feet per minute.

h_v = velocity head or pressure in inches of water.

d = weight of air in pounds per cubic foot.

For standard air (70 F and 29.92 in. barometer) $d = 0.07495$ lb per cubic foot. Substituting this value in Equation 1:

$$V = 1096.5 \sqrt{\frac{h_v}{0.07495}} = 4005 \sqrt{h_v} \quad (2)$$

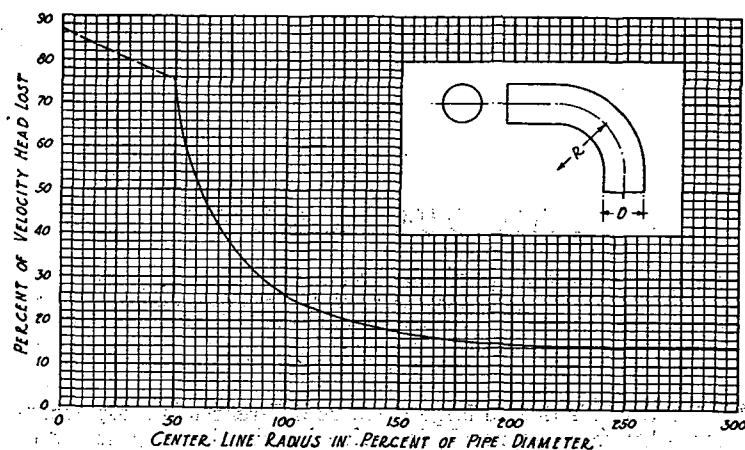


FIG. 1. CURVE SHOWING LOSS OF PRESSURE IN ROUND ELBOWS

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses are those due to the friction of the air against the sides of the duct. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Dynamic Losses

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

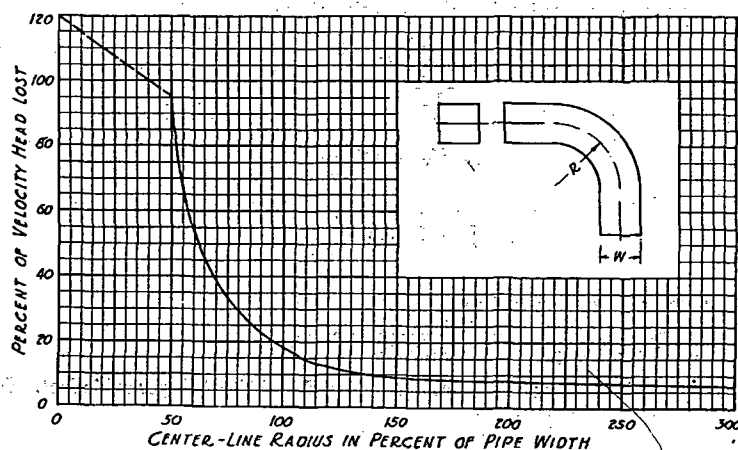


FIG. 2. CURVE SHOWING LOSS OF PRESSURE IN SQUARE ELBOWS

0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design. It is customary to express dynamic losses in terms of the percentage of the velocity head; in other words, the percentage of that pressure corresponding to the average velocity in the duct which is expressed in terms of inches of water gage. Figs. 1 and 2 show the effect of changing the radius of elbows of square and rectangular section. These charts are based on tests of pipe elbows of ordinary good sheet metal construction. For example, a five-piece round pipe elbow having a centerline radius of one diameter has a loss of about 25 per cent of the velocity head. At a velocity of 2000 fpm the corresponding head is 0.25 in. water gage, and at this velocity the elbow just referred to would cause a pressure drop of 0.063 in. water gage. Experience has shown that good results may be obtained when the radius to the center of the elbow is $1\frac{1}{2}$ times the pipe diameter. The pressure drop will then be approximately 17 per cent of the velocity head for round ducts, and 9 per cent for square ducts. Very little advantage is gained in making elbows with a radius of more than two diameters.

Friction Losses

Friction losses vary directly as the length of the duct, directly as the square of the velocity, and inversely as the diameter. Since length is a fixed quantity for any system, the factors subject to modification are the area and the velocity, which determine the relation between the first cost of the duct system and the cost of the power for overcoming friction.

The friction between the moving air and pipe surface causes a loss of head which is numerically equal to the pressure required to maintain a given velocity, and is expressed in the following modification of Fanning's formula:

For round pipe and standard air (70 F and 29.92 in. barometer)

$$h_L = f \frac{L}{D} h_v = \frac{L}{C D} \left(\frac{V}{4005} \right)^2 \quad (3)$$

For rectangular ducts

$$h_L = f L \left(\frac{a+b}{2ab} \right) h_v = \frac{L}{C} \left(\frac{a+b}{2ab} \right) \left(\frac{V}{4005} \right)^2 \quad (4)$$

where

h_L = loss of head, inches of water.

$h_v = \left(\frac{V}{4005} \right)^2$ = velocity head, inches of water.

V = velocity of air, feet per minute.

L = length of pipe

D = diameter of pipe

a, b = sides of rectangular duct

f = coefficient of friction.

$C = \frac{1}{f}$ = length of pipe in diameters for one head loss.

For all practical purposes C varies only with the nature of the pipe surface: $C = 60$ for perfectly smooth pipe; $= 55$ for pipe as used in planning mill exhaust systems; $= 50$ for heating and ventilating ducts; $= 45$ for

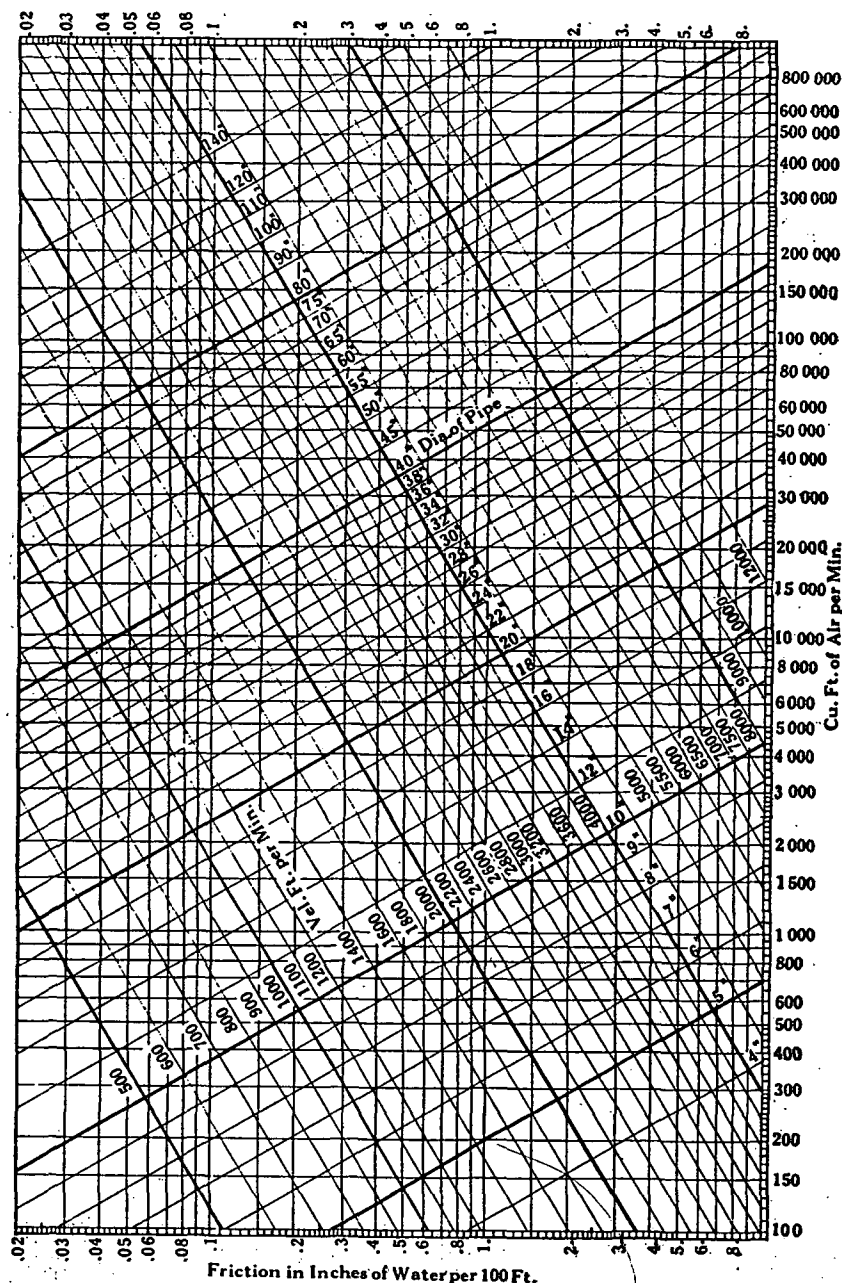


FIG. 3. FRICTION OF AIR IN PIPES

smooth and 40 for rough conduits of tile, brick or concrete. However, Fritzsche states (and numerous tests check very closely) that f varies inversely as the $2/7$ power of the pipe diameter, and inversely as the $1/7$ power of the velocity, or inversely as the $1/7$ power of capacity, which is the same thing. Thus Formula 3 may be revised as follows, based upon a loss of one velocity head (at 2000 fpm) in a length equal to 50 diameters of 24-in. galvanized swaged pipe:

$$h_L = 1.1 \frac{L}{CD^{5/7}} \left(\frac{V}{4005} \right)^{13/7} \quad (5)$$

The preceding formulae are based on standard air, and for other conditions the friction varies directly as the air density and inversely (approximately) as the absolute temperature. The increase of friction due to increase of air viscosity with increased temperature is small and is generally neglected.

Friction Loss Chart

Fig. 3 is a convenient chart for determining the friction loss for various air quantities in ducts of different sizes. The general form of this chart is familiar, but it should be noted that it is corrected for changes in the coefficient of friction based on the rule that the coefficient of friction varies inversely as the $2/7$ power of the diameter, and inversely as the $1/7$ power of the velocity. Fig. 3 is based on a loss of one velocity head (at a velocity of 2000 fpm) in a length equal to 50 diameters of 24-in. round galvanized-iron duct of the usual construction. Although this chart is laid out for a value of C equivalent to 50, it may be used for other values of C by varying the friction inversely as this constant. For example, if a rougher pipe is used with 40 as the value of C , the friction loss

as read from the chart should be multiplied by $\frac{50}{40}$.

Example 1. Assume that it is desired to pass 10,000 cfm of air through 75 ft of 24-in. diameter pipe. Find 10,000 cfm on the right scale of Fig. 3 and move horizontally left 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 0.59 in.; then for 75 ft the friction will be $0.75 \times 0.59 = 0.44$ in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Proportioning the Losses

Other losses of pressure occur at the entrance to the duct, through the heating units, and at the air washer. In ordinary practice in ventilation work it is usual to keep the sum of the duct losses $\frac{1}{3}$ to $\frac{1}{2}$ and the loss through the heating units at less than $\frac{1}{2}$ of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

DUCT SIZES

The sizes of ducts and flues for gravity or mechanical circulation of air are usually based on the losses due to friction, and these losses must be kept within the available pressure difference. This pressure difference in

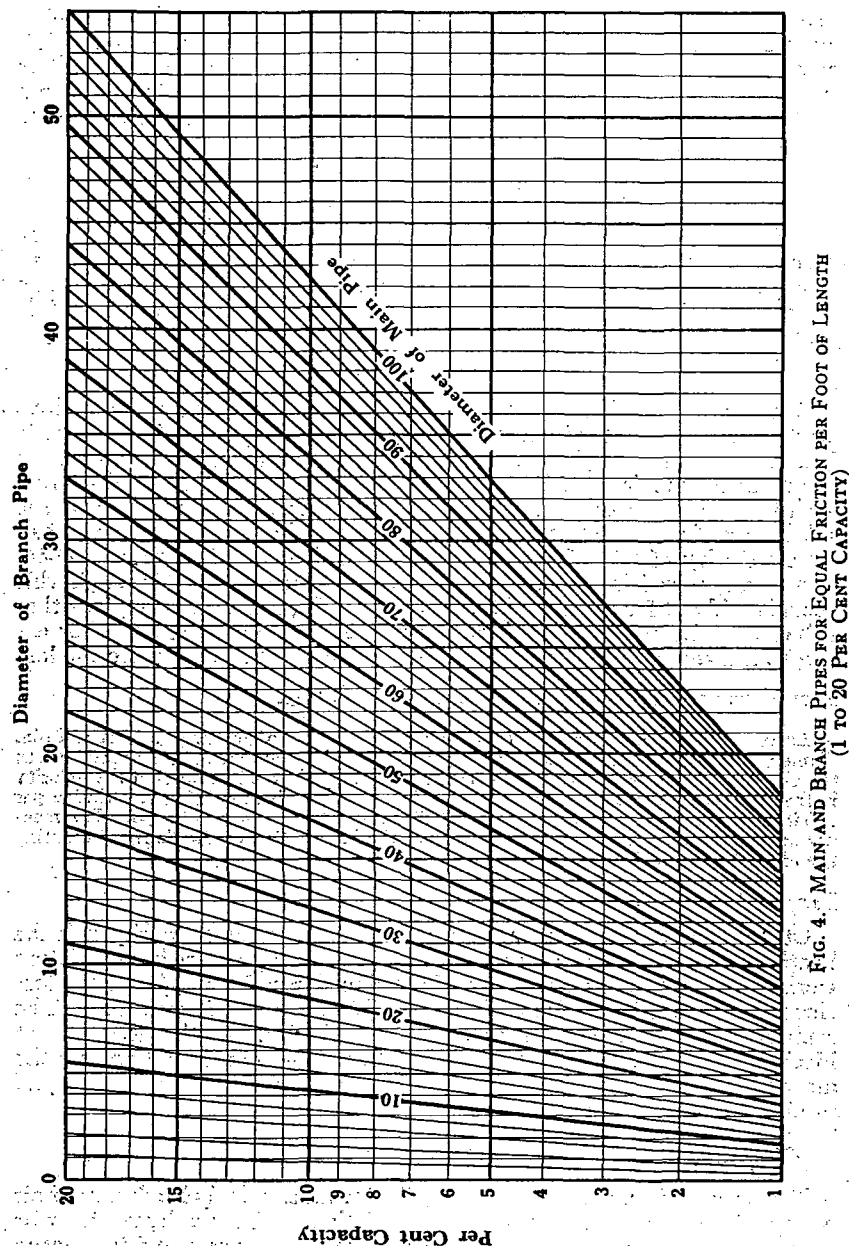


FIG. 4.—MAIN AND BRANCH PIPES FOR EQUAL FRICTION PER FOOT OF LENGTH (1 TO 20 PER CENT CAPACITY)

mechanical ventilation is that derived from the fan, while in gravity ventilation the aspirating effect due to the temperature and height of the column of heated air causes the pressure difference.

General Rules

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.
3. The sides of all ducts or flues should be as nearly equal as possible. (In no case should the ratio between long and short sides be greater than 10 to 1.)

Procedure for Duct Design

The general procedure for designing a duct system is as follows:

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 heat.
3. Divide the building into zones and proportion the volume of air necessary to supply the heat 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.
5. Calculate the sizes of all main and branch ducts by either of the following two methods:
 - a. *Velocity Method.* Arbitrarily fix 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.
 - 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 in 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 following velocities of air are considered standard for public buildings:

1. Through the outside air intakes, 1000 fpm.
2. Through connections to and from heating unit, 1000 to 1200 fpm.
3. Through the main discharge duct, from 1200 to 1600 fpm.
4. In branch ducts, 600 to 1000, and in vertical flues, 400 to 800 fpm.
5. In registers or grilles, 200 to 400 fpm depending upon the size and location. If diffusers of proper design are used, 25 per cent higher air velocities are permissible.

These duct velocities may safely be increased 20 per cent if first-class construction is used to prevent any breathing, buckling, or vibration. High velocities at one point in the system neutralize the effect of proper design at all other points; hence the importance of splitters in elbows and similar precautions. For industrial buildings noise is seldom considered, and main duct velocities as high as 2800 or 3000 fpm may be used where conditions will permit. For department stores and similar buildings,

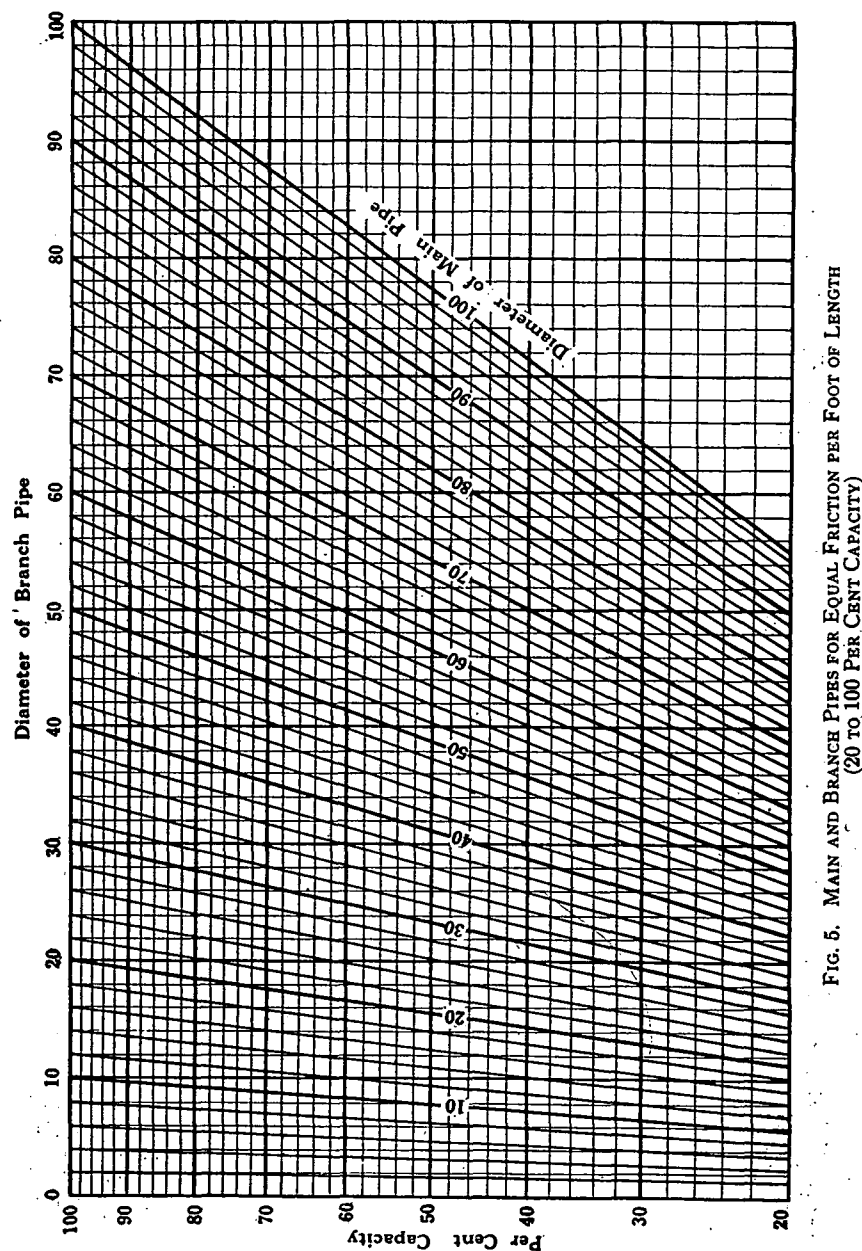


FIG. 5. MAIN AND BRANCH PIPES FOR EQUAL FRICTION PER FOOT OF LENGTH (20 TO 100 PER CENT CAPACITY)

maximum velocities with good construction and design may be as high as 2000 or 2200 fpm in main ducts, with suitable reduction in branches and outlets. With these velocities first-class duct construction is essential.

Proportioning the Size for Friction

By means of Figs. 4 and 5 the diameter of branch pipes necessary to carry a given percentage of the total air in the main pipe and to maintain equal friction per foot of the length through the entire system may be determined. These charts, as well as Fig. 3, are based on the assumption that the coefficient of friction varies inversely as the $1/7$ power of the capacity.

Example 2. Suppose a 60-in. main pipe is to be used, and it is desired to know the size of branch pipe required to carry 50 per cent of the total air in the main. Find 50 per cent at the left of the chart, move right to the 60-in. diagonal line and note directly above at the top of the chart that the branch pipe will be 46.5 in. in diameter.

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have the same friction per foot of length. Table 1 gives directly the circular equivalent of rectangular ducts for equal friction and capacity. To obtain the size of rectangular ducts for different capacities, but of the same friction per foot of length, first obtain the equivalent round pipe for equal friction. Thus, if a branch of sufficient size to carry 30 per cent of a 12 x 36-in. pipe is desired, it is found from Table 1 that the main is equivalent to a 22.2-in. diameter round pipe. From Fig. 5, 30 per cent of this is a pipe 14.3 in. in diameter, and referring again to Table 1, the rectangular equivalent branch is a 12 x 14-in., 10 x 17 $\frac{1}{4}$ -in., or any other desirable combination.

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 just twice that of a 40 x 12-in. duct, or $2 \times 23.3 = 46.6$ in.

DUCTS FOR PUBLIC BUILDINGS

A main duct with branches is generally used to convey tempered air for ventilation purposes only. In place of individual ducts, a comparatively large main duct supplies air by branches to the room or rooms. The velocities vary according to the nature of the installation and the degree of quietness required. At the start of the run a velocity as high as 2000 fpm may be used, but this is considered the maximum for public building work, and is reduced to from 400 to 800 fpm in the risers. This duct system may be designed so that the loss of pressure in the branches is equalized in a manner similar to that previously described.

Equal Friction Method

Example 3. 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 7 $\frac{1}{2}$ -minute air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The clear area of the fresh air inlet is based on a velocity of 1000 fpm or $\frac{22,935}{1000} =$

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION*

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	17.8	18.4	19.0	19.7	20.3	20.9	21.5	22.0	22.5	23.0	23.5
21	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.8	20.5	21.1	21.7	22.3	22.9	23.4	23.9	24.4
22	10.0	11.3	12.4	13.5	14.5	15.5	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.7	24.4	25.1	25.8	26.5
24	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	23.7	24.4	25.1	25.8	26.5	27.2
26																				
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.2	23.1	23.9	24.6	25.3	26.0	26.8	27.5	28.2
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	27.0	27.7	28.4	29.1
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	27.0	27.9	28.8	29.6	30.4	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	33.6	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	21.2	22.6	24.0	25.3	26.6	27.8	28.9	30.0	31.1	32.2	33.2	34.1	35.0	35.9	36.8	38.9
56	14.3	16.4	18.2	20.0	21.7	23.1	24.5	25.8	27.1	28.3	29.4	30.5	31.6	32.7	33.7	34.6	35.5	36.4	37.4	39.5
58	14.6	16.8	18.6	20.4	22.1	23.5	24.9	26.2	27.5	28.7	29.8	30.9	32.0	33.1	34.1	35.0	35.9	36.8	37.8	40.0
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	35.9	36.8	37.7	38.7	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	36.5	37.5	38.5	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

*Additional sizes: $4 \times 5 = 4.9; 4 \times 6 = 5.4; 4 \times 7 = 5.8; 5 \times 5 = 5.5; 5 \times 6 = 6.3; 5 \times 7 = 6.5$.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Continued)

Side Rectangular Duct	26	28	30	32	34	36	38	40	42	44	46	48	Side Rectangular Duct	50	54	60	66	72	78	84	88
26	28.6												50	55.0							
28	29.7	30.8											52	56.1							
30	30.7	31.9	33.0										54	57.2	59.4						
32	31.7	32.9	34.1	35.2									56	58.3	60.5						
34	32.7	33.9	35.1	36.3	37.4								58	59.3	61.6						
36	33.7	34.9	36.1	37.3	38.5	39.6							60	60.3	62.7	66.0					
38	34.6	35.9	37.1	38.4	39.5	40.7	41.8						62	61.3	63.7	67.1					
40	35.3	36.7	38.0	39.3	40.5	41.7	42.9	44.0					64	62.2	64.7	68.2					
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2				66	63.2	65.7	69.3	72.6				
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4			68	64.1	66.6	70.3	73.7				
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6		70	65.0	67.6	71.3	74.8				
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	72	65.9	68.5	72.3	75.9	79.2			
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	74	66.8	69.4	73.3	76.9	80.3			
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	76	67.6	70.3	74.2	77.9	81.4			
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	78	68.4	71.2	75.2	78.9	82.5	85.8		
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	80	69.2	72.1	76.1	79.9	83.6	86.9		
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	82	70.1	73.0	77.1	80.9	84.6	88.0	92.4	
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	84	70.9	73.8	78.0	81.9	85.6	89.1		
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	86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
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	88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
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	90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
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	92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
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	94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
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	96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

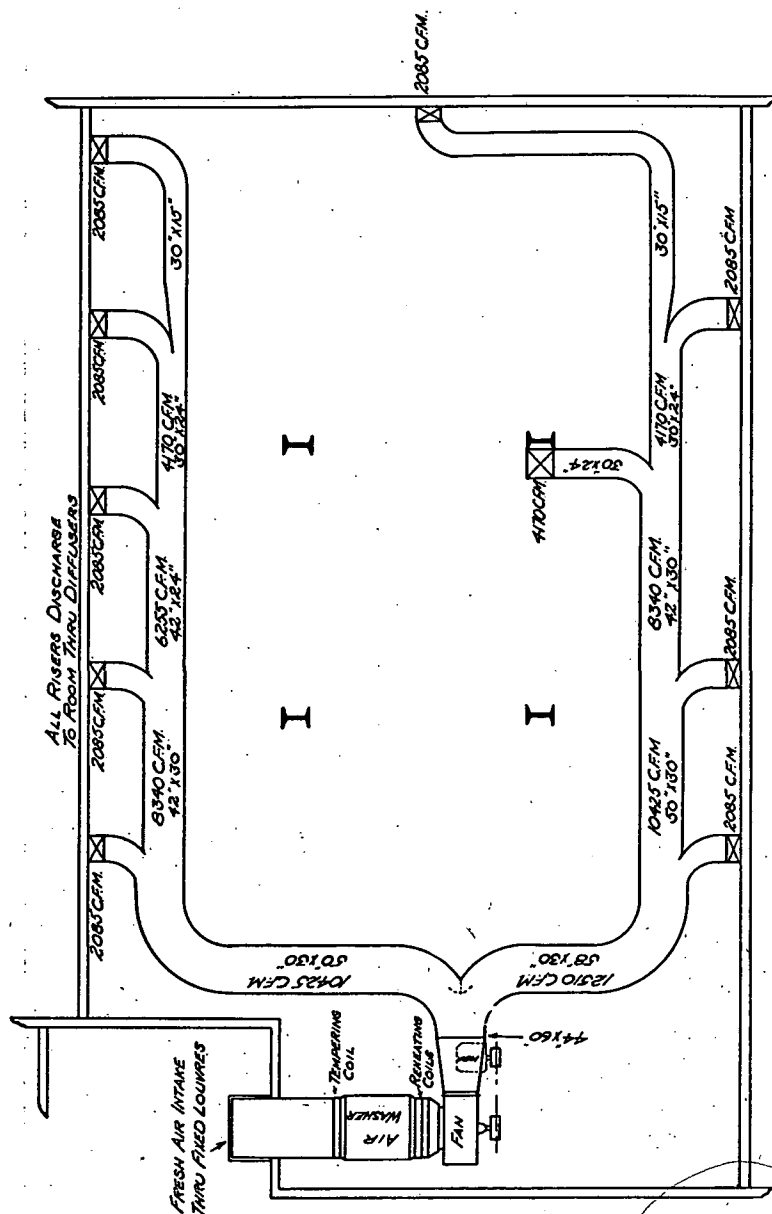


FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

22.94 sq ft. If the air washer is provided with automatic humidity control, the tempering coil should raise the temperature of the entering air to 32 F. The washer with its automatic control will then raise the temperature from 32 F to 42 F. If the washer is not provided with automatic humidity control, the tempering coil must raise the temperature of the entering air to at least 55 F to allow for some temperature drop in the washer due to evaporation. The reheating coil is selected to raise the temperature of the air from that leaving the air washer to 70 F. The air washer should have a maximum velocity of 500 fpm through the clear area, which, in this case, is 46 sq ft. For more detailed information on tempering coil and air washer control, see Chapters 14 and 23.

Since the plan shows a moderately short run of main duct with no risers near the fan outlet, a fan should be selected which will have the required capacity of 22,935 cfm with a maximum velocity through the fan outlet of 1400 fpm. The outlet area, therefore, should be $16\frac{1}{2}$ sq ft.

TABLE 2. PIPE SIZES FOR EXAMPLE 3^a

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
22,935	100.0	56	60 x 44
12,510	54.6	45	58 x 30
10,425	45.4	42	50 x 30
8,340	36.3	39	42 x 30
6,255	27.2	35	42 x 24
4,170	18.2	29½	30 x 24
2,085	9.1	23	30 x 15

^aVelocity through diffusers (not shown) to be approximately 300 fpm.

The main pipe size should be selected to give a velocity equal to or less than the velocity at the fan outlet. Choosing a 56-in. pipe with a cross-sectional area of 17.1 sq ft, the velocity in the main pipe will be 1340 fpm. Using the friction pressure loss method this 56-in. main pipe will be taken as the basis of calculation.

Fig. 6 shows the amount of air to be handled by each section of pipe. Expressing the volume handled by each section as a percentage of the total volume and using the charts, Figs. 4 and 5, the pipe sizes are as shown in Table 2.

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 of each riser 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 that shown in Fig. 6 where the main duct divides into the 58 in. x 30 in. and 50 in. x 30 in. branches.

The rectangular equivalents are selected from Table 1; the width to depth proportion will be determined by construction requirements and ease of fabrication. The calculation of the friction is as follows:

The longest run from the fan outlet to diffuser is 150 ft 0 in.; 150 ft of 56-in. pipe is equivalent to $\frac{150 \times 12}{56} = 32.2$ dia.

Two 45-in., 90-deg elbows ($2 \times \frac{45}{56} \times 8.5$) 13.7 dia.

(Assume each elbow equivalent to 8.5 diameters of duct, Fig. 1.)

Two 23-in., 90-deg elbows ($2 \times \frac{23}{56} \times 8.5$) 7.0 dia.

Two 23-in., 90-deg elbows in riser ($2 \times \frac{23}{56} \times 30$) 24.7 dia.

(Two bad elbows in riser, each equivalent to 30 diameters of duct.)

Total diameter of 56-in. pipe 77.6

The velocity head corresponding to a velocity of 1340 fpm is $\left(\frac{1340}{4005}\right)^2 = 0.112$ in.

Taking 50 diameters as one head loss, then $\frac{77.6}{50} \times 0.112 = 0.174$ in. static loss in duct.

Where the connection pieces are made with long easy slopes and the general workmanship is good, a regain in static pressure may be deducted from the foregoing pressure loss. This can be taken as approximately two-thirds the difference in velocity pressures at the fan outlet and the last run of pipe. The velocity in the riser is 667 fpm with a corresponding velocity pressure of 0.027 in. The fan outlet velocity is 1400 fpm with a corresponding velocity pressure of 0.122 in. The regain equals $\frac{2}{3} (0.122 - 0.027) = 0.063$ in.

The net static pressure loss in the duct only is then:

0.174 in. — 0.063 in. 0.111 in.

Other friction losses are as follows:

- (1) Fresh air intake 1000-fpm velocity ($1\frac{1}{2}$ heads $\times 0.0625$) 0.094 in.
- (2) Tempering coil loss (from manufacturer's tables) 0.100 in.
- (3) Air washer loss (from manufacturer's tables) 0.250 in.
- (4) Reheating coil loss (from manufacturer's tables) 0.100 in.
- (5) Allowance for regulating dampers and diffusers 0.100 in.

Static pressure loss of system 0.755 in.

The fan should be selected from the manufacturer's ratings which, according to the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers¹, will deliver 22,935 cfm at a static pressure of 0.755 in. and which has an outlet area of $16\frac{1}{2}$ sq ft.

The method of design used in Example 3 is the *equal friction method* described under the heading Procedure for Duct Design. This involves the arbitrary reduction of velocity from the fan outlet to the point of discharge to the room, and the friction is calculated by adding the pressure losses of each section of duct. This method requires dampering in the risers.

Example 4. Fig. 7 shows an exhaust system layout for exhausting from buildings of the same type as in Example 3. Assume the air requirements based on the number of air changes per hour to be 16,800 cfm. Using a velocity of 1400 fpm in the main duct at

TABLE 3. PIPE SIZES FOR EXAMPLE 4^a

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
16,800	100.0	47	38 x 48
11,550	68.8	41	30 x 46
9,450	56.2	38	30 x 40
5,250	31.3	31	24 x 34
4,200	25.0	28.5	24 x 28
3,150	18.8	25.3	16 x 34
2,100	12.5	21.6	16 x 24

^aVelocity through intake grilles (not shown) to be approximately 400 fpm.

¹See Chapters 17 and 44.

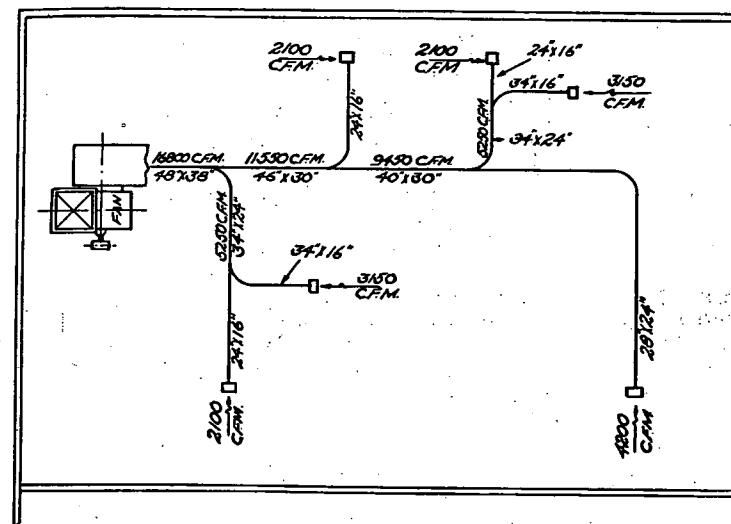


FIG. 7. EXHAUST SYSTEM LAYOUT

the fan inlet, which is an average velocity for this type of system, the area of the main is 12 sq ft, which corresponds to a 47-in. pipe.² Referring to Example 3, and using the charts, Figs. 4 and 5, the pipe sizes are as indicated in Table 3.

All risers will require dampering as in Example 3. The calculation of the friction is as follows:

The longest run from the intake grille to fan inlet is 100 ft.

- (1) Duct friction 100 ft of 47-in. pipe $\left(\frac{100 \times 12}{47}\right)$ 25.6 dia.

- Two 28½-in., 90-deg elbows in riser $\left(\frac{2 \times 28.5 \times 30}{47}\right)$ 36.4 dia.

(Two bad elbows in riser each equivalent to 30 diameters of duct).

- One 28½-in., 90-deg elbow in horizontal run $\left(\frac{28.5 \times 8.5}{47}\right)$ 5.2 dia.

Total diameter of 47-in. pipe 67.2 dia.

Velocity head corresponding to 1400 fpm is $\left(\frac{1400}{4005}\right)^2 = 0.122$ in.

Taking 50 diameters as one head loss, then $\frac{67.2 \times 0.122}{50}$ 0.164 in.

- (2) Intake loss from grille ($1\frac{1}{2}$ heads at a 400 fpm velocity $1\frac{1}{2} \times 0.01$) 0.015 in.
- (3) Static pressure required to produce one velocity head at 1400 fpm 0.122 in.
- (4) Loss occasioned by step-up of velocity (0.20×0.122) 0.024 in.

(This loss varies from 0.05 to 0.40 velocity head depending upon the nature of the change. For average systems 0.20 velocity head is a close approximation.)

Static pressure loss on inlet side 0.325 in.

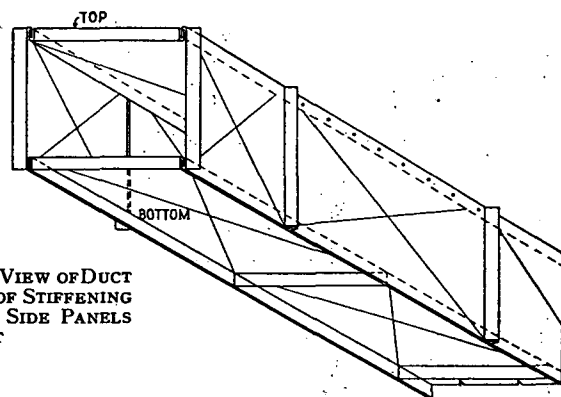


FIG. 8. ISOMETRIC VIEW OF DUCT SHOWING LOCATION OF STIFFENING SEAMS ON TOP AND SIDE PANELS OF DUCT

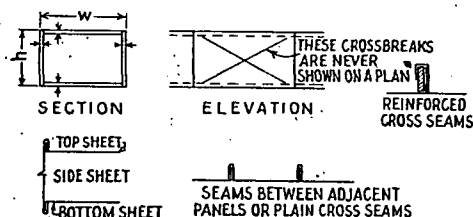


FIG. 9. DETAILS OF SEAMS

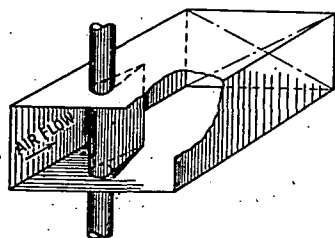


FIG. 11. INSTALLATION OF EASEMENT IN DUCT AROUND OBSTRUCTION



FIG. 10. METHOD OF INSTALLING HEATING UNIT

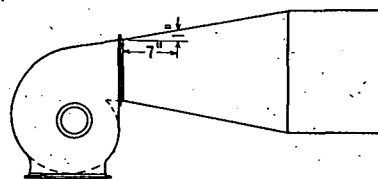


FIG. 12. FAN DISCHARGE CONNECTION

CHAPTER 20—AIR DUCT DESIGN

To this must be added the resistance on the discharge side of the fan. A fan outlet velocity of approximately 1500 to 1600 fpm may be used. Assuming the fan outlet to be equivalent in area to a 45-in. pipe, the velocity is 1525 fpm.

Loss on discharge (15 ft from fan outlet to discharge):

$$\frac{15 \times 12}{45} = 4 \text{ diameters of 45-in. pipe.}$$

The velocity head corresponding to a velocity of 1525 fpm is 0.145 and the discharge-side loss is $\frac{0.145 \times 4}{50} = 0.012$ in. The total static pressure loss of the system is then:

$$0.012 + 0.325 = 0.337 \text{ in.}$$

The fan will be selected to handle 16,800 cfm at a static pressure of 0.337 in. and to have an outlet velocity of 1525 fpm. Outlet area 11 sq ft.

Where there are one or more ducts with branches, the velocity of air in the ducts may be either chosen arbitrarily or calculated for friction losses. When arbitrary values are assigned, a certain amount of dampering should be provided for; this will be small when the method chosen permits a drop in velocity as the quantity of air is reduced.

After the total air quantity and the size of fan are ascertained, the main duct is usually fixed as being at least equal in area to the fan outlet, or perhaps 10 per cent greater. From this main pipe all others are proportioned. For example, if the main duct is 30 in. in diameter, a branch to carry 10 per cent of the total capacity should be 12.7 in. in diameter (see Fig. 4) in order to have the same friction per foot of length, while one carrying one-half the total capacity of a 30-in. main with the same friction loss per foot would be 23.4 in. in diameter. By this method of equalizing friction it is unnecessary to consider the resistance of each section of pipe independently, but only to know the distance from the fan outlet to the end of the longest run of pipe, the number and size of elbows, and the diameter and velocity in the largest pipe.

Example 5. If the greatest length of piping in a system is 130 ft with a 26-in. diameter main pipe and one 20-in. elbow, the piping having been designed for equal friction per foot of length, the friction would be the same as for 130 linear feet of 26-in. pipe, or 60 diameters. To this should be added the friction loss in elbows, in this case one 20-in. elbow, which has a loss equivalent to 8.5 diameters of 20-in. pipe. This in turn is $\frac{20}{26} \times 8.5 = 6.6$ diameters of 26-in. pipe. The total equivalent length of the system will then be $60 + 6.6$, or 66.6 diameters. Since 50 diameters is equivalent to one velocity head, the loss is $\frac{66.6}{50} = 1.33$ times the velocity head. If the velocity is, for example, 2200 fpm, corresponding to 0.3-in. pressure, the friction loss of the system will be $1.33 \times 0.3 = 0.399$ in.

Frequently the prevention of sound in a heating or ventilating system imposes more severe restrictions than the prevention of excessive pressure drop. This question is highly involved and requires consideration of many factors. The air velocities to be used will vary with the standard of construction used in the ducts themselves as well as with the nature of the occupancy and the construction of the building. In general, architects and engineers who leave the details of duct construction to the contractor must, of necessity, design for lower velocities than might be required for quiet operation if proper construction details were always followed. The

contractor may be expected to build the ducts by the least expensive methods, and the engineer must anticipate this. For further information on noise reduction, see Chapter 18.

Details of Duct Construction

If panel construction is used with standing seams or similar reinforcement, and the panels are cross-broken to give rigidity, there is less likelihood of vibration due to air flow, or deflection due to air pressure. Elbows made without splitters, and improperly shaped transformation sections produce high local velocities which are the cause of noise in duct work. The use of first-class duct construction with well-designed transformation sections and splitters in elbows tends to maintain relatively uniform velocities with decrease in turbulence and in the noise produced.

TABLE 4. SHEET METAL GAGES FOR RECTANGULAR DUCT CONSTRUCTION^a

GAGE	WIDTH OF DUCT	SEAM	REINFORCED SEAM
26	Up to 12 in.		
24	13 in. to 30 in.	1	
22	31 in. to 48 in.	1	
22	49 in. to 60 in.	1½	⅛ in. x 1⅜ in.
20	61 in. to 90 in.	1½	⅛ in. x 1⅜ in.

^aIf panels are not cross-broken two gages heavier material should be used.

Figs. 8 to 12 show acceptable construction details for rectangular ducts, elbows, and transformation pieces or connections. Other methods are also acceptable, such as the use of angle iron stiffeners for large ducts. Good construction is essential to the elimination of duct noises and for the prevention of a flimsy installation.

Fig. 8 is an isometric view of a duct showing the location of the stiffening seams on the top and side panels. The cross seams should not occur at the same place but should be staggered as indicated. Heating units should be installed as shown in Fig. 10 with the duct connections making an angle of not less than 45 deg, but preferably 60 deg. Fan discharge connections should have a maximum slope of 1 in 7, as indicated in Fig. 12. Whenever a pipe or other obstruction passes through a duct an easement should be placed around the pipe as indicated in Fig. 11. The recommended gages for rectangular sheet metal duct construction are given in Table 4.

REFERENCES

- Fan Engineering*, Buffalo Forge Co.
Heat Power Engineering by Barnard, Ellenwood, and Hirshfeld, Part III.
Mechanical Engineers' Handbook by Lionel S. Marks, McGraw-Hill Book Co.
 The Flow of Liquids, by W. H. McAdams, *Refrigerating Engineering*, February, 1925, p. 279.
 A Study of the Data on the Flow of Fluids in Pipes, by Emory Kemler, *A.S.M.E. Transactions*, Hydraulics Section, August, 31, 1933, p. 7.

PROBLEMS IN PRACTICE

1 ● Determine the equivalent number of diameters of straight pipe equivalent to a 90 deg elbow having center line radii of (a) 100 per cent, (b) 150 per cent, and (c) 200 per cent of the pipe diameter.

Assume 1 velocity head lost in 50 diameters.

From Fig. 1 the per cent of velocity head lost:

- For 100 per cent radius is 25.5 per cent $\times 50 = 12.8$ diameters straight pipe.
- For 150 per cent radius is 17.0 per cent $\times 50 = 8.5$ diameters straight pipe.
- For 200 per cent radius is 14.5 per cent $\times 50 = 7.3$ diameters straight pipe.

2 ● Why is it desirable to make elbows with a radius equal to one and one-half times the pipe diameter?

Reference to Figs. 1 and 2 will show that while the loss of velocity head, as indicated by the curves, shows considerable variation for elbows between the range of 50 and 150 per cent radius, the line is practically straight after 150 per cent, indicating very little variation in loss of head for elbows of larger radius.

3 ● What is the best shape to use for ducts?

The shapes to be used in designing ducts, in the order of their preference, are round, square, and rectangular.

4 ● What determines which shape to use?

Structural and space conditions. Because ducts are as a rule part of the building or structure, it is necessary to proportion their sizes to fit the spaces available.

5 ● What is meant by "arbitrarily fix the velocity in the various sections?"

When using the velocity method as a basis for design, the maximum allowable velocity is fixed for the main supply duct at the fan, and this velocity is gradually decreased as each branch or outlet is taken off the main supply duct.

6 ● Which system of duct design is to be preferred, the velocity method or the friction pressure loss method?

The friction pressure loss method can be used to advantage where no structural or building conditions limit the shape of the ducts. Where these limiting conditions exist the velocity method is to be preferred.

7 ● Are the grille sizes figured on the same basis as the outlets?

The free area through the grilles is figured the same as the outlets, and this area is increased from 20 to 50 per cent, depending on the design of the grille, to allow for the loss of area caused by the construction of the face of the grille.

8 ● Where it is necessary to provide steel angle braces, how far apart should they be spaced?

Angle braces for large ducts should be placed on 3-ft 0-in. centers.

9 ● How much air will a 10-in. by 24-in. duct handle if it is part of a system designed on a pressure drop of 0.1 in. per 100 feet of run?

1450 cfm (Table 1 and Fig. 3).

10 ● How does a splitter at a duct junction influence the volume of the air going through each branch?

A splitter facing the direction of air flow cuts off the air and delivers the desired amount to the branch.

11 ● Why does a wide, shallow duct offer more resistance to the flow of air than does a square duct of equal cross-sectional area?

The perimeter of the wide, flat duct is greater than that of the square-section duct, so the former has the greater frictional area which increases the resistance and thus reduces the volume at any given pressure.

12 ● What methods are used to keep large ducts from vibrating because of air pulsations, and from sagging because of their own weight?

External bracing, such as standing seams, or structural shapes, like tees or angles, should be placed across the top and bottom. Exterior braces or cross buckling of metal sheets in diagonal panels may be used for the sides of large ducts.

13 ● What velocities of air flow should be used in the trunk ducts of a ventilating system in a public building?

From 1200 to 1600 fpm.

14 ● In a ventilating system in a residence, what is the recommended air velocity through supply registers and grilles?

400 fpm.

Chapter 21

INDUSTRIAL EXHAUST SYSTEMS

Classification of Systems, General Rules for Design, Suction and Velocity Requirements, Hoods, Design of Duct Systems, Collectors, Resistance of Systems, Efficiency of Exhaust Systems, Selection of Fans and Motors

SOME type of exhaust and collecting system is necessary in almost every industry and the present chapter attempts to give general information relating to the design of factory exhaust systems in order that efficient and economical control of dusts and fumes may be achieved.

CLASSIFICATION OF SYSTEMS

There are two general arrangements, the central and the group systems. In the central system a single or double fan is located near the center of the shop with a piping system radiating to the various machines to be served. In the group system, which is sometimes employed where the machines to be served are widely scattered, small individual exhaust fans are located at the center of the machine groups. The group arrangement has the advantage of flexibility.

Exhaust systems are also classified by the means employed to collect dust or other material handled. The dust or refuse may be collected and controlled by enclosing hoods, open hoods, inward air leakage, or by exhausting the general air of the room.

With some classes of machinery it is not feasible to closely hood the machines 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.

The 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. When the hood must be placed at some distance above the machine it should be large enough to encompass an area of considerable extent as diffusion is usually quite rapid.

Consideration must also be given to the natural movement of the fumes. For those that are lighter than air the hood should 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 required. If it is attempted to remove heavy dust such as lead oxides by an overhead hood the conditions may be worse than if no exhaust were used at all, owing to the rising air current carrying the dust up through the

breathing zones. The objective to keep in mind in all cases is to take advantage of the natural tendency of the material to move upward or downward.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, the removal of some dust from the machine or enclosure being 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 strong current of clean air is drawn across the operator, and away from him toward the work, where the dust is picked up and carried from the room.

GENERAL RULES FOR DESIGN

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. Finally, in choosing the fan, the area of the intake should be equal to or greater than the sum of the areas of the branch ducts. The speed, of course, 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 as follows¹:

1. Hoods, ducts, fans and collectors should be of adequate size.
2. The air velocities should be sufficient to control and convey the materials collected.
3. The hoods and ducts should not 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 design of an exhaust system should afford easy access to parts for inspection and care.

¹For more detailed requirements see Safe Practice Pamphlets Nos. 32 and 37, published by the National Safety Council, Chicago.

SUCTION AND VELOCITY REQUIREMENTS

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 to a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the

TABLE 1. SIZE OF CONNECTIONS FOR WOOD-WORKING MACHINERY

TYPE OF MACHINE	DIAMETER OF CONNECTIONS IN INCHES
Circular saws, 12-in. diam.	4
Circular saws, 12-24 in. diam.	5
Circular saws, 24-40 in. diam.	6
Band saws, blade under 2 in. wide	4
Band saws, blade 2-3 in. wide	5
Band saws, blade 3-4 in. wide	6
Band saws, blade 4-5 in. wide	7
Band saws, blade 5-6 in. wide	8
Small mortisers	6
Single end tenoners	6
Double end tenoners	7
Double end, double head tenoners	10
Planers, matchers, moulders, stickers, jointers, etc.—	
With knives, 6-10 in.	5-6
With knives, 10-20 in.	6-8
With knives, 20-30 in.	6-10
Shapers, light work	4-5
Shapers, heavy work	8
Belt sander, belt less than 6 in. wide	5
Belt sander, belt 6-10 in. wide	6
Belt sander, belt 10-14 in. wide	7
Drum sander, 24 in.	5
Drum sander, 30 in.	6
Drum sander, 36 in.	7
Drum sander, 48 in.	8
Drum sander, over 48 in.	10
Disc sander, 24 in. diam.	5
Disc sander, 26-36 in. diam.	6
Disc sander, 36-48 in. diam.	7
Arm sander	4

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 should 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.

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. This is of considerable value where exhaust systems adapted to particular operations have been standardized by practice. Tables 1 and 2 present the duct sizes usually employed for standard wood-working machinery and for grinding and buffing wheels. Static pressures which in practice have been found necessary to control and convey various materials, are given in Table 3. It must be remembered, however, that the *suction* is merely a rough

TABLE 2. SIZE OF CONNECTIONS FOR GRINDING AND BUFFING WHEELS

DIAMETER OF WHEELS	MAX. GRINDING SURFACE SQ. IN.	MIN. DIAM. OF BRANCH PIPES IN INCHES
Grinding—		
6 in. or less, not over 1 in. thick	19	3
7 in. to 9 in., inclusive, not over 1½ in. thick	43	3½
10 in. to 16 in., " " " 2 in. " "	101	4
17 in. to 19 in., " " " 3 in. " "	180	4½
20 in. to 24 in., " " " 4 in. " "	302	5
25 in. to 30 in., " " " 5 in. " "	472	6
Buffing—		
6 in. or less, not over 1 in. thick	19	3½
7 in. to 12 in., inclusive, not over 1½ in. thick	57	4
13 in. to 16 in., " " " 2 in. " "	101	4½
17 in. to 20 in., " " " 3 in. " "	189	5
21 in. to 27 in., " " " 4 in. " "	338	6
27 in. to 33 in., " " " 5 in. " "	518	7

measure of the air volume handled and consequently of the air velocity at the opening of the hood. The elimination of any dusty condition requires added information concerning the shape, size and location of the hood used with regard to the operation in question.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. The static suction requirements, which range from 1½ to 5 in. water displacement in a U-tube, should be followed although in several instances they may 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 about the lower rim of the wheel.

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations they should not be less

TABLE 3. SUCTION PRESSURES REQUIRED AT HOODS

TYPE OF INSTALLATION	STATIC SUCTION IN INCHES OF WATER
Exhausting from grinding and buffing wheels	1½-5
Exhausting from tumbling barrels	2
Exhausting from wood-working machinery—light duty	2
Exhausting from wood-working machinery—heavy duty	2-4
Shoe machinery exhaust	2-3
Exhausting from rubber manufacturing processes	2
Flint grinding exhaust	2
Exhausting from pottery processes	2
Lead dust and fume exhaust	2-4
Fur and felt machinery exhaust	2-3
Exhausting from textile machinery	2-3
Exhausting from elevating and crushing machinery	2
Conveying bulky and heavy materials	3-5

than 200 fpm at the point of origin. For granite dust generated by pneumatic devices, Hatch et al² give velocities from 150 to 200 fpm, depending on the type of hood used, as sufficient for safe control. 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 below.

HOODS

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 dusts and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many 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, the following formula may be used to determine the air velocity at any point along the axis³:

$$V = \frac{0.1 Q}{x^2 + 0.1 A} \quad (1)$$

where

V = velocity at point, feet per minute.

A = area of opening, square feet.

x = distance along axis, feet.

Q = volume of air handled, cubic feet per minute.

Velocity Contours

It is possible by use of a specially constructed pitot-tube⁴ 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 hood⁵.

¹Hatch, Theodore, Drinker, Philip, and Choate, Sarah P., 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. (*Journal of Industrial Hygiene*, Vol. XII, No. 3, March, 1930).

²The Control of Industrial Dust, by J. M. DallaValle (*Mechanical Engineering*, Vol. 55, No. 10, October, 1933).

³Studies in the Design of Local Exhaust Hoods, by J. M. DallaValle and Theodore Hatch (TRANSACTIONS A.S.M.E., Vol. 54, 1932).

⁴Velocity Characteristics of Hoods under Suction, by J. M. DallaValle (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

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.

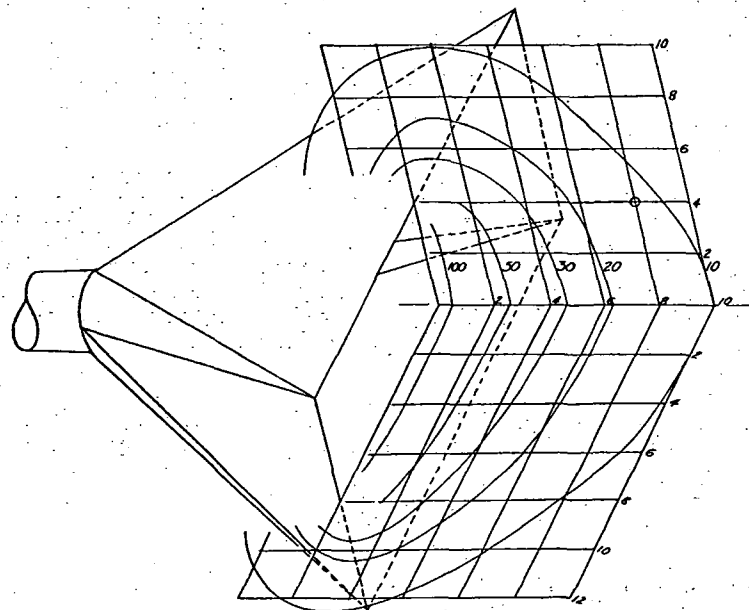


FIG. 1. VELOCITY CONTOURS FOR A RECTANGULAR OPENING WITH A SIDE RATIO OF ONE-HALF. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE VELOCITY AT THE OPENING

Air Flow from Static Readings

The volume of air flow through any hood may be determined from the following equation:

$$Q = 4005 f a \sqrt{h_t} \quad (2)$$

where

- Q = volume of air flow, cubic feet per minute.
- a = area of connecting duct, square feet.
- h_t = static suction at 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:

$$f = \sqrt{\frac{h_v}{h_t}} \quad (3)$$

where h_v is the velocity head in the connecting duct.

The term *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 respect to the hood are known. This is clearly indicated by Equation 1 which shows that the velocity at any point along the axis varies inversely as the area of the opening and the square of the distance. However, this formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Large Open Hoods

Large hoods, such as are used for electroplating and pickling tanks, should be subdivided 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 distribution 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 following simple formula:

$$Q = 1.4PDV \quad (4)$$

where

- Q = total volume of air handled by hood, cfm.
- P = perimeter of the tank, feet.
- D = distance between tank and hood opening, feet.
- V = air velocity desired along edges and surface of tank, fpm.

Lateral Exhaust Systems

The lateral exhaust method, as developed for chromium plating⁶, 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 2 in. wide and for effective

⁶See Health Hazards in Chromium Plating, by Bloomfield, J. J., and Blum, Wm. (U. S. Public Health Report, Vol. 43, No. 26, September 7, 1928).

ventilation a 2,000 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.

Flexible Exhaust Systems

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 below 100 parts per million. 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 in a horizontal position slightly below the object sprayed. Stagnant regions within the booth should be carefully avoided or should be provided with exhaust. The air volume should be sufficient to maintain a velocity of 150 to 200 fpm over the open area of the booth, and the vapors may be discharged through a suitable stack to permit dilution, but it is better practice to pass the fumes or vapors through baffle type washers or scrubbers designed for efficient spray fume 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 opened to its maximum height.

DUCT SYSTEM DESIGN

The duct system should be large enough to transport the fumes or material without causing serious obstruction to the air flow. It is good practice to proportion the ducts to obtain the desired velocities and suction pressures at the hoods, although in many cases only an approximation to an ideal design is possible. Many exhaust hoods, and par-

ticularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an

TABLE 4. GAGE OF SHEET METAL TO BE USED FOR VARIOUS DUCT DIAMETERS

DIAMETER OF DUCT	GAGE OF METAL
8 in. or less.....	24
9 to 18 in.....	22
19 to 25 in.....	20
26 in. or more.....	18

acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points where the material from one branch would tend to enter the branch on the opposite side of the main.

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.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

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 sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

*For a discussion of spray booths, see Special Bulletin No. 16, Spray Painting in Pennsylvania, Department of Labor and Industry, 1926, Harrisburg, Pa.

TABLE 5. AIR SPEEDS IN DUCTS NECESSARY TO CONVEY VARIOUS MATERIALS

MATERIAL	AIR VELOCITIES (FPM)
Grain dust.....	2000
Wood chips and shavings.....	3000
Sawdust.....	2000
Jute dust.....	2000
Rubber dust.....	2000
Lint.....	1500
Metal dust (grindings).....	2200
Lead dusts.....	5000
Brass turnings (fine).....	4000
Fine coal.....	4000

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 necessary and is frequently done at the expense of a reduced air velocity, it is none the less advisable where future expansion of the exhaust system is contemplated.

Air Velocities in Ducts

When the static suction has been fixed for a given hood, the air velocity in the duct may be determined from Equation 2. Air velocities for conveying a material should be moderate. Table 5 gives the velocities generally employed for conveying various substances. Equations 5a and 5b may be used as tests to determine the conveying efficiency of a system⁸. Velocities determined from these formulae should be increased by at least 25 per cent since they represent the minimum at which a stated size and density of material can be transported.

$$\text{For vertical ducts: } V = 13,300 \frac{s}{s+1} d^{0.670} \quad (5a)$$

$$\text{For horizontal ducts: } V = 6000 \frac{s}{s+1} d^{0.398} \quad (5b)$$

where

V = air velocity in duct, feet per minute.

s = specific gravity of particles.

d = average diameter of largest particles conveyed, inches.

Example 2. Granular material, the largest size of which is approximately 0.37 in. in diameter, with a specific gravity of 1.40 is to be conveyed in a vertical pipe the velocity of the air in which is 4100 fpm; find whether the material can be transported at this velocity.

Substitute data in Equation 5a and multiply by 1.25:

$$V = 1.25 \times 13,300 \times \frac{1.4}{2.4} \times 0.37^{0.67}$$

Antilog $(0.57 \times \log 0.37) = 0.568$; the required velocity is, therefore, 5500 fpm.

⁸DallaValle, J. M.: Determining Minimum Air Velocities for Exhaust Systems. (A.S.H.V.E. Journal Section, Heating, Piping and Air Conditioning, September, 1932).

TABLE 6. LOSS THROUGH 90-DEG ELBOWS

ELBOW CENTER LINE RADIUS IN PER CENT OF PIPE DIAMETER	LOSS IN PER CENT OF VELOCITY HEAD
50	75
100	26
150	17
200 to 300	14

Hence, the duct velocity must be increased either by speeding up the fan or decreasing the diameter of the duct, or both.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 3, Chapter 20. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 50 to 300 per cent of pipe diameter, the loss may be estimated from Table 6. 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 throat radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a throat diameter radius $1\frac{1}{2}$ times the diameter, the resistance is about the same as that of seven diameters of straight pipe.

COLLECTORS

The most common method of separating the dust and other materials from the air is to pass the mixture through a centrifugal or *cyclone* collector. In this type of collector the mixture of the air and material is introduced on a tangent, near the cylindrical top of the collector, and the whirling motion sets up a centrifugal action causing the comparatively heavy materials suspended in the air to be thrown against the side of the separator, from which position they spiral down to the tail piece, while the air escapes through the stack at the center of the collector.

The diameter of the cyclone should be at least $3\frac{1}{2}$ times the diameter of the fan discharge duct. When two or more separate ducts enter a cyclone, gates should be provided to prevent any back draft through a system which may not be operating. Cyclones working in conjunction with two or more fans should be designed to operate efficiently at two-thirds capacity rating. The following formula is useful in computing the loss through a cyclone when the velocity of the air in the fan discharge duct is known:

$$h_c = 0.13 \left(\frac{V}{1000} \right)^2 \quad (6)$$

where

h_c = the pressure drop through the cyclone, inches of water.

V = the air velocity in the fan discharge duct, feet per minute.

If a cyclone is used to collect light dusts such as buffing wheel dusts,

feathers and lint, the exhaust vent should be large enough to permit an air velocity of 200 to 500 fpm. This will, of course, require a cyclone of larger dimensions than given for the foregoing general case.

When a high collection efficiency is desired, or the material is very fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain separation. The capacities and characteristics of this type of separator should be obtained from the manufacturers.

Cloth Filters

Filters are used when the material collected by an exhaust system is valuable or cannot be separated efficiently from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. Bag houses used in the manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen cloths are employed, the filtering capacities vary from $\frac{1}{2}$ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The time increase of resistance varies with the amount of material permitted to build up on the surface of the filter and can be determined only by experiment. The limits of the increase may be regulated by adjustment of the shaking or cleaning mechanism. These limits may be regulated further according to the capacity of the fan and the effective performance of the hoods and the duct system.

For additional information on Dust and Cinders, see Chapter 15, Air Pollution, p. 293.

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 pipes.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. The collector drop in inches of water is given approximately by Equation 6, but where possible the resistance of the particular collector to be used should be ascertained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. Find the velocities 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 6. 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 limits⁹.

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. Commonly accepted values of threshold limits for the usual gases and vapors are given in Table 7.

SELECTION OF FANS AND MOTORS

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. For

TABLE 7. THRESHOLD LIMITS OF COMMON VAPORS AND GASES^a

SUBSTANCE	SPEC. GRAV. OF GAS OR VAPOR (AIR 1)	INFLAMMABLE LIMITS (%)	PHYSIOLOGICAL ACTION	MAXIMUM ALLOWABLE CONCENTRATION (PPM)
Chlorine.....	2.486	non-inflamm.	irritant	0.35
Ozone.....	5.5	do	do	0.80
Hydrogen chloride.....	1.2678	do	do	10.0
Sulphur dioxide.....	2.2638	do	do	10.0
Carbon monoxide.....	0.9671	12.5-74	asphyxiant	100.0
Hydrogen sulphide.....	1.190	4.3-46	do	85-130
Benzene.....	2.73	1.4-7.0	anesthetic	100.0
Methanol.....	1.1	7.5-26.5	do	100.0
Carbon tetrachloride.....	5.3	non-inflamm.	do	100.0

^aTaken from The Prevention of Occupational Diseases, by R. R. Sayers and J. M. DallaValle (*Mechanical Engineering*, Vol. 57, No. 4, April, 1935).

particular features concerning special fans, consult the *Catalog Data Section* of THE GUIDE and manufacturers' data. 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.

If no future expansion of an exhaust system is contemplated, the fan motor should be chosen to provide the calculated air volume. Should, however, the exhaust system be required to handle more air in the future, the motor should be adequate for the maximum load anticipated. Further information regarding the choice of fans and motors is given in Chapters 17 and 42.

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

⁹Criteria for Industrial Exhaust Systems, by J. J. Bloomfield (A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934).

TABLE 8. MATERIALS TO BE USED FOR THE PROTECTION OF EXHAUST SYSTEMS AGAINST CORROSION^a

TYPE OF FUME CONVEYED	PROTECTIVE MATERIAL TO BE USED
Chlorine.....	Rubber lining or chrome-nickel alloys
Hydrogen sulphide.....	Aluminum coated iron, aluminum, high chrome-nickel alloys
Ammonia.....	Iron or steel
Sulphurous gases.....	High chrome-nickel alloys
Hydrochloric acid.....	Rubber lining, chrome-nickel alloys
Nitrous gases.....	Nickel-chrome alloys

^aCondensed from data given by Chilton and Huey (*Industrial and Engineering Chemistry*, Vol. 24, 1932).

chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 8. 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.

PROBLEMS IN PRACTICE

1 ● What determines the efficiency of an exhaust system?

It is dependent upon its effectiveness in reducing the concentration of dust, fumes, vapors and gases below the safe or threshold limit.

2 ● Are state regulatory requirements as to suction applicable to all sorts of dust collecting installations?

As a rule the regulations refer only to grinding wheel and buffing wheel systems.

3 ● What is the most common method of reducing total air volumes handled in cases employing large hoods over apparatus covering a large area?

The use of the petticoats on large hoods which permits a comparatively high air velocity at the rim of the hood and controllably small velocities in the center.

4 ● What other types of collectors are available for use in the place of cyclones and filters when chemical and physical conditions obviate the possibility of the use of them?

Devices such as scrubbers and contactors, using water or other contacting liquids, and electrical precipitators.

5 ● What is the most frequent error made in dust collector system design?

The omission of some means of putting into the workroom air having the proper characteristics to replace that which has been exhausted.

6 ● Are there available means for testing the performance of dust collecting systems when they are required to meet high industrial hygienic standards?

Yes. Such means are set up by the United States Public Health Service and by the Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. (Chapter 44).

7 ● Why is it not permissible to connect up emery wheels and buffing wheels to the same exhaust system?

Emery wheels and buffing wheels should be handled by separate systems because of the fire hazard, as it is possible for sparks from the emery wheels to ignite the lint and dust from the buffing wheels when both are carried through the same system.

8 ● Give an important characteristic of centrifugal type dust collectors which should be given consideration when applying this type of collector to installations requiring high separating efficiencies.

The separating action of a cyclone or centrifugal type collector depends largely on centrifugal force. Reducing the radius of air flow increases the centrifugal force for a given velocity of flow. Accordingly, the smaller size units usually give higher separating factors, and better results can sometimes be obtained by using a number of small collectors instead of one large unit.

9 ● Mention some general suggestions relating to the design of efficient industrial exhaust systems.

a. Endeavor to obtain a maximum degree of effectiveness with a minimum volume of air, by the use of well designed hoods closing in the sources of fumes or material to be removed so located as to take advantage of the natural direction taken by the fumes or materials when leaving their source.

b. Give particular care to the velocity of flow. The duct velocities for material conveying systems must be high enough to properly carry the material, but they should not be higher than necessary because excessive velocities increase the pressure requirements and result in a waste of power.

c. Select the type of fan best suited to the job. For installations where stringy material is handled do not use a fan wheel which has a shroud.

d. When handling the refuse from various machines, study the grouping and operating cycles of the machines. Connecting a large number of machines into one system is frequently very uneconomical.

e. Avoid unnecessary distances and bends in laying out the piping system.

10 ● The static pressure measured at the throat of a buffing wheel hood is 2 in. and the velocity head measured with a Pitot tube is 1.6 in. Calculate the restriction coefficient f.

From Equation 2, $V = 4005 f \sqrt{h_t}$.

From the theory of air flow, $V = 4005 \sqrt{h_v}$.

Hence, $\sqrt{h_v} = f \sqrt{h_t}$.

or

$$f = \sqrt{\frac{h_v}{h_t}} = \sqrt{\frac{1.6}{2.0}} = 0.89$$

11 ● A tank, 4 ft by 8 ft, contains a fluid which gives off injurious vapors. A large hood is located 30 in. above the top of the tank and extends slightly over its edges. Assuming that a velocity of 60 fpm is required to adequately control the vapors near the edges of the tank, calculate the air flow required.

Using Equation 4, $P = 2 \times 4 + 2 \times 8 = 24$ ft; $D = 30$ inches = 2.5 ft; $V = 60$ fpm.

Hence, $Q = 1.4 \times 24 \times 2.5 \times 60 = 5040$ cfm.

12 ● Silica dust with a specific gravity of 2.65 is being conveyed in a duct system. The velocity measured in a vertical portion of the system is found to be 2700 fpm. What is the maximum diameter particle transported at this velocity?

$$\text{Using Equation 5a, } 2700 = 13,300 \times \frac{2.65}{3.65} \times d^{0.670}$$

from which

$$d = (0.28)^{1.75} = 0.11 \text{ in.}$$

13 ● What important factors must be considered in the determination of resistance for an exhaust system?

The maintained resistance of an exhaust system is composed of three factors, namely, (1) loss through the hoods, (2) friction drop through the pipe system, and (3) collector drop.

14 ● What special materials may be used to resist chemical corrosion in a system exhausting gases and fumes?

Various protective materials are available for exhaust systems depending largely upon the type of fumes conveyed. Nickel-chrome alloys, aluminum coated metals and rubber linings are extensively used. Also protective rubberized paints are available which may be applied for conveying chlorine and hydrochloric acid fumes.

Chapter 22

FAN SYSTEMS OF HEATING

Types of Systems, Blow-Through, Draw-Through, Heating Units, Design, Temperatures, Weight of Air to be Circulated, Temperature Loss in Ducts, Heat Supplied, Heating Units and Washer, Grate Area, Boiler Selection, Weight of Condensate, Static Pressure, Fans and Control

A FAN system of heating depends upon fans and blowers to distribute air through ducts from one centrally located plant. This chapter considers heating and humidifying systems of this type whereas similar systems arranged for cooling and dehumidifying are discussed in Chapter 9. A special type of central fan system, the mechanical warm air or fan furnace system, which is especially adapted to residences, churches, halls, and other small buildings, is covered in Chapter 23.

TYPES OF SYSTEMS

In the indirect type of central fan heating and air conditioning systems, steam is usually the medium by which heat is transferred from the boiler, or other source of heat, to the heating units. If the system is intended solely for heating, the air is passed over one or more stacks or batteries of heating units and then conveyed to the spaces for which it is intended through a system of ducts. In some cases, a predetermined amount of outside air is introduced for ventilating purposes, whereas in others the moisture content is controlled by passing the air through a washer or humidifier. If the apparatus is designed to control simultaneously the temperature, humidity, air motion, and distribution, it is known as an air conditioning system.

In the *split system*, the heating is accomplished by means of radiators or convectors, and the ventilating or air conditioning by means of the central fan apparatus. In the *combined system*, the entire operation of heating, ventilating, and air conditioning is handled by the central fan system.

A common arrangement of the central fan system of heating is illustrated by Fig. 1 and consists of a fan, a heating unit (heater) enclosed by a sheet metal casing connected with the suction side of the fan, a sheet metal casing connected to the heating unit casing run to the outside of the building and provided with an adjustable opening inside the building for recirculation of the air when desired, and a duct system attached to the fan outlet to convey and distribute the air to various parts of the building to be warmed by the apparatus. The fan is ordinarily motor-driven; there are, however, many cases when a direct-connected steam engine may be used to advantage. In this event the exhaust from the engine can be con-

nected to one or more sections of the heater, depending upon the condensation rate of the engine. The recirculation duct connected with the opening in the suction duct should be extended to a point as near the floor as possible.

When ventilation is not a requirement or is considered relatively unimportant, as in shop and factory heating, and the number of persons vitiating the air is small compared with the cubical contents of the building, or the process does not generate obnoxious gas or vapors, the air may be recirculated, sufficient outside air for ventilation being supplied by infiltra-

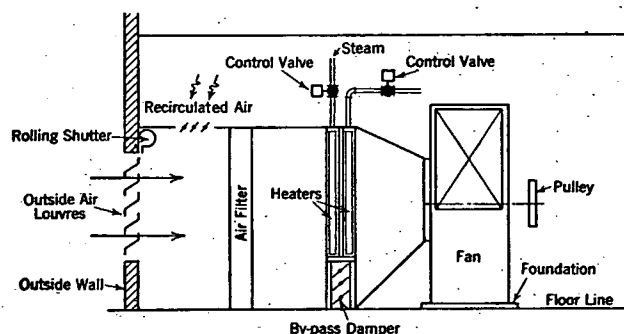


FIG. 1. ARRANGEMENT OF A CENTRAL FAN HEATING SYSTEM (DRAW-THROUGH)

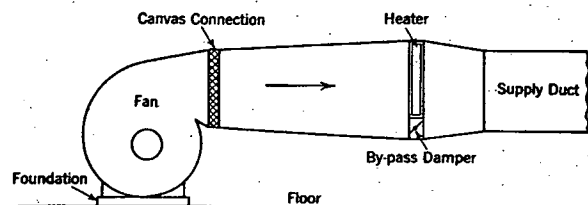


FIG. 2. ARRANGEMENT FOR HEATING UNIT (BLOW-THROUGH)

tion. The amount of heat to be supplied the heating unit in this case is the same as would be required for a direct radiation installation.

When ventilation is a requirement to be met, an arrangement similar to that shown by Fig. 1 may be employed. Since the amount of air necessary for heating is generally in excess of the amount required for ventilation, considerable fuel economy may be effected by recirculating a portion of the air. In this case only sufficient outside air is drawn into the system to meet the ventilation requirement and the remainder of the air, required for heating, is recirculated. This may be readily effected by an arrangement of ducts and dampers on the suction side of the fan as previously mentioned. If the outside air introduced is to be washed or conditioned the washer or humidifier and tempering coil may be added between the inlet for the recirculated air and the fresh air intake.

Blow-Through, Draw-Through

When the heating unit is located on the suction side of the fan, the system is known as *draw-through*. (See Fig. 1.) When the heating unit is located in the discharge from the fan, the system is known as *blow-through*. (See Fig. 2.) The draw-through combination is used for factory and toilet room installations because a more compact arrangement of the apparatus usually is possible. In addition, air leakage will be inward. The blow-through combination is used principally in schools and public buildings, and for all booster coil arrangements where different temperatures and independent temperature regulation are required for different heated spaces. In public building installations, the fan frequently blows the heated air into a plenum chamber from which the air ducts radiate to the various rooms of the building; this arrangement is sometimes called the *plenum* system.

HEATING UNITS

The heating units for central fan systems using steam as the heating medium may be classified as (1) tempering coils, (2) preheater coils, (3) reheater coils, (4) booster coils, and (5) water heaters, either open or closed. *Tempering coils* are used with ventilating and air conditioning systems for raising the temperature of the outside cold air to above freezing, or 32 F. They are not required for heating systems where all of the air is recirculated, since the temperature of the recirculated air will be above freezing. *Preheater coils* are used with air conditioning systems to raise the temperature of the air from that leaving the tempering coils to such a temperature that in passing through the water sprays of the washer (without water heater) the air will become partially saturated (adiabatically) having a moisture content corresponding to the required dew-point temperature. *Preheater coils* therefore supply heat as necessary to control the dew-point temperature. The *reheater coils* are used to raise the temperature of the air leaving the tempering coils (in the case of a heating or ventilating system) or the air leaving the washer (in the case of an air conditioning system) to that necessary to maintain the desired temperature in the rooms or spaces to be heated or conditioned, except where booster coils are used, in which case the reheater coils raise the air temperature to approximately room temperature, or slightly higher. *Booster coils* are installed in the duct branches to control the temperature of the air entering the rooms or spaces for which it is intended. *Water heaters* are used on an air conditioning system to control the dew-point temperature. They are used mainly for industrial work, seldom for comfort conditioning. They are not used where preheater coils are employed. The open type supplies steam directly to the spray water, while the closed type utilizes a heat interchanger by which the steam imparts its heat to the spray water. Where water heaters are required for comfort conditioning, the closed type is used.

The heating units for central fan systems in use at the present time consist either of pipe coils, finned tubes of steel, copper, brass or other metal, cast-iron sections with extended surfaces, or the cellular type. Steam is passed through these heating units and the air to be heated is passed over their exterior surfaces.

In selecting a heating unit for any particular service, the choice should be based on the desired requirements as follows:

1. Final temperature desired.
2. Loss in pressure (or friction) of air passing over the heating unit.
3. Air velocity over the heating unit.
4. Free area or face area of heating unit.
5. Ratio of heating surface to net free (or face) area.
6. Air volume required.
7. Number of rows of pipes, tubes, or sections.
8. Amount of heating surface.
9. Steam pressure drop through the heating unit.
10. Weight of heating unit.

Final Temperature Desired. The choice of a heating unit is largely influenced by the final temperature desired, when the entering air temperature and steam pressure available at the heating unit are specified. These data are obtainable from manufacturers' catalogs.

Loss in Air Pressure (or Friction). The allowable friction through the heating unit is one of the first factors to be determined in the selection of the apparatus. The velocities of air through various types of heating units will not necessarily be the same, but for any particular job the velocity through the heating unit should be a secondary consideration and the allowable friction or air pressure loss should be fixed approximately before proceeding with the selection of the heating unit. The loss in air pressure (or friction) through the heating unit should not exceed a predetermined maximum allowable amount for economical operation and for moderate size and first cost of installation.

In public building work, the maximum allowable friction through both tempering coil and reheater coils should never exceed $\frac{5}{8}$ in. of water and it is advisable that the friction be kept considerably lower than this figure if possible. A tempering coil friction ranging from 0.10 to 0.20 in. of water is considered satisfactory. The air pressure loss for reheaters ordinarily ranges from 0.20 to 0.40 in. of water. In factory work, the maximum friction through the heater should never exceed 0.8 in. or 1 in. of water and it is advisable to figure the heaters at lower frictions if possible.

Velocity through Heating Unit. This velocity has generally been given in manufacturers' tables as being measured at 70 F and in most cases refers to the velocity through the net free area of the heating unit, or through the net space between the pipes, tubes or sections. Although most manufacturers give suitable velocities measured at 70 F, certain manufacturers show velocities measured at 65 F and others indicate velocities measured at the average air temperature through the heating unit. Many new heating units, however, specify net face areas with corresponding velocities instead of velocities through net free areas. In either case, manufacturers publish the corresponding friction or air-pressure loss in tables. The velocity through the net free area of the heating unit averages about 1000 fpm and that through the net face area about 500 fpm.

The volume of air to be heated in any particular case is determined after consideration of the ventilation requirements, heat losses, and quantity of air required for proper circulation, as explained in Chapters 3 and 7.

The number of rows of pipes, tubes, or sections or the amount of heating surface to be used may be selected from manufacturers' catalogs after the quantity of air handled and the heat load are known. Savings in operating expense or cost of installation should result from a proper selection of heater and by-pass areas. For example, instead of having the entire air quantity go through a one-row heating unit, it may be advantageous to use a two-row heating unit and a properly sized by-pass. Thus, when no heating is being done, a suitable by-pass damper may be opened to place a lighter load on the fan.

The steam pressure drop through the heating unit is also tabulated in manufacturers' data tables. The sizing of steam supply and return piping, allowing for drops through heating units, is explained in Chapter 32.

Weight of Heating Unit. In the design of a heating system, the weight limitations of heating units are determined by the location of the units. Obviously, if there is no loading limitation imposed, any type of heating unit may be selected. On the other hand if the heating unit is to be hung from the ceiling, it may be desirable to use the lightest unit which will accomplish the work required.

DESIGNING THE SYSTEM

The general procedure for the design of central fan systems is as follows:

1. Calculate the heat loss for each room or space to be heated.
2. Determine volume of outside air to be introduced.
3. Assume or calculate temperature of air leaving registers or supply outlets.
4. Calculate weight of air to be circulated.
5. Estimate temperature loss in duct system.
6. Calculate heat to be supplied the heating units and washer.
7. Select heating units and washer from manufacturers' data and performance curves.
8. Calculate total heat to be supplied.
9. Calculate grate area and select boiler.
10. Design duct system.
11. Calculate total static pressure of system.
12. Select fan, motor, and drive.

The heat losses (H) should be calculated in accordance with the procedure outlined in Chapter 7. If a positive pressure is maintained by the central fan system in the room or space to be ventilated or conditioned, there will ordinarily be very little infiltration of cold outside air through the cracks and crevices of the space. Consequently, the volume of air introduced into the space at the assumed or calculated outlet temperature need only be sufficient to provide for the transmission losses, plus about one-third of the infiltration losses. The exfiltration of heated or conditioned air through the cracks and crevices of the space should be provided for by making the usual allowance for the infiltration losses in arriving at the total heat loss of the space. The air required to make up for this exfiltration of heated or conditioned air will be brought in at the outside air intake and may be included as a part of the outside air neces-

sary for the ventilating requirements. The heat required to raise this air to the conditions maintained in the room must be provided by the tempering coils, preheater coils, and reheater coils. If a positive pressure is not maintained in the room or space to be conditioned, the normal infiltration of outside cold air will take place in this room, and the outlet temperature, together with the required air volume at this temperature, must be sufficient to provide for both infiltration and transmission losses.

Volume of Outside Air

The volume of outside air required for ventilation or air conditioning purposes may be determined from data in Chapter 3. In no case shall less than 10 cfm per person be introduced.

The heat required to warm the outside air introduced for ventilation purposes (H_o) may be determined by means of the following formula:

$$H_o = 0.24 (t - t_o) M_o \quad (1)$$

where

0.24 = specific heat of air at constant pressure.

t = room temperature, degrees Fahrenheit.

t_o = outside temperature, degrees Fahrenheit.

M_o = weight of outside air to be introduced per hour, in pounds = 60 $d_o Q_o$.

Q_o = volume of outside air to be introduced, cubic feet per minute.

d_o = density of air at t_o , pounds per cubic foot.

Example 1. A building in which the temperature to be maintained at 70 F requires 10,000 cfm. If the outside temperature is 20 F, how much heat will be required to warm the air introduced for ventilation purposes to the room temperature?

Solution. $10,000 \times 60 = 600,000$ cfh; $d_o = 0.08276$ (Table 1, Chapter 1); $M_o = 0.08276 \times 600,000 = 49,656$ lb; $t = 70$ F; $t_o = 20$ F; $H_o = 0.24 \times (70 - 20) \times 49,656 = 595,872$ Btu per hour.

Temperature of Air Leaving Registers

If the system is to function only as a heating system, that is, entirely as a recirculating one, the temperature of the air leaving the register outlets must be assumed. For public buildings, these temperatures may range from 100 to 120 F, whereas for factories and industrial buildings the outlet or register temperature may be as high as 140 F. In no case should the outlet temperature exceed these values.

For ventilating or conditioning systems, the temperature of the air leaving the supply outlets may be estimated by means of the following formula:

$$t_y = \frac{H}{60 d Q \times 0.24} + t \quad (2)$$

where

t_y = outlet temperature, degrees Fahrenheit.

H = heat loss of room or space to be conditioned, Btu per hour.

Q = total volume of air to be introduced at the temperature t , cubic feet per minute.

d = density of air, pounds per cubic foot.

If the outlet temperature (t_y) as determined from Equation 2 exceeds 120 F for public buildings, or 140 F for factories or industrial buildings,

these respective outlet temperatures should be used as factors in the following equation to determine the volume of air to be introduced into the room or space:

$$Q = \frac{H}{60 d \times 0.24 (t_y - t)} \quad (3)$$

Example 2. The heat loss of a certain auditorium to be conditioned is 100,000 Btu per hour. The ventilating requirements are 1,500 cfm and the room temperature 70 F. Determine the outlet temperature.

Solution. Substituting in Formula 2,

$$t_y = \frac{100,000}{60 \times 0.07495 \times 1500 \times 0.24} + 70 = 131.7 \text{ F}$$

Inasmuch as this temperature is excessive, it will be necessary to assume an outlet temperature, which will be taken as 120 F, and to calculate the amount of air to be introduced into the room at this temperature to provide for the heat loss. Substituting in Equation 3,

$$Q = \frac{100,000}{60 \times 0.07495 \times 0.24 (120 - 70)} = 1850 \text{ cfm (at temperature } t)$$

Weight of Air to be Circulated

The total weight of air (M) to be introduced into the room or space to be heated or conditioned is given by the following formulae:

$$M = \frac{H}{0.24(t_y - t)} = 60 d Q \quad (4)$$

$$M = M_o + M_r \quad (5)$$

$$M_o = 60 d_o Q_o \quad (6)$$

where

d = density of air at temperature t , pounds per cubic foot.

d_o = density of air at temperature t_o , pounds per cubic foot.

Q_o = volume of outside air at temperature t_o , cubic feet per minute.

M_o = weight of outside air, pounds per hour.

M_r = weight of recirculated air, pounds per hour.

Example 3. Using the data of Example 2 and an outside temperature of 20 F, what will be the values of M , M_o and M_r ?

Solution. $d = 0.07495$; $d_o = 0.08276$; $Q = 1850$; $Q_o = 1500$; $H = 100,000$.

$$M = \frac{100,000}{0.24 \times (120 - 70)} = 8,333 \text{ lb}$$

$$M_o = 0.08276 \times 60 \times 1500 = 7,448 \text{ lb}$$

$$M_r = M - M_o = 8,333 - 7,448 = 885 \text{ lb}$$

Temperature Loss in Ducts

The allowances (t_z) to be made for temperature drop through the duct system are as follows:

1. When the duct system is located in the enclosure to which the air is being delivered, as in a factory, it may be assumed that there is no loss between the reheater coil and the point or points of discharge into the enclosure.

2. For ducts run underground an allowance shall be made based on the estimated heat loss of the duct, assuming the average temperature of the ground to be 55 F.

3. For galvanized ducts with the usual ranges of air temperature and velocity, the coefficient of heat transmission may be taken as 1.7 Btu per hour per degree difference between the mean temperature of the air in the duct and that surrounding the duct.

The heat loss may then be expressed by

$$H = 1.7 \pi DL \left(\frac{t_i + t_y}{2} \right) - t_o \quad (7)$$

and also by

$$H = 0.24 M (t_i - t_y) = 60 \frac{\pi}{4} D^2 V d \times 0.24 (t_i - t_y) \quad (8)$$

Equating (7) and (8)

$$1.7 \pi DL \left(\frac{t_i + t_y}{2} \right) - t_o = 3.6 \pi D V d (t_i - t_y)$$

$$\frac{t_i + t_y - 2t_o}{t_i - t_y} = \frac{4.235 D V d}{L} \quad (9)$$

where

H = heat loss from the duct, Btu per hour.

D = diameter of duct, feet.

L = length of duct, feet.

t_i = temperature of air entering the duct.

t_y = temperature of air leaving the duct.

t_o = temperature of air surrounding the duct.

M = weight of air passing a given cross section of the duct per hour.

V = velocity of air in the duct, feet per minute, at specified temperature.

d = density of the air at the specified temperature at which V is measured.

Usually all the values in Formula 9 may be approximated except t_i , the initial or entering temperature, which can be readily found where the others are known or assumed.

Example 4. Determine the temperature drop in a galvanized duct 20 in. diameter and 60 ft long carrying air at a velocity of 1200 fpm measured at 70 F, to be delivered at a temperature of 140 F when the air surrounding the duct is at a temperature of 50 F.

Solution. Substituting in Formula 9,

$$\frac{t_i + 140 - (2 \times 50)}{t_i - 140} = \frac{4.235 \times 1.666 \times 1200 \times 0.07495}{60}$$

$$t_i = 158.7 \text{ F}$$

$$\text{temperature drop} = 158.7 - 140 = 18.7 \text{ F}$$

Example 5. An uninsulated 12 in. diameter galvanized duct extends 50 ft through an unconditioned room to supply 80 F cool air to an adjacent space. If the average temperature of the air surrounding the duct in the unconditioned room is 100 F and the velocity of the air through the duct is 1000 fpm, measured at 70 F, determine the temperature gain which must be allowed in passing the air through the unconditioned room.

Solution. Substituting in Formula 9,

$$\frac{t_i + 80 - (2 \times 100)}{t_i - 80} = \frac{4.235 \times 1.0 \times 1000 \times 0.07495}{50}$$

$$t_i - 120 = 6.34 (t_i - 80)$$

$$-5.34 t_i = 120 - 507$$

$$t_i = 72.3 \text{ F}$$

$$\text{temperature gain} = 80 - 72.3 = 7.7 \text{ F}$$

Therefore, air having a temperature of 72.3 F must be introduced at the end of the 50 ft duct in order to supply 80 F air to the conditioned space.

Heat Supplied Heating Units and Washer

The following cases may arise in practice:

A. The heating of the building is done entirely by means of a central fan system, all of the air being drawn from the outside.

B. Similar to A, except that all of the air is recirculated.

C. A portion of the air is recirculated, and the remainder is drawn in from the outside.

D. Air at the same temperature is to be delivered to all the rooms. A constant relative humidity is maintained in the building and all of the air circulated is drawn from outside the building. (Not applicable to the heating of various rooms where individual control of each room is desired.)

E. Outside air, return air, and by-pass air are used with the reheater located in by-pass air chamber.

F. Arrangement of apparatus where individual control of the temperature for each room is required in conjunction with air washer equipment to maintain a constant relative humidity in the rooms. The air washer is provided with a water heater for the spray water, capable of fully saturating the air. A section of preheater may be used for this purpose in place of the water heater. With this arrangement and with a uniform temperature of air entering the rooms, it is impossible to maintain the same room temperature throughout the building because the weight of air to be delivered to each room is determined and fixed by the ventilating requirements.

In analyzing these cases, the following symbols will be used:

H = heat loss of the room or building, Btu per hour.

H_1 = heat to be supplied to the reheater coil, Btu per hour.

H_2 = heat supplied tempering coil, or tempering coil and preheater, Btu per hour.

H_3 = heat supplied air washer by water heater, Btu per hour.

H_4 = heat to be supplied booster coil, Btu per hour.

M = weight of air to be introduced into the room or building, pounds per hour.

M_r = weight of recirculated air, pounds per hour.

M_b = weight of air by-passing washer, pounds per hour.

M_o = weight of air drawn in from outside, pounds per hour.

t_o = mean temperature of outside air, degrees Fahrenheit.

t = mean air temperature to be maintained in the room or building, degrees Fahrenheit.

t_1 = mean temperature of the air entering the reheater coil.

t_2 = mean temperature of the air leaving the reheater coil.

t_z = temperature loss in the duct system.

t_y = temperature of the air leaving the duct outlets.

t_x = average temperature of air entering tempering coil.

t_w = temperature of air entering washer.

0.24 = specific heat of air at constant pressure.

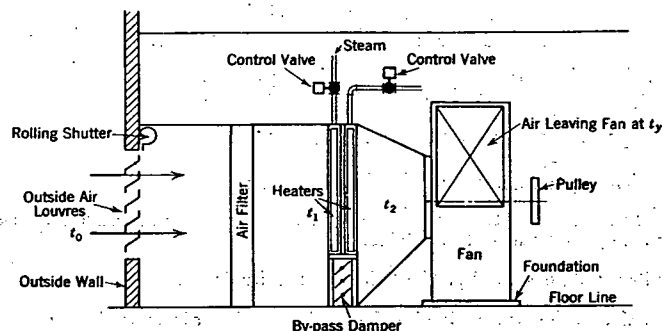


FIG. 3. HEATING UNIT AND FAN ARRANGED FOR OUTSIDE AIR CIRCULATION (Case A)

Case A. (Fig. 3) All of the air circulated to be drawn from outside the building, in which case $t_x = t_o$.

$$H_2 = 0.24 (t_1 - t_o) M_o \quad (10)$$

$$H_1 = 0.24 (t_2 - t_1) M_o \quad (11)$$

Example 6. The heat loss H for a certain factory building is 700,000 Btu per hour. The mean inside temperature t to be maintained is 65 F. The assumed outside air temperature t_o is 0 F; $t_2 = 0$, $t_y = t_2$ and is assumed to be 140 F. The temperature leaving the tempering coil is assumed to be 35 F. Required, H_1 and H_2 . From Equation 4,

$$M = \frac{700,000}{0.24 (140 - 65)} = 38,889 \text{ lb per hour.}$$

$$H_2 = 0.24 \times (35 - 0) \times 38,889 = 326,667 \text{ Btu per hour.}$$

$$H_1 = 0.24 \times (140 - 35) \times 38,889 = 980,003 \text{ Btu per hour.}$$

$$H_2 + H_1 = 326,667 + 980,003 = 1,306,670 \text{ Btu per hour.}$$

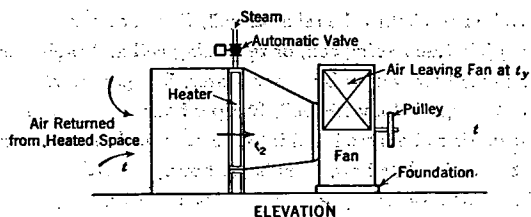


FIG. 4. ARRANGEMENT FOR RECIRCULATION (Case B)

Case B. (Fig. 4) All of the air is to be recirculated, in which case $t_1 = t$.

$$M_r = 38,889 \text{ lb}$$

$$M_1 = 0.24 (t_2 - t_1) M_r$$

$$H_1 = 0.24 (140 - 65) \times 38,889 = 700,000 \text{ Btu per hour.}$$

This example illustrates the saving in fuel consumption by the recirculation of the air. The heat to be supplied the apparatus is the same as that required for a direct system of heating and is equal to the heat loss of the building ($H_1 = H$), in the example 700,000 Btu per hour as compared with 1,306,670 for Case A.

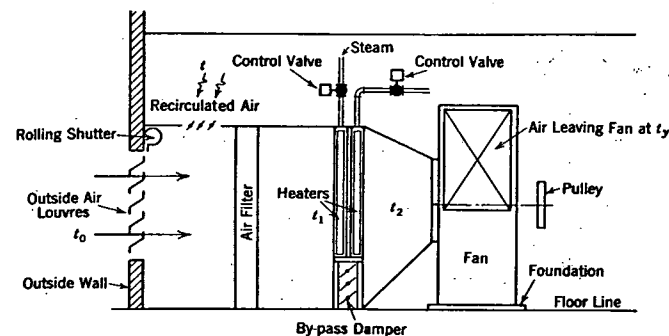


FIG. 5. COMBINATION OF RECIRCULATED AIR AND OUTSIDE AIR (Case C)

Case C. (Fig. 5) A portion of the air circulated is recirculated air and the remainder, as may be required for ventilating purposes, is drawn in from the outside. According to Equations 4 and 5,

$$M = M_o + M_r = \frac{H}{0.24 (t_y - t)}$$

The temperature of the resulting mixture of outside and recirculated air entering the tempering coil is:

$$t_x = \frac{M_o t_o + M_r t}{M} \quad (12)$$

Example 7. Assuming that a positive supply of outside air ($d_o = 0.0864$) is required for ventilation at the rate of 90,000 cu ft per hour in the preceding example, then $M_o = 0.0864 \times 90,000 = 7776$ lb per hour are required, measured at 65 F.

$$M_r = M - M_o = 38,889 - 7776 = 31,113 \text{ lb}$$

$$t_x = \frac{7776 \times 0 + 31,113 \times 65}{38,889} = 52 \text{ F}$$

$$H_1 = 38,889 \times 0.24 (140 - 52) = 821,336 \text{ Btu.}$$

This amount of work may be accomplished with one or more banks of heating units, that is, either a single reheater or a tempering coil and reheater.

The three preceding cases refer to installations in which conditioning the air to maintain certain relative humidity requirements does not enter into the problem, as for example, certain types of industrial installations. In practically all modern public buildings, theaters, schools, and in many industrial installations, the ventilating requirements include the provision for washing and humidifying the air delivered to the various rooms of the structure.

In the following cases it is assumed that in addition to maintaining a mean room temperature t , the heating and ventilating apparatus is required to maintain a constant relative humidity in the rooms.

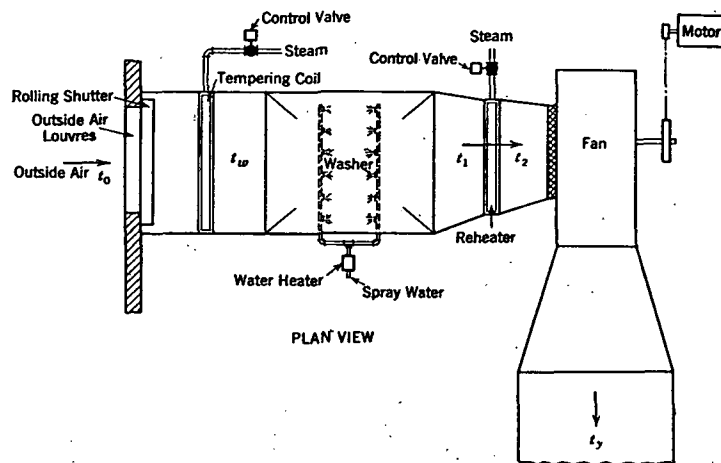


FIG. 6. OUTSIDE AIR CIRCULATED; CONSTANT RELATIVE HUMIDITY IN ROOM (Case D)

Case D. (Fig. 6) The maximum relative humidity that may be maintained within the building without the precipitation of moisture on single glazed sash when the outside temperature is 30 F is approximately 35 per cent. If the inside temperature t is 70 F, 35 per cent relative humidity corresponds to a dew-point temperature of 41 F. (See psychrometric chart.)

The installation shown in Fig. 6 contemplates the use of a tempering coil, an air washer provided with a water heater, and a reheater. The tempering coil, one section in depth, warms the incoming air to approximately 35 F to prevent freezing any of the spray water. The air passing through the spray chamber is saturated and leaves at a temperature of $t_1 = 41$ F.

The heat to be supplied the reheater is:

$$H_1 = 0.24 (t_1 - 41)M \text{ Btu per hour.}$$

The heat to be supplied the tempering coil is:

$$H_2 = 0.24 (35 - t_o)M \text{ Btu per hour.}$$

The amount of heat, per pound of air circulated, to be supplied the humidifying washer or humidifier is the difference between the heat content of the assumed dry air entering the washer at a temperature of $t_w = 35$ F and that of the leaving saturated air at $t_1 = 41$ F (Table 5, Chapter 1), or:

$$15.657 - 8.397 = 7.26 \text{ Btu per pound of dry air.}$$

The amount of heat required for the washer is:

$$H_3 = 7.26 M \text{ Btu per hour.}$$

The total amount of heat required by the apparatus is, therefore:

$$H_1 + H_2 + H_3 \text{ Btu per hour.}$$

If a washer having a *humidifying efficiency* of 67 per cent *without water heater* is employed it will be necessary to heat the outside air drawn into the apparatus by means of the tempering and preheater coils to such a temperature that the air in passing through

the water sprays will become partially saturated (adiabatically) having a moisture content per pound of air equal to saturated air at 41 F. If the incoming air is warmed to $t_w = 88$ F (requiring a two-section-depth heating unit) it will be cooled in the washer to 64 F, with a temperature drop of $88 - 64 = 24$ deg.

If the *humidifying efficiency* of the washer were 100 per cent, the air would become adiabatically saturated at 52 F after a temperature drop of $88 - 52 = 36$ F. The efficiency of the washer is, however, only 67 per cent, so that the actual temperature drop will be 0.67×36 deg or 24 deg, as used.

The heat to be supplied the reheater is in this case $H_1 = 0.24 (t_1 - 64) M$ Btu per hour, and the heat to be supplied to the tempering coil and preheater is $H_2 = 0.24 (88 - t_o) M$. The total heat required by the apparatus is $H_1 + H_2$, no heat being supplied to the washer.

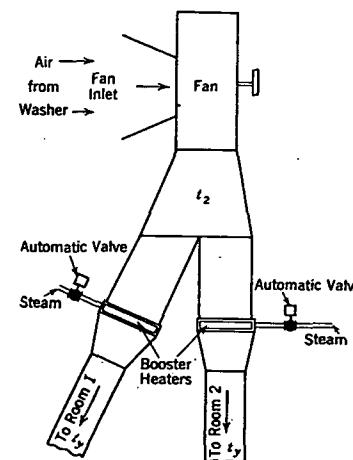


FIG. 7. OUTSIDE AIR CIRCULATED; CONSTANT TEMPERATURE AND RELATIVE HUMIDITY MAINTAINED IN EACH ROOM (Case E)

Case E. (Fig. 7) The temperature t_y will ordinarily be different for each room. With H and M fixed, $0.24 (t_y - t) M = H$, or

$$t_y = \frac{H}{0.24 M} + t$$

In order to provide the proper temperature for each room, a booster coil is generally installed in each supply duct near the outlet to control the outlet temperature t_y . The amount of steam supplied to these booster units is usually controlled automatically by individual thermostats. The heat required by the booster coils depends on the temperature range through which the air is heated and the quantity of air, or

$$H_4 = 0.24 (t_y - t_1 - t_2) M \quad (13)$$

Heat to be Supplied

The amount of heat to be supplied (H') is equal to the sum of the heat requirements of the various heating units and the water heater of the washer, if any, plus the allowance for piping tax. (See preceding Cases A to E.)

Grate Area, Boiler Selection

The required grate area may be determined by the following formula:

$$G = \frac{H}{F \times E \times C} \quad (14)$$

where

G = required grate area, square feet.

F = calorific value of fuel, Btu per pound.

C = combustion rate, pounds per square foot of grate per hour.

E = boiler and grate efficiency, per cent.

Example 8. Using the data in Example 5, and assuming coal having a calorific value of 12,000 Btu per pound, a combustion rate of 7 lb per square foot, and a performance efficiency of 0.60, and neglecting the piping tax,

$$G = \frac{1,306,670}{12,000 \times 0.60 \times 7} = 26 \text{ sq. ft.}$$

Weight of Condensate

The normal weight of condensate to be handled from central fan systems may be estimated by means of the following formula:

$$W = \frac{60 dQ \times 0.24 \times \Delta t}{h_{fg}} \quad (15)$$

where

W = weight of condensate, pounds per hour.

Q = total volume of air, cubic feet per minute.

Δt = temperature rise of air, degrees Fahrenheit.

h_{fg} = latent heat of steam in the system, Btu per pound.

Ducts and Outlets, Air Filters, Air Washers

The design of the duct system should be based on data contained in Chapter 20. Air washers and humidifiers are described in Chapter 11. For information on air filters, see Chapter 16.

Static Pressure

The total static pressure against which the system must operate may be found by summing up the static losses through the complete system from the outside air intake to the discharge outlets or nozzles. This means that the loss due to friction must be determined for each piece of apparatus involved. Most of these values may be obtained from manufacturers' data tables. For a simple system, the following static pressure drops may be assumed:

1. Outside air inlet, comprised of screen, louver and short duct, may have a loss of 0.2 in. of water.
2. A typical oil filter at rated capacity and velocity has a drop of 0.25 in. of water.
3. The loss of one row of a standard make tempering stack equals 0.09 in. water.
4. The loss of one row of a standard make preheater equals 0.10 in. water.
5. A standard humidifier at rated velocity may have a loss of about 0.35 in. water.
6. The loss through one row of a standard make reheater equals 0.12 in. water.
7. A fair assumption for duct losses on a simple system is 0.25 in. water.
8. The static pressure for a nozzle type outlet may be taken as 0.1 in. water.

The sum of these values equals $0.2 + 0.25 + 0.09 + 0.10 + 0.35 + 0.12 + 0.25 + 0.1 = 1.46$ in. which is the static pressure against which the system must operate.

Fans and Control

The selection of fans may be based on data contained in Chapter 17 and for motors in Chapter 42. Because centrifugal fans reach their maximum efficiency when working against the resistance offered by the average central fan heating system, they are well adapted to such systems and are generally used. Information on temperature control for central fan systems is given in Chapter 14.

PROBLEMS IN PRACTICE

1 ● Consider a blast heating system handling 10,000 cfm. The resistance to air flow offered by one coil arrangement is 0.9 in. of water and by another coil arrangement is 0.2 in. of water. The fan operates 4000 hours per year and the combined efficiency of motor and fan is 60 per cent. Determine the annual energy saving if the second coil is used.

Difference in system resistance = $0.9 - 0.2 = 0.7$ in. of water.

$$\text{Reduction in power input} = \frac{10,000 \times 0.7}{6356 \times 0.60} = 1.83 \text{ hp}$$

$$\text{Annual energy saving} = 1.83 \times 0.764 \times 4000 = 5480 \text{ kw. hr.}$$

2 ● What saving results from recirculating some of the room air and reducing the amount of outside air?

Because outside air must be heated to room temperature, reducing the amount of outside air produces a proportionate saving in heat or fuel.

3 ● What items make up the total heating load in a central fan heating system?

1. The net heat loss from the conditioned space.
2. The heat required for evaporation of water for humidification.
3. The heat required to raise the temperature of outside air to room temperature.
4. Heat losses from pipes and ducts.

4 ● Why is it necessary to determine the total static pressure of a central fan heating system?

To select a fan of maximum efficiency and to determine the power required to operate the fan.

5 ● A group of three drafting rooms, having a total volume of 27,000 cu ft, a transmission loss of 110,100 Btu per hour, and an infiltration loss of 34,200 Btu per hour on the basis of 0 F outdoors and 70 F room temperature, is to be heated by a recirculating hot blast heating system with air entering the rooms at 116 F. How many cubic feet per minute, measured at 70 F, will be required?

$$\text{Substitute in Equation 3. } H = 110,100 + 34,200 = 144,300 \text{ Btu per hour; } t_r = 116 \text{ F; } t = 70 \text{ F; } Q = \frac{144,300}{60 \times 0.07495 \times 0.24 (116 - 70)} = 2900 \text{ cfm.}$$

6 ● In the preceding question, if the hot air loses 4 F between heater and rooms, how many pounds of steam per hour at 1-lb gage will the heating sections condense?

Substitute in Equation 15. $Q = 2900$ cfm, from solution of Question 5; $\Delta t = 116 + 4 - 70 = 50$ F; $h_{fg} = 968$ Btu, from steam table in Chapter 1.

$$W = \frac{60 dQ \times 0.24 \times \Delta t}{h_{fg}} = \frac{60 \times 0.07495 \times 2900 \times 0.24 \times 50}{968} = 161.8 \text{ lb per hour.}$$

7 • The same rooms are converted to chemical laboratories, requiring the introduction of 12 changes of outside air, measured at 70 F, per hour to permit the exhaust fans connected to the chemical hoods to maintain only a slight negative pressure in the rooms. At what temperature must the air enter the rooms to maintain 70 F with 0 F outside?

Substitute in Equation 2. $H = 110,100 + 34,200 = 144,300$ Btu per hour; $Q = \frac{12}{60} \times 27,000 = 5400$ cfm; $t = 70$ F; $t_y = \frac{H}{60 dQ \times 0.24} + t = \frac{144,300}{60 \times 0.07495 \times 5400 \times 0.24} + 70 = 94.7$ F.

8 • In the preceding question, if the air drops 2 F between the heater and the rooms, how many pounds of steam per hour at 1-lb gage will the heating system condense?

Substitute in Equation 15. $Q = 5400$ cfm; $\Delta t = 94.7 + 2 = 96.7$ F, from solution of Question 7; $h_{fg} = 968$ Btu, from steam table in Chapter 1.

$$W = \frac{60 dQ \times 0.24 \times \Delta t}{h_{fg}} = \frac{60 \times 0.07495 \times 5400 \times 0.24 \times 96.7}{968} = 583 \text{ lb per hour.}$$

9 • The combination hot blast heating and ventilating system for the dining rooms of a hotel is to heat the rooms to 70 F with 0 F outside, and permit the exhaust fan from the adjoining kitchen to draw 5000 cfm from the dining rooms. The transmission losses from the dining rooms total 240,000 Btu per hour. The infiltration into the dining rooms amounts to 1000 cfm from outdoors and 1000 cfm from heater rooms. How many cubic feet per minute, measured at 70 F, must be supplied the dining rooms if the air enters at 112 F?

First find the infiltration loss by substituting in Equation 1.

$$t = 70 \text{ F; } t_o = 0; M_o = d \times Q = 0.07495 \times 60 \times 1000 = 4497 \text{ lb per hour. In this case } d \text{ and } Q \text{ are figured at } 70 \text{ F. } H_o = 0.24 (t - t_o) M_o = 0.24 (70 - 0) \times 4497 = 75,550 \text{ Btu per hour.}$$

Next by substituting in Equation 3, find the cubic feet per hour to be circulated. $H =$ sum of transmission and infiltration losses in room $= 240,000 + 75,550 = 315,550$ Btu per hour; $t_y = 112$ F; $t = 70$ F;

$$Q = \frac{H}{60 d \times 0.24 (t_y - t)} = \frac{315,500}{60 \times 0.07495 \times 0.24 (112 - 70)} = 6950 \text{ cfm.}$$

10 • In Question 9, 3000 cfm of outside air will be drawn in by the supply fan and 3950 cfm will be recirculated. What will be the output of the heating sections in Btu per hour if there is a loss of 2 F between the heaters and the room?

The average temperature of the mixture of outdoor and recirculated air entering the heater $= \frac{3000 \times 0 + 3950 \times 70}{6950} = 39.8$ F. Air leaves the heater at $112 + 2 = 114$ F.

Referring to Equation 15, $W \times h_{fg} =$ amount of heat required per hour $= 60 dQ \times 0.24 \times \Delta t = H$. $Q = 6950$ cfm; $\Delta t = 114 - 39.8 = 74.2$ F. $H = 60 \times 0.07495 \times 6950 \times 0.24 \times 74.2 = 557,000$ Btu per hour.

Chapter 23

MECHANICAL WARM AIR FURNACE SYSTEMS

Fan Furnaces, Fans and Motors, Elimination of Noise, Air Washers and Filters, Cooling Methods, Duct Design, Controls, Selecting the Furnace, Selecting the Fan, Humidity, Provision for Cooling System, Heavy Duty Fan Furnaces

MECHANICAL warm air or fan furnace heating systems, which are a special type of central fan systems, are particularly adapted to residences, small office buildings, stores, banks, schools, and churches. Circulation of air 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 its bottom, as in gravity systems described in Chapter 24. The advantages of mechanical systems, as compared with gravity systems are:

1. The furnace can be installed in a corner of the basement, leaving more basement room available for other purposes.
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 except in the furnace room.
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 air washers or filters, or both.
6. Some cooling effect in summer will result from the installation of a properly designed system.
7. 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.
8. 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.

Much of the equipment used in central fan systems is the subject matter of other chapters. It is the purpose of this chapter to discuss the co-ordinated design and to deal in detail only with problems not covered elsewhere which refer particularly to the whole problem of fan warm air furnace heating and air conditioning.

FAN FURNACES

Furnaces for mechanical warm air systems may be made of cast-iron, steel, or alloy. Cast-iron furnaces are usually made in sections and must be assembled and cemented or bolted together on the job. Steel furnaces are made with welded or riveted seams. The proper design of the furnace

depends largely on the kind of fuel to be burned. Accordingly, various manufacturers are making special units for coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as follows:

1. Coal Burning:
 - a. *Bituminous*—Large combustion space with easily accessible secondary radiator or flue travel.
 - b. *Anthracite or coke*—Large fire box capacity and liberal secondary heating surfaces.
2. Oil Burning:
 - a. Liberal combustion space.
 - b. Long fire travel and extensive heating surface.
3. Gas Burning:
 - a. Extensive heating surface.
 - b. Close contact between flame and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential heaters. A higher combustion rate is

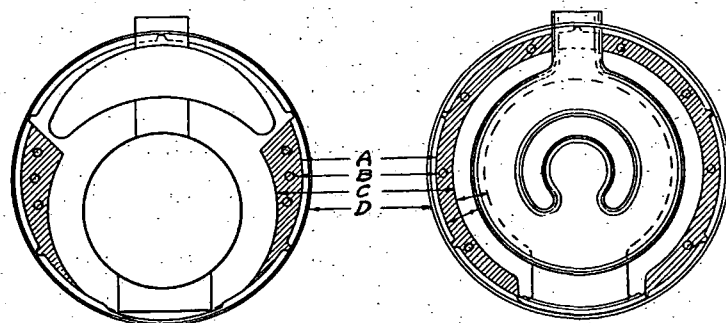


FIG. 1. USUAL METHOD OF BAFFLING ROUND CASINGS FOR FAN FURNACE WORK

A. Liner, 1 in. from casing. B. Hole to vent baffle.
C. Baffle, closed top and bottom. D. Outer casing.

permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

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. It is recommended that the system be designed for blow-through installations, so that the furnace shall be under external pressure in order to minimize the possibility of leakage of the products of combustion into the air circulating system.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run 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 any number of furnaces, using one or more fans.

Casings are usually constructed of galvanized iron, 26-gage or heavier, but they may also be constructed of brick. Galvanized iron casings should

be lined with black iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. Casings for commercial or heavy duty furnaces, if built of galvanized iron, should be insulated with fireproof insulating material at least 2-in. thick. It is generally believed that either brick or sheet metal casing should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

For furnace casings sized for gravity flow of air, where a fan is to be used, many manufacturers recommend the use of special baffles to restrict the free area within the casing and to force impingement of the air against the heating surfaces. The method of making these baffles for furnaces with top horse-shoe radiators and for furnaces with back crescent radiators is illustrated in Fig. 1.

Either square or round casings may be used. Where square casings are used, the corners must be baffled to reduce the net free area and to force impingement of air against the heating surfaces. Fig. 2 shows the usual method of baffling square furnace casings for fan-furnace work.

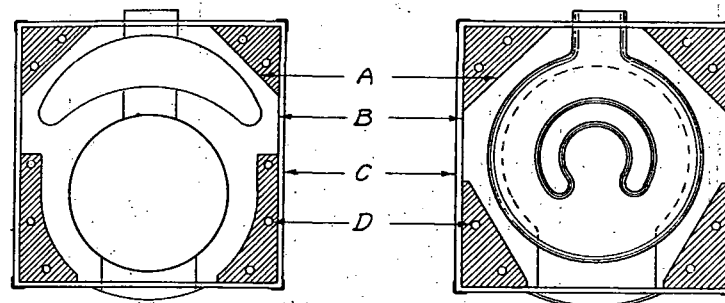


FIG. 2. METHOD OF BAFFLING SQUARE FURNACE CASING FOR FAN FURNACE WORK

A. Baffle, closed top and bottom. B. Liner, 1 in. from casing. C. Outer casing. D. Hole to vent baffle.

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. Fig. 3 illustrates a complete residence fan furnace installation showing location of fan, furnace, filters, plenum chamber and method of take-off of warm air pipe.

FANS AND MOTORS

Centrifugal type fans are most commonly used, and these may be equipped with either backward or forward curved blades. Low tip speed is desirable for the elimination of air noise, especially where forward curved blades are used. Motors may be mounted on the fan shaft or outside of the fan with belt connection. Multi-speed motors or pulleys

are desirable to provide a factor of safety and to allow for more rapid circulation for summer cooling.

For additional information on fans and motors, see Chapters 17 and 42.

NOISE ELIMINATION

Special attention must be given to the problem of noise elimination. The fan housing must not be directly connected with metal, either to the furnace casing or to the return air piping. It is common practice to use canvas strips in making these connections. Motors and their mountings

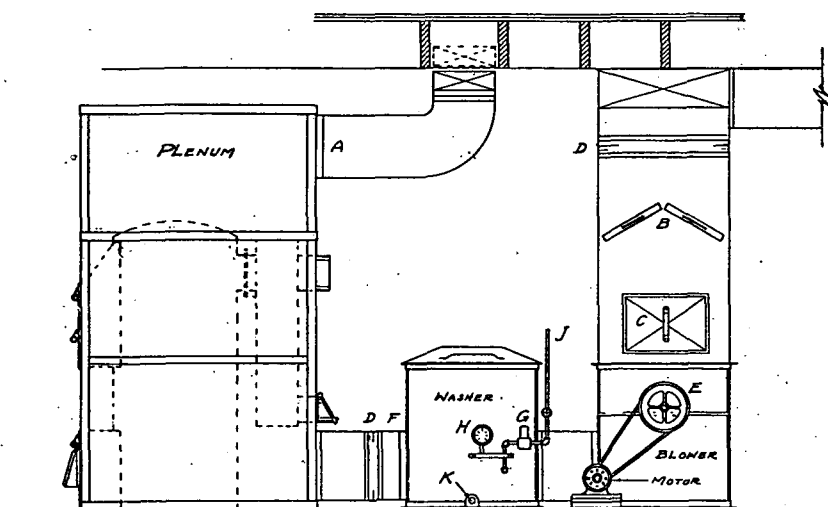


FIG. 3. COMPLETE RESIDENCE FAN FURNACE INSTALLATION SHOWING LOCATION OF FAN, FURNACE, FILTERS, PLENUM CHAMBER AND METHOD OF TAKE-OFF OF WARM AIR PIPE

- | | |
|---------------------------|--------------------|
| A. Transition fitting. | F. Eliminator. |
| B. Filters. | G. Solenoid valve. |
| C. Capped opening. | H. Pressure gage. |
| D. Canvas connection. | J. Water supply. |
| E. Pulley—3 diam. V-type. | K. Drain. |

must be carefully chosen for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with fan housing. The installation of a fan directly under a cold air grille is not recommended on account of the noise objection. See also Chapter 18.

AIR WASHERS AND FILTERS

Washers for residence systems may be provided in separate housings to be installed on the inlet or outlet side of the fan, or they may be integral with the fan construction. They operate at water pressures of from 10 to 30 lb and use two or more spray nozzles for washing and humidification. The sprays should be adjusted to completely cover the air passages.

Washers 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. Washers used in connection with commercial or heavy duty plants should be a regulation type of commercial washer.

There are many satisfactory types of filters on the market. These include dry filters, viscous filters, oil filters and other types, some of which must be cleaned, some of which must be cleaned and recharged with oil, and some of which are inexpensive and may be discarded when they become dirty, and replaced with new ones.

The resistance of a filter must be considered in the design of the system since the resistance rises rapidly as the filter becomes dirty, thus impairing the heating efficiency of the furnace, in fact, endangering the life of the furnace itself. Manufacturers' ratings of filters must be carefully regarded, and ample filter area must be provided. Filters must be replaced or cleaned when dirty. See also Chapter 16.

COOLING METHODS

Some cooling may be obtained under certain conditions by the use of basement air. A more positive cooling effect may be obtained through air washers where the temperature of the water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the 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 is necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with the fan and duct system for the heating installation, and to cool the building by this method, 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 9 and 10.

Study of these problems sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating and Air Conditioning Association* is in progress at the University of Illinois. The following conclusions may be drawn from the studies thus far completed, subject to the limitations of the conditions under which the tests were run¹:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hours on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

¹See A.S.H.V.E. research paper entitled Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933). Study of Unit Room Coolers in the Research Residence, by A. P. Kratz, M. K. Fahnestock, and S. Konzo (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November, 1934).

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.

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 results of the tests suggest the use of a fan at night either to provide more comfortable conditions during the following day without provision for cooling, or to reduce the load required for cooling during the following day. Experience has shown that the volume of air required for cooling, depending upon the climate and the construction of the building, must usually be from 50 to 100 per cent greater than that required for heating. If the size of the fan is based upon the summer requirement, its output may be reduced sufficiently to meet winter heating needs.

7. Attic exhaust fans are becoming popular adjuncts for night duty. (See Chapter 12.)

DUCT DESIGN

The ducts may be either round or rectangular. Rectangular ducts should be as nearly square as possible; the width should not be greater than four times the breadth. The radii of elbows should 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.

The ducts or piping may be designed either as a trunk line system or as a system of individual ducts from the furnace casing to each register. The engineering problems incident to the design of a trunk line system are somewhat more difficult than for the individual duct system. The trunk line system is generally a tailor-made job, whereas the individual duct system with which either round or square ducts may be used may frequently be assembled from stock materials and thus installed at a considerable saving. Individual ducts may frequently be grouped to simulate a trunk duct system in appearance. The design of ducts for air flow is described in Chapter 20.

Dampers

Suitable dampers are essential to any trunk or individual duct system, as it is virtually impossible to so lay out a system that it will be absolutely in balance without the use of dampers. 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. See Figs. 1 and 2, Chapter 20.

Three types of dampers are commonly used in trunk and individual duct systems. *Volume dampers* are used to completely cut off or reduce the flow through pipes. (See *A* and *B*, Fig. 4.) *Splitter dampers* are used where a branch is taken off from a main trunk. (See *C*, Fig. 4.) *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. (See *D*, Fig. 4.) It is essential that a damper be provided for each main or duct branch. A positive locking device should be used with each type of damper.

Supply and Return Air Registers

Supply registers located in the floor are effective, but as they require frequent attention to keep them clean they should be avoided where another effective register location can be found. Unless registers located in the baseboard are well proportioned and designed to harmonize with

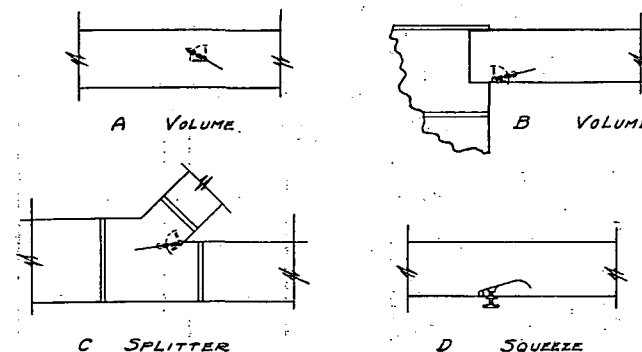


FIG. 4. THREE TYPES OF DAMPERS COMMONLY USED FOR TRUNK AND INDIVIDUAL DUCT SYSTEMS

the trim, they may be unsightly. Registers which are located in side walls above the baseboard or in the ceiling should be of an effective air-diffusing type. All registers should be sealed against leakage around the borders or margins.

Velocities through registers may be reduced by the use of registers larger than the connecting pipes. Some suggestions for equalizing velocities over the face area of the register by means of diffusers are illustrated

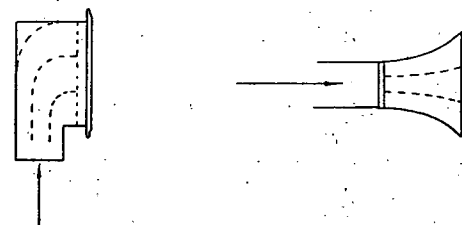


FIG. 5. DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

in Fig. 5. Merely to use a larger register may not result in materially reduced velocities unless such diffusers are used.

Care should be exercised in making the connection between the supply register and its box to prevent streaking of the wall. All warm air registers should be equipped with dampers or, better, with diffuser dampers which may be used to direct air currents in such a way that they will not be objectionable. (See Chapter 19.)

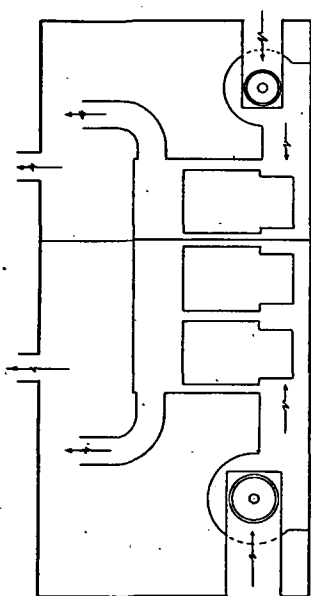


FIG. 7. TWO BATTERIES OF HEATERS AND FANS FOR INDEPENDENT SERVICE USING OUTSIDE AIR AND EXHAUSTING TO ATMOSPHERE

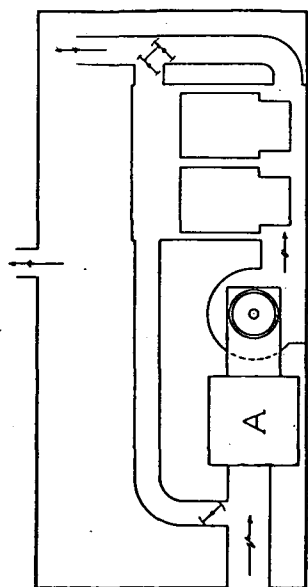


FIG. 9. HEATER ARRANGED FOR USE OF AIR WASHER OR FILTER (A) WITH HEATED AIR TO MIX WITH OUTSIDE AIR FOR TEMPERING, SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR AND EXHAUST TO ATMOSPHERE

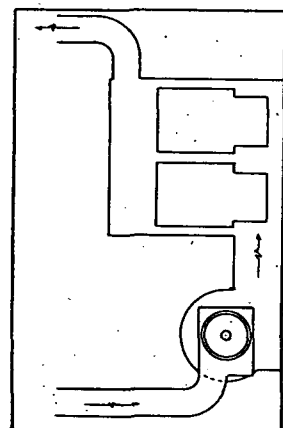


FIG. 6. HEATER ARRANGED FOR COMPLETE RECIRCULATION OF AIR

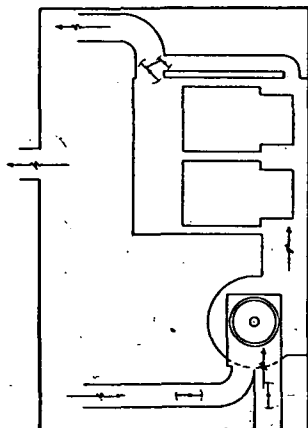


FIG. 8. HEATER ARRANGED FOR PARTIAL RECIRCULATION, ALSO SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR CHAMBERS, AND PARTIAL EXHAUST TO ATMOSPHERE

CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in the system can be largely eliminated through proper care in the planning and installation of the control system. The essential requirements of the control are:

1. To keep the fire burning when using solid fuel regardless of the weather.
2. To avoid excessive bonnet temperatures with resultant radiant heat losses into the basement.
3. To avoid the overheating of certain rooms through gravity action during off periods of blower operation.
4. To have a sufficient supply of heat available at all times to avoid lag when the room thermostat calls for heat.
5. To prevent cold air delivery when heat supply is insufficient.
6. To avoid heat loss through the chimney by keeping stack temperatures low.
7. To provide quick response to the thermostat, with protection against overrun.
8. To provide for humidity control.
9. To provide a means of summer control of cooling.
10. To protect against fire hazards.

The following controls are desirable:

1. A *thermostat* located at a point where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. This location would be near an outside wall but not upon it, in a sun room, or in a room with some unusual exposure. The thermostat, of course, should not be located 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 or register.
2. A *furnacestat* located in the bonnet to permit blower operation only between the temperatures of 100 F and 150 F. In certain extreme cases it may be necessary, or weather conditions may make it advisable, to adjust the high limit to a higher temperature than that given. Another location sometimes used for the furnacestat is in the main duct near the frame opening from the bonnet.
3. A *protective limit control* located in the bonnet to shut down the system independently of the thermostat if the bonnet temperature exceeds 225 F.
4. On oil and gas burner installations, a control is usually included which will shut down the system if the fire goes out or if there is a failure of the ignition system.
5. A *humidistat* to regulate the moisture supplied to the rooms.
6. 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.

While it is usually all right to start and stop the fan in residential control work and in auditoriums and other places where many people may gather, the fan should as a rule be allowed to run continuously and the control should be cared for in other ways.

SELECTING THE FURNACE

The following formula may be used to compute the grate area of a residence furnace, assuming a ratio of heating surface to grate area of 20 to 1:

$$G = \frac{H}{F \times C \times E} \quad (1)$$

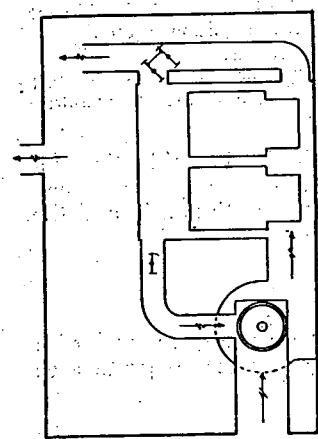


FIG. 10. HEATER ARRANGED TO USE COMPLETE OUTSIDE AIR WITH TEMPERED AIR CHAMBER ABOVE WARM AIR CHAMBER SHOWING MIXING DAMPER TO TEMPER AIR, ALSO MIXING DAMPER TO DUCT FROM WARM AND TEMPERED AIR CHAMBERS

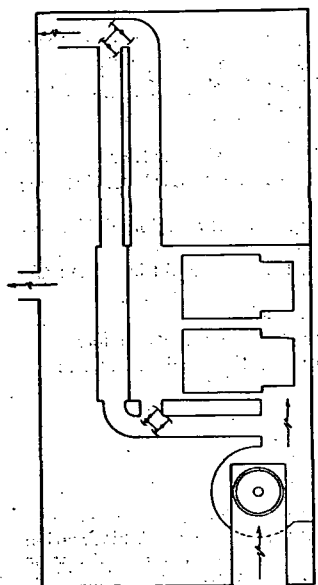


FIG. 11. HEATER ARRANGED TO USE OUTSIDE AIR, WITH TEMPERED AIR CHAMBER ABOVE WARM AIR CHAMBER SHOWING DOUBLE DUCT DISTRIBUTING SYSTEM AND MIXING DAMPERS AT BASE OF RISERS

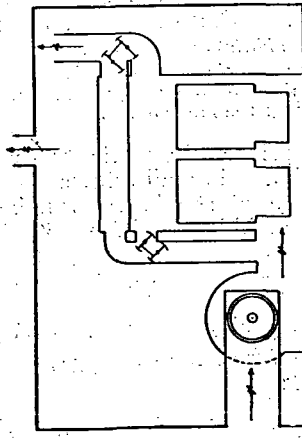


FIG. 12. HEATER ARRANGED FOR OUTSIDE AIR WITH WARM AIR CONNECTION TO FAN INLET TO TEMPER AIR, ALSO MIXING DAMPERS FROM WARM AIR AND TEMPERED AIR CHAMBER

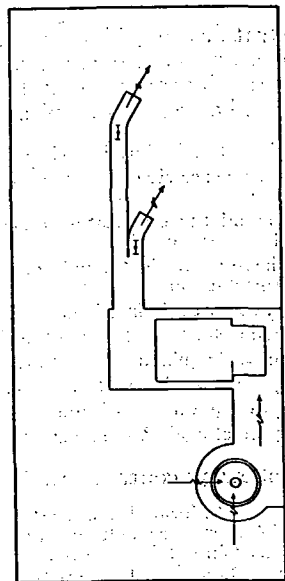


FIG. 13. HEATER ARRANGED FOR RECIRCULATION WITH TRUNK LINE DISTRIBUTING SYSTEM SHOWING TEMPERATURE CONTROL BY AIR VOLUME AT OUTLET, OR DIRECT ON HEATER

where

- G = required grate area, square feet.
- H = total heat loss from building, Btu per hour.
- F = calorific value of coal, Btu per pound.
- C = combustion rate in pounds of fuel per square foot of grate per hour.
- E = furnace efficiency based on heat available at register faces.

In practice it is customary to use the following constants:

- F = 13,000 (For specific values, see Table 1, Chapter 27).
- C = 5 to 10 lb (Use 8 lb as maximum in residence work).
- E = 55 per cent to 65 per cent depending on fuel burned. Lower efficiency must be used with highly volatile solid fuel.

Where ratio of heating surface to grate area is less or greater than 20 to 1, deduct or add 2 per cent from or to rating of furnace for each unit decrease or increase in ratio, as the case may be. The foregoing procedure for determining the size of the furnace to be used applies to continuously heated buildings.

Although intermittently heated buildings usually have their heat losses computed according to the standard rules for determining such losses, these rules do not take into account the heat which will be absorbed by the cold material of the building after the air is raised in temperature. This heat absorption must be added to the normal heat loss of the building to determine the load which the heating plant must carry through the warming-up process. It is customary to increase the normal heat loss figure by from 50 to 150 per cent depending upon the heat capacity of the construction material, the higher percentage applying to materials of high heat capacity such as concrete and brick. Fan furnace systems are well adapted for heating intermittently heated buildings as these systems do not require the warming of intermediate piping, radiators, or convectors, the generation of steam, or the heating of hot water.

Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected. Use the manufacturers' Btu ratings of furnaces designed for exclusive use with oil, and select a burner with liberal excess capacity.

The selection of the proper size gas furnace for a constantly heated building can be easily made by using the following *American Gas Association* formula:

$$R = \frac{H}{0.9} \quad (2)$$

where

- H = total heat loss from building in Btu per hour.
- R = official A.G.A. output rating of the furnace in Btu per hour.

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output. The proper sizing may be done by means of the following formula:

$$I = 1.56H \quad (3)$$

where

$I = \text{Btu per hour input.}$

The factor 1.56 is the multiplier necessary to care for a 10 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

SELECTING THE FAN

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the following items:

1. The frictional resistance of a warm air trunk or leader.
2. The frictional resistance of a return air trunk or duct.
3. The resistance to the flow of total volume of air through the furnace casing or hood, which is usually considered from 0.10 to 0.15 in. of water.
4. The frictional resistance through any other accessories, such as washers or filters.
5. A factor of safety of 10 per cent of the resistance calculated above.

HUMIDITY

Mechanical warm air systems offer an excellent means of proportioning and distributing moisture-bearing air; consequently, during the winter months humidifiers may be employed to deliver water vapor to the fan-driven air stream in proper amounts to produce a more humid atmosphere, with increased comfort for people and increased life for household furnishings. Temperatures and relative humidities should be governed within the limits of the generally accepted standards. See Chapters 3 and 11 for more detailed information on this point.

In earlier types of furnaces, water evaporating pans were usually placed in the cool portions of the air stream, but modern types usually locate them 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. Without the addition of this heat, termed the latent heat of evaporation, water injected into the air will be carried along in the form of tiny globules until it falls out of the stream or is deposited upon some surface.

Furthermore, when dry air is in contact with water for a sufficient length of time without the presence of a sizable body of water or a source other than air from which this latent heat of evaporation can be taken, such heat is supplied from the air. There is, therefore, 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. (See Chapter 11.)

Humidification for Residences

The principles underlying humidity requirements and limitations for residences are summarized in *University of Illinois Bulletin No. 48*², as follows:

²See Humidification for Residences, by A. P. Kratz (*University of Illinois, Bulletin No. 48*).

1. Optimum comfort is the most tangible criterion for determining the air conditions within a residence.

2. An effective temperature of 65 deg³ represents the optimum comfort for the majority of people. Under the conditions in the average residence a dry-bulb temperature of 69.5 F with relative humidity of 40 per cent is the most practical for the attainment of 65-deg effective temperature.

3. Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air leakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.

4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight-fitting storm sash or the equivalent is installed.

5. The problems of humidity requirements and limitations cannot be separated from considerations of good building construction, and the latter should receive serious attention in the installation of humidifying apparatus.

The following conclusions were drawn from the experimental results reported in the aforementioned bulletin:

1. None of the types of warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except only in moderately cold weather.

2. The water pans used in the radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

PROVISION FOR COOLING SYSTEM

If the system is to be used for cooling, the following provisions should be made:

1. Where cooling is to be secured through air circulation only:
 - a. Provide for an increase of 50 to 100 per cent in fan capacity through multi-speed pulleys or other means.
 - b. If basement air or outside night air is to be used, provide suitable basement opening in duct system, or outdoor air intake.
2. Where water below 55 F or artificial refrigeration or ice is to be used:
 - a. Provide outside air duct for circulation of cool night air for economy.
 - b. Make provision in return duct system for cooling unit.
 - c. Make provision for control of the fan speed, during winter operation, to give a sufficient and draftless air movement.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings are available in sizes ranging from 400,000 to 3,000,000 Btu per hour per unit. Heavy duty heaters may be arranged in combinations of one or more units in a battery. A few possible arrangements are shown in Figs. 6 to 13, inclusive.

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

³66 deg is the optimum winter effective temperature recommended by the A.S.H.V.E. Committee on Ventilation Standards. See Chapter 3.

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 on 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 22. Ducts are designed by the method outlined in Chapter 20.

PROBLEMS IN PRACTICE

1 ● A residence furnace, having a ratio of heating surface to grate area in excess of 20 to 1, is to be selected to heat a house which has a computed load of 225,000 Btu per hour. If coal having a calorific value of 13,000 Btu per lb is to be burned, if the furnace will burn 8 lb of coal per square foot of grate per hour, and if the furnace efficiency is 60 per cent, determine the square feet of grate area necessary in the furnace to be selected.

Substituting in Equation 1:

$$G = \frac{225,000}{13,000 \times 0.60 \times 8} = 3.6 \text{ sq ft of grate area.}$$

A furnace having at least 3.6 sq ft of grate area should therefore be selected.

2 ● Why should secondary surface be designed for easy cleaning?

If the combustion is not perfect, soot is formed immediately above the fire and is apt to form a deposit on the secondary surface from which it should be removed. If the secondary surface is so designed that there are horizontal passages, fine gray ash will settle out in these to form an insulation between the hot gases of combustion and the metal of the furnace; consequently, these should be readily cleaned. If the passages are vertical they are largely self-cleaning of ash, but provision should be made for easy and thorough cleaning of the collection chamber below them.

3 ● Why is baffling inside the casing necessary on fan systems?

Because the movement of air is independent of its temperature, air must be guided by baffles of one form or another to bring it in contact with the hot surfaces so it will not pass through the casing unheated. On the other hand, if the air is held against a hot surface too long it might become overheated, for the average register temperature on a fan system should not exceed 120 F.

4 ● What practical points should be observed in designing a fan system in order to eliminate noise?

- Use a large fan so it can be run at slow speed.
- Set the fan and motor on a solid foundation.
- Insulate the fan and motor from the foundation with rubber, cork, or other springy material according to the principles given in Chapter 18, provided, of course, that such insulation is of value.
- See that the air velocity is not too high in the ducts. Properly designed splitters in the elbows will avoid high velocities at the turns in cases where the velocity through the ducts themselves is not too high.
- Use canvas connections between the ducts and any running equipment.
- Be sure the ducts have a relatively smooth interior and are rigid.

Chapter 24

GRAVITY WARM AIR FURNACE SYSTEMS

Procedure for Design, Estimating Heating Requirements, Sizes of Leader Pipes, Proportioning Wall Stacks, Register Sizes, Recirculating Ducts and Grilles, Return Connection to Furnace, Furnace Capacity, Examples, Booster Fans

WARM air heating systems of the gravity type are described in this chapter¹, and those of the mechanical type are described in Chapter 23. In the gravity type, 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. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater, enclosed in a casing of sheet metal or brick, 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.

The air supply to the furnace may be taken (1) entirely from inside the building through one or more recirculating ducts, (2) entirely from outside the building, in which case no air is recirculated, or (3) through a combination of the inside and the outside air supply systems.

PROCEDURE FOR DESIGN

The design of a furnace heating system involves the determination of the following items:

- Heat loss in Btu from each room in the building.
- Area and diameter in inches of warm-air pipes in basement (known as leaders).
- Area and dimensions in inches of vertical pipes (known as wall stacks).
- Free and gross area and dimensions in inches of warm-air registers.
- Area and dimensions of recirculating or outside air ducts, in inches.
- Free and gross area and dimensions in inches of recirculating registers.

¹All figures and much of the engineering data which follow are from *Bulletins No. 141, 188 and 189, Warm Air Furnaces and Heating Systems, Part II*, by Professor A. C. Willard, A. P. Kratz, and V. S. Day, Engineering Experiment Station, University of Illinois.

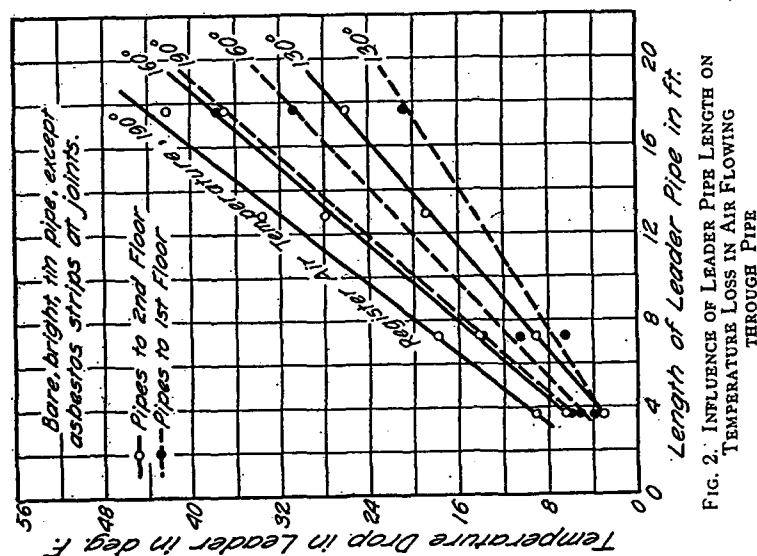


FIG. 2. INFLUENCE OF LEADER PIPE LENGTH ON TEMPERATURE LOSS IN AIR FLOWING THROUGH PIPE

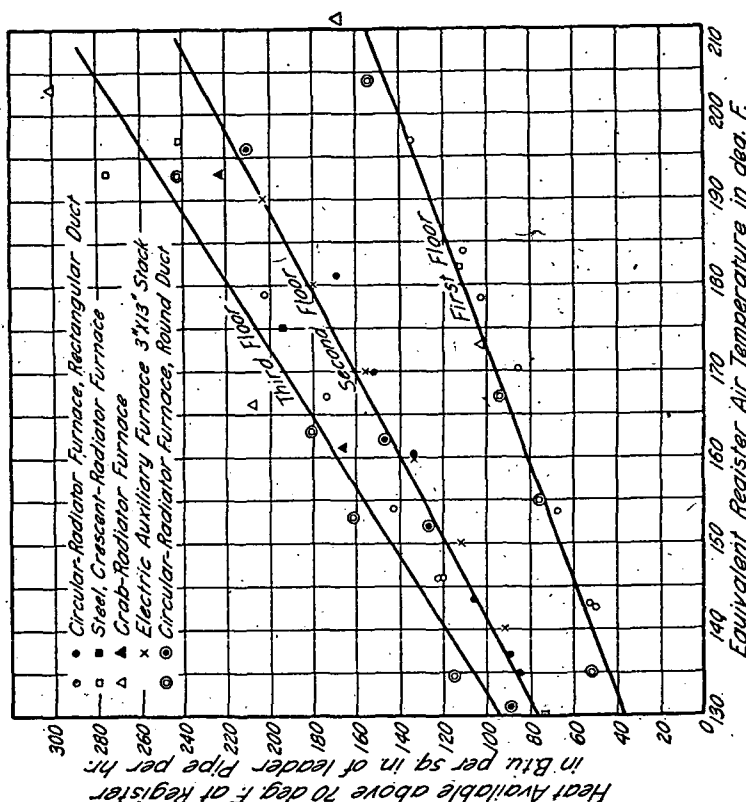


FIG. 1. VALUE OF SQUARE INCH OF LEADER PIPE AREA FOR FIRST, SECOND, AND THIRD FLOORS FOR SIMPLE SYSTEM HAVING LEADERS 8 FT IN LENGTH

7. Size of furnace necessary to supply the warm air required to overcome the heat loss from the building. This size should include square inches of leader pipe area which the furnace must supply. It is also desirable to call for a minimum bottom fire-pot diameter in inches, which is the nominal grate diameter.

8. Area and dimensions in inches of chimney and smoke pipe. If an unlined chimney is to be used, that fact should be made clear.

The heat loss calculations should be made in accordance with the procedure outlined in Chapter 7, taking into consideration the transmission losses as well as the infiltration losses.

SIZES OF LEADER PIPES

In a gravity circulating warm-air furnace system the size of the leader to a given room depends upon the temperature of the warm air entering the room at the register. A reasonable air temperature at the registers must, therefore, be chosen before the system can be designed. The *National Warm Air Heating and Air Conditioning Association* has approved an air temperature of 175 F at the registers as satisfactory for design purposes. At this temperature, the heat-carrying capacity (heat available above 70 F) per square inch of leader pipe per hour for first, second or third floors is shown by Fig. 1 at 175 F to be 105, 170 and 208 Btu, respectively. For average calculations, the values 111, 166 and 200 will simplify the work and may be satisfactorily substituted for these heat-carrying capacities. If H represents the total heat to be supplied any room, the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{111} = \text{approximately } 0.009H \quad (1)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \quad (2)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{200} = \text{approximately } 0.005H \quad (3)$$

In designing for a lower warm-air register temperature, say 160 F, the factors 111, 166 and 200 become 80, 140 and 166 (Fig. 1 at 160 F), and the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{80} = \text{approximately } 0.012H \quad (4)$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{140} = \text{approximately } 0.007H \quad (5)$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \quad (6)$$

These equations are applicable to straight leaders from 6 to 8 ft in length. Longer leaders must be very thoroughly covered or else the vertical stacks must be increased in area as discussed under wall stacks. If some provision is not made for these longer leaders, the air temperature may be much lower than anticipated and the room will not be properly heated.

While Fig. 1 takes care of the drop in temperature in straight leaders up to 8 ft in length connected to stacks having about 75 per cent of the

area of the leader, the designer must make allowances for all other conditions. The temperature drop in leaders of various lengths at three different register temperatures is shown in Fig. 2, and should be used to obtain new register temperatures, lower than 175 F, on which to base selections from the curves of Fig. 1, and thereby new constants for Equations 1, 2 and 3.

Leader sizes should in general be not less than those obtained by Equations 1 to 3 nor should leaders less than 8 in. in diameter be used. It is not considered good commercial practice to specify diameters except

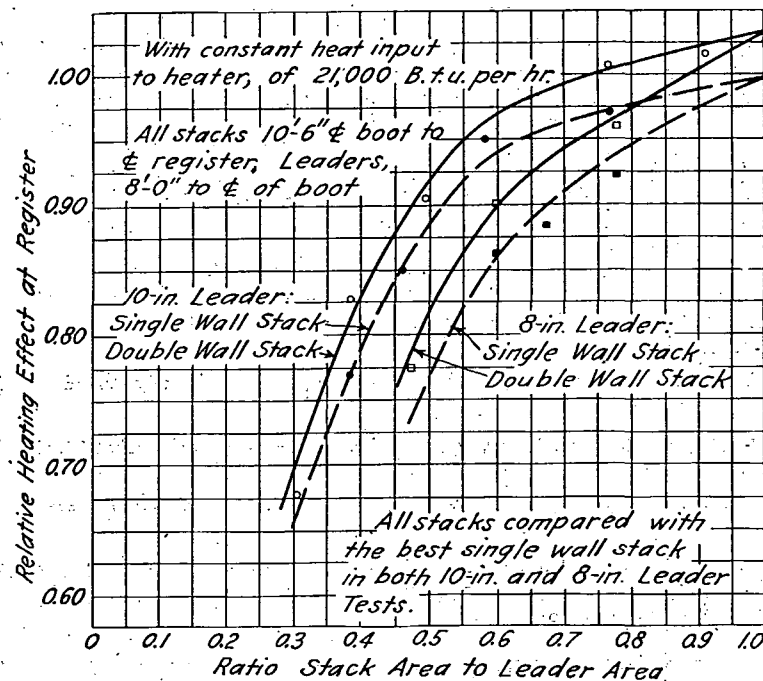


FIG. 3. RELATIVE HEATING EFFECT OF STACKS AT CONSTANT HEAT INPUT TO FURNACE

Note.—Exterior surface of all ducts is bright tin except at joints where asbestos sealing strips are used.

in whole inches. The tops of all leaders should be at the same elevation as they leave the furnace bonnet, and from this point there should be a uniform up-grade of 1 in. per foot of run in all cases. Leaders over 12 ft in length are to be avoided or should receive very special attention.

PROPORTIONING WALL STACKS

The wall stack for an upper floor should be made not less than 70 per cent of the area of the leader which has been selected from Fig. 1. So long as the leader is short and straight as was the case for Fig. 1,

such a practice is probably justified, since the loss (Fig. 3) in capacity occasioned by the smaller stack is not very serious for stacks having areas in excess of 70 per cent of the leader area. For leaders over 8 ft in length or for leaders which are not straight, the ratio of stack area to leader area should be greater than 70 per cent in order to offset the greater temperature losses (Fig. 2) in the longer leader. In gravity circulating systems, this stack to leader area ratio is a very important consideration. Specific data for a great variety of cases are presented in Figs. 4 and 5 and the designer should check the stack to leader combinations with the nearest comparable case as shown in these figures. Any second-floor stack supplying heat to a room whose heat loss is 9,000 Btu or more (see Figs. 4 and 5 which show that high temperatures are necessary if rooms of more than 9,000 Btu requirement are heated by one stack each in 4-in. studding) should be run within 6-in. studded walls or should have multiple stacks. Stack sections, wherever possible, should be changed from the thin rectangular to the more nearly square shape.

REGISTER SIZES

The registers used for discharging warm air into the rooms should have free or net area not less than the area of the leader in the same run of piping. The free area should be at least 70 per cent of the gross area of the register. No upper-floor register should be wider horizontally than the wall stack, and it should be placed either in the baseboard or side wall, if this can be done without the use of offsets. First-floor registers may be of the baseboard or floor type, with the former location preferred.

RECIRCULATING DUCTS AND GRILLES

The ducts through which air is returned to the furnace should be designed to minimize friction and turbulence. They should be of ample area, in excess of the total area of warm-air pipes, and at all points where the air stream must change direction or shape, streamline fittings should be employed. Horizontal ducts should pitch at least $\frac{1}{2}$ in. per foot upward from the furnace.

The recirculating grilles (or registers) should have a free area at least equal to the ducts to which they connect, and their free area should never be less than 50 per cent of their gross area.

The location and number of return grilles will depend on the size, details and exposure of the house. Small compactly built houses may frequently be adequately served by a single return effectively placed in a central hall. More often it is desirable to have two or more returns, provided, however, that in two-story residences one return must be placed to effectively receive the cold air returning by way of the stairs.

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 should be brought as close to the furnace as possible, but if the room has a bay window, French doors, or other large sources of cooling or leakage of cold air, the grille should be placed close by, so as to collect the cool air and prevent drafts. When long ducts of this type are employed they must be made

oversize and favored in every way. This precaution is particularly important when long ducts and short ducts are used in the same system.

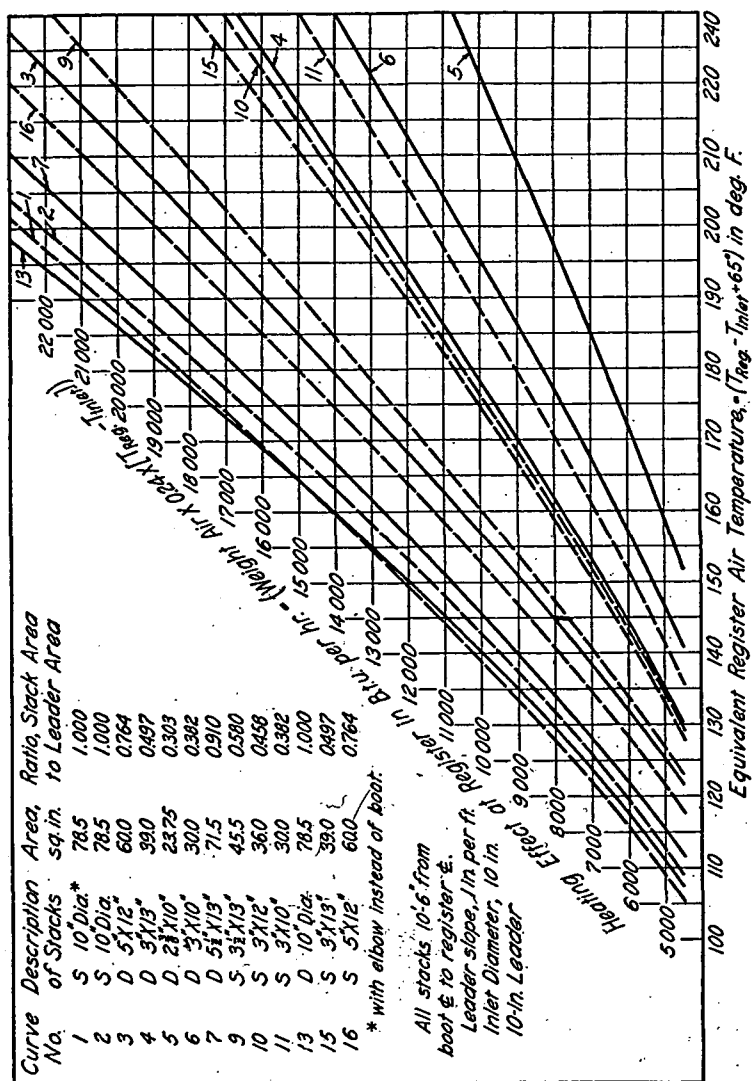


FIG. 4. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 10-IN. LEADER

The long ducts must be oversize, if they are to operate satisfactorily in parallel with short ducts.

Return ducts from upstairs rooms may be necessary in apartments or other spaces closed off or badly exposed. Metal linings are advisable in such ducts. It is important that these ducts be free from unnecessary

friction and turbulence, and that they be located to prevent preheating of the air before it reaches the furnace.

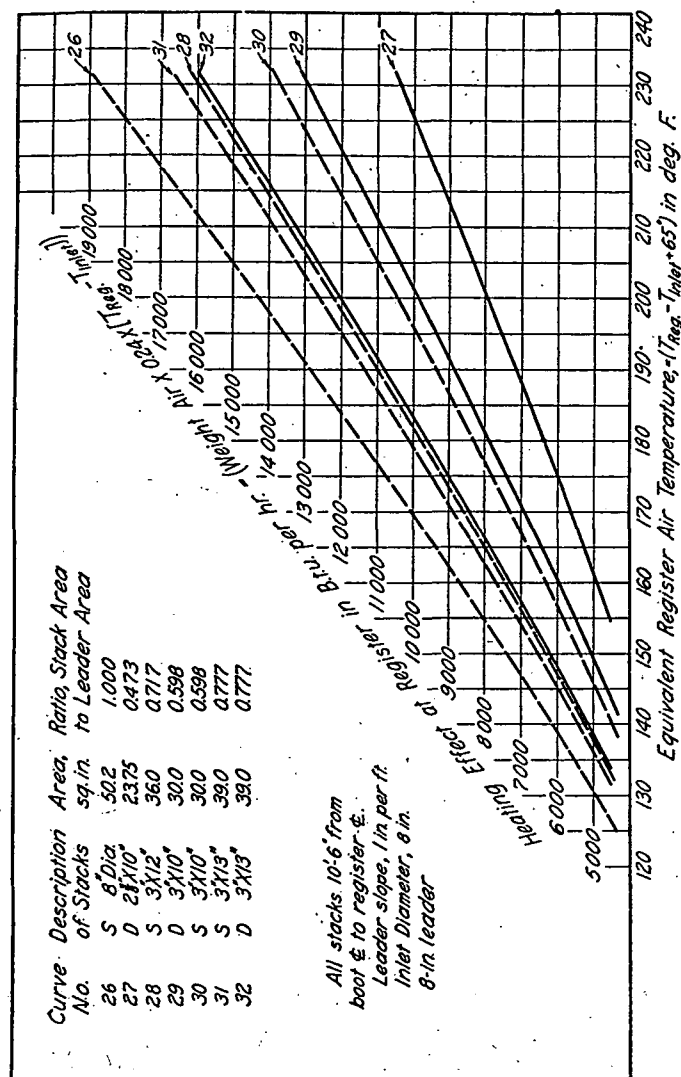


FIG. 5. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 8-IN. LEADER

Return Connection to Furnace

Circulation is accelerated if the drop to the furnace is through a round inclined pipe with, say, two 45-deg elbows rather than through a vertical drop and two 90-deg elbows. The top of the shoe should never enter

the casing above the level of the grate in the furnace. To accomplish this the shoe must be wide.

Tests of six different systems of cold air returns, Fig. 6, made at the University of Illinois², resulted in the following conclusions:

1. In general, somewhat better room temperature conditions may be obtained by returning the air from positions near the cold walls.
2. Friction and turbulence in elaborate return duct systems retard the flow of air, and may seriously reduce furnace efficiency, and lessen the advantages of such a design.
3. The cross-sectional duct area is not the only measure of effectiveness. Friction and turbulence may operate to make the air flow out of all proportion to the various duct areas.

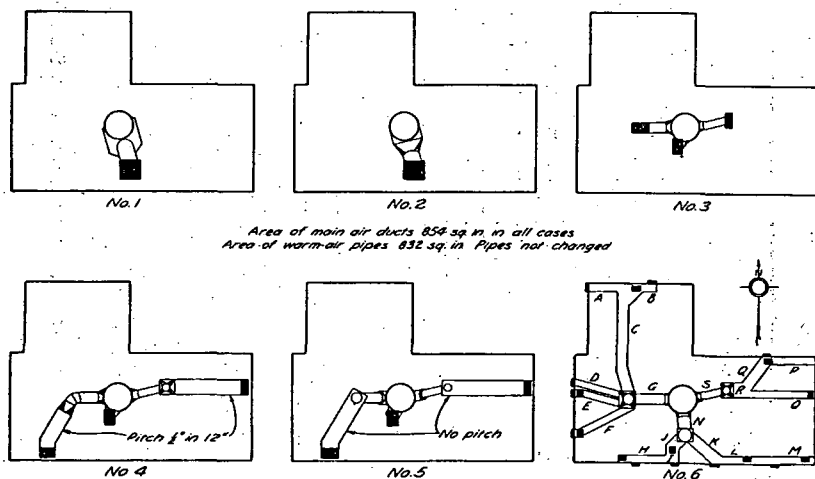


FIG. 6. ARRANGEMENT OF COLD AIR RETURNS FOR SIX INSTALLATIONS

FURNACE CAPACITY

The size of furnace should, of course, be such as will provide the necessary air heating capacity, usually expressed in square inches of leader pipe area, and at the same time provide a grate of the proper area to burn the necessary fuel at a reasonable chimney draft. The total leader-pipe area required is easily obtained by finding the sum of the leader pipe areas as already designated.

The grate area will depend on several factors of which four are very important. First of all, the air temperature at the register for which the plant has been designed must be determined. Usually, this temperature is taken as 175 F. Second in importance is the combustion rate, which must always correspond with the register air temperature, as is shown by reference to a set of typical furnace performance curves (Fig. 7) for a cast-iron circular radiator furnace with a 23-in. diameter grate and 50-in. diameter casing. The conditions shown on these curves which seem to

approximate nearest to the 175 F register warm-air temperature are: combustion rate, 7 lb; warm-air register temperature, 173 F; efficiency of the furnace, 58.5 per cent. The third factor is efficiency, which, in turn, is a function of the combustion rate varying with it as shown by the efficiency curve of Fig. 7. The fourth factor is the heat value per pound of fuel burned, which was 12,790 Btu. This is not shown on the curves since it was constant for all combustion rates.

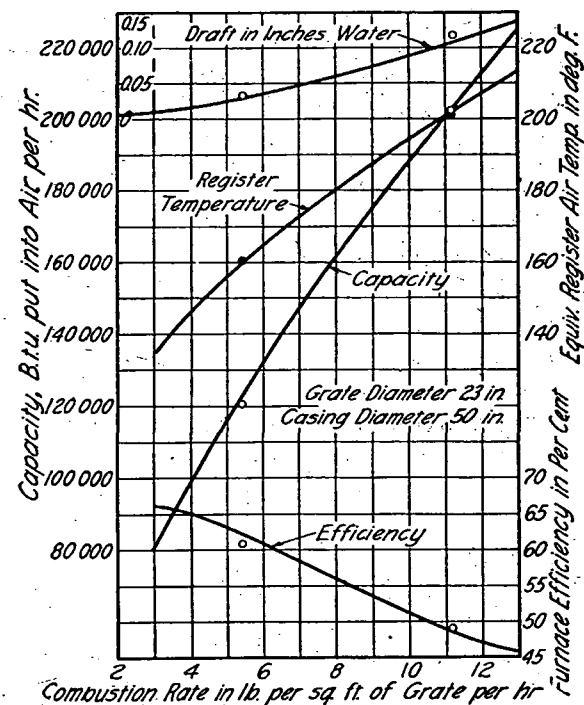


FIG. 7. TYPICAL PERFORMANCE CURVES FOR A WARM AIR FURNACE AND INSTALLATION IN A THREE STORY TEN LEADER PLANT, OPERATING ON RECIRCULATED AIR

From the relation existing among these factors it is found (Fig. 7) that the capacity of the furnace under test is 147,750 Btu per hour for the total grate, which gives the capacity at the furnace bonnet per square foot of grate as 51,200 Btu and per square inch of grate as 356 Btu per hour.

Suppose it is desired to select a furnace to deliver air to the rooms at a register temperature approximating 160 F rather than 175 F. Referring to the curves, the relation is: combustion rate, 5.5 lb; register warm-air temperature, 160 F; and efficiency of the furnace, 62 per cent. Under this condition the capacity of the furnace at the furnace bonnet per square foot of grate area is 43,200 Btu per hour, and per square inch of grate it is 300 Btu per hour. From these performance values, the grate area for any

²Investigation of Warm-Air Furnaces and Heating Systems, Part N, by A. C. Willard, A. P. Kratz and V. S. Day (University of Illinois Engineering Experiment Station Bulletin No. 189).

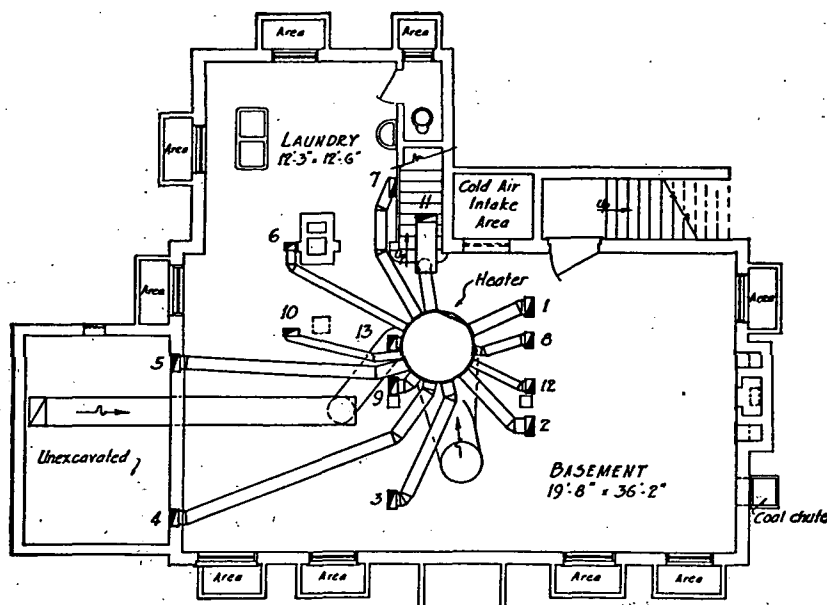


FIG. 8. BASEMENT PLAN, RESEARCH RESIDENCE

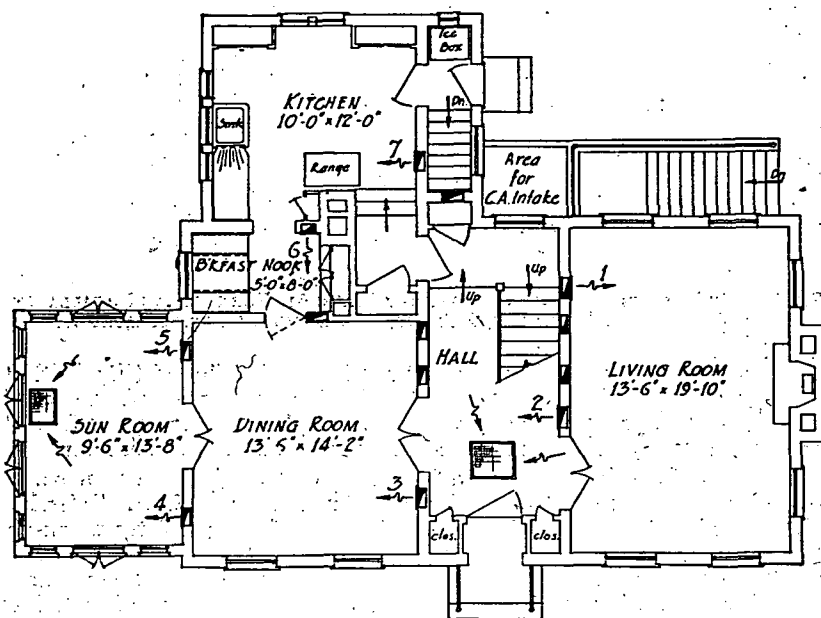


FIG. 9. FIRST-FLOOR PLAN, RESEARCH RESIDENCE

CHAPTER 24—GRAVITY WARM AIR FURNACE SYSTEMS

plant requirement (allowing 20 per cent heat loss between furnace and registers) will be:

$$\text{Grate area (175 F register temperature), square inches} = \frac{1.2 H}{356} = 0.0034H \quad (7)$$

$$\text{Grate area (160 F), square inches} = \frac{1.2 H}{300} = 0.0040H \quad (8)$$

Here H = Btu heat loss from the entire house per hour = summation of all room losses $H_1 + H_2$ + etc. + the Btu necessary to heat the fresh air, if any, at intake. This fresh air loss in Btu per hour will be approximately 1.27 times the cubic feet of air admitted through the intake per hour on a zero day. For systems which recirculate all the air this value will be zero. For systems which have a fresh air intake, controlled by damper, this value might well be approximated, since this loss will probably be reduced to a minimum on a zero day. Assume for such cases that the building loss is increased by 25 per cent, and that there is the usual 20 per cent loss between furnace and registers.

It is not always possible to obtain performance curves, and the following method is suggested as being a close check. An addition of 2 per cent of the furnace capacity is proposed for each unit that the heating surface to grate area ratio of the furnace exceeds 20. This addition is based on tests made at the University of Illinois, of four types of furnaces having various ratios of heating surface to grate area.

Let E = efficiency of the furnace.

f = fuel value of the coal, Btu per pound.

p = pounds of coal burned per square foot of grate per hour.

R = ratio of heating surface to grate area.

H = total heat requirements of the house.

$$\text{Grate area, square inches} = \frac{1.2 \times 144 H}{Efp [1 + 0.02 (R - 20)]} \text{ for all inside air.} \quad (9)$$

For coal having a heat value of 12,000 Btu, and a furnace having 60 per cent efficiency, with 6 lb of coal burned per square foot of grate per hour, and 20 sq ft of heating surface for 1 sq ft of grate, this becomes:

$$\text{Grate area, square inches} = \frac{1.2 \times 144 H}{0.60 \times 12,000 \times 6} \text{ for all inside air,} \quad (10)$$

and for another furnace having 24 sq ft of heating surface for 1 sq ft of grate the expression is

$$\text{Grate area, square inches} = \frac{1.2 \times 144 H}{0.60 \times 12,000 \times 6 [1 + 0.02 (24 - 20)]} \quad (11)$$

The air temperatures at the registers corresponding to the conditions of Equation 11 would be approximately 165 F, and for 175 F and 12,000 Btu the combustion rate would be about 7.5 lb with an efficiency of 57 per cent, using the curves of Fig. 7 as a guide.

TYPICAL DESIGN

The application of the preceding data to an actual example may be of assistance to the designer. Figs. 8, 9, 10 and 11 represent the plans of

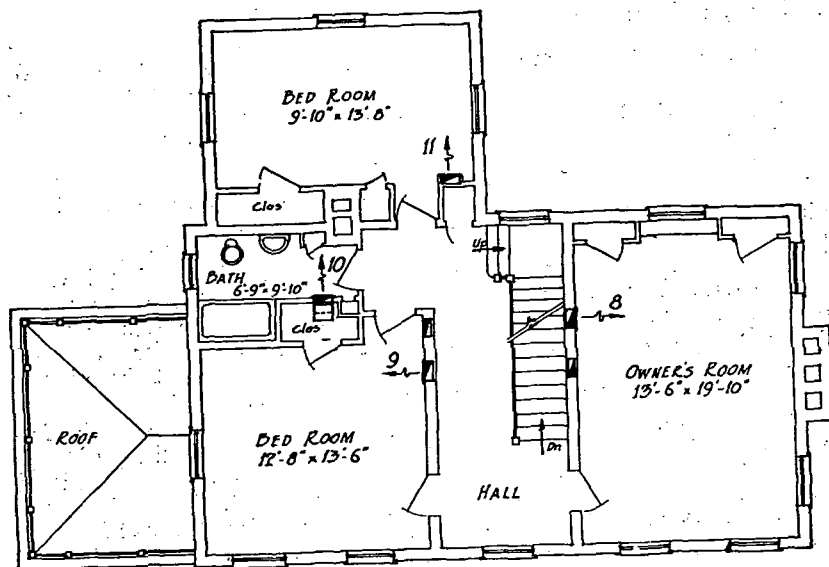


FIG. 10. SECOND-FLOOR PLAN, RESEARCH RESIDENCE

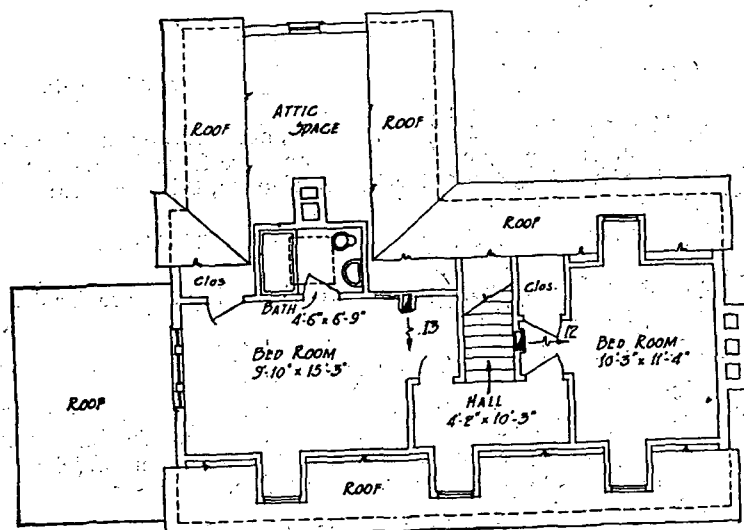


FIG. 11. THIRD-FLOOR PLAN, RESEARCH RESIDENCE

the Warm Air Research Residence of the *National Warm Air Heating and Air Conditioning Association* erected at the University of Illinois³.

Leaders, Stacks and Registers. (Direct Method)

Living Room, 1st floor:

$17,250 \div 111 = 155$ sq in. leader area. See summary, Table 1; also example under Standard Code⁴, Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 14 in.

Register size = 155 sq in. net area. Gross area = net area $\div$ 0.7 = 14 in. $\times$ 16 in.

Owner's Room, 2nd floor:

$15,030 \div 167 = 90$ sq in. leader area. See Summary Table; also example under Standard Code⁴, Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 11.4, say 12 in.

Stack area = $0.7 \times 90 = 63$ sq in. = say 5 in. $\times$ 12 in.

Register area = 90 sq in. net area. Gross area = net area $\div$ 0.7 = 12 $\times$ 12 or 12 in. $\times$ 14 in.

In like manner the leaders, stacks and registers are calculated for each room in the house.

Leaders, Stacks and Registers. (Code⁴ Method. See Art. 3, Sec. 1, 2, 3)

Living Room (Glass = 90, Net wall = 405, Cubic contents = 2405)

$$\text{Leader} = \left(\frac{90}{12} + \frac{405}{60} + \frac{2405}{800} \right) 9 = 155 \text{ sq in.}$$

Register, same as Direct Method.

Owner's Room (Glass = 68, Net wall = 394, Cubic contents = 2275)

$$\text{Leader} = \left(\frac{68}{12} + \frac{394}{60} + \frac{2275}{800} \right) 6 = 90 \text{ sq in.}$$

Stack and Register, same as Direct Method.

Assuming all air recirculated, the minimum furnace for the plant will be:

$$\text{Grate area} = 0.0034 \times 132,370 = 450 \text{ sq in.} = 24 \text{ in. diameter at } 175 \text{ F register temperature. (Equation 7)}$$

$$\text{Grate area} = 0.0040 \times 132,370 = 530 \text{ sq in.} = 26 \text{ in. diameter at } 160 \text{ F register temperature. (Equation 8)}$$

If provision should be made for certain outside air circulation, then increase the building heat loss by, say 25 per cent and obtain by Equation 7 a 27-in. grate and by Equations 8 and 10 a 29-in. grate.

Experiments at the University of Illinois⁵ have shown that the capacity of a furnace may be increased nearly three times by an adequate fan,

³Plans used with permission. Bathroom on third floor not heated.

⁴Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences. This code has been sponsored by the *National Warm Air Heating and Air Conditioning Association*, the *National Association of Sheet Metal Contractors*, and the *AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS*. It is recommended that the installation of all gravity warm air heating systems in residences be governed by the provisions of this code, the eighth edition of which may be obtained from the *National Warm Air Heating and Air Conditioning Association*, 50 W. Broad St., Columbus, Ohio.

⁵See University of Illinois Eng. Exp. Sta. Bulletin No. 120, p. 129.

with a constant register or delivery temperature maintained, *provided that the rate of fuel consumption can be increased to provide the necessary heat*. In other words, the capacity of a forced circulation system is limited by the ability of the chimney to produce a sufficient draft.

TABLE 1. SUMMARY OF DATA APPLIED TO WARM AIR RESEARCH RESIDENCE

Rooms	From Chapter 7 Estimating Heat Losses Btu Heat Losses H	Leader Area Sq. In.	Stack Area Sq. In. 0.7 × LA	Leader Diameter Inches	Stack Size Net	Register Size Gross
First Floor		= 0.009H				
Living.....	17250	155	---	14	---	14 × 16
Dining.....	6810	61	---	9	---	8 × 12
Breakfast.....	2300	21	---	8	---	8 × 10
Kitchen.....	9210	83	---	11 or 12	---	12 × 14
Sun.....	25710	230	---	Two 12	---	Two 12 × 14
Hall and stair	12570	113	---	12	---	12 × 14
Second Floor		= 0.006H				
Owner's.....	15030	90	63	11 or 12	5 × 12	12 × 14
S. W. Bed.....	9800	59	41	9	3½ × 12	8 × 12
Bath.....	2450	15	10	8	3 × 10	8 × 10
N. Bed.....	14800	89	62	11 or 12	5 × 12	12 × 14
Third Floor		= 0.005H				
E. Bed.....	8220	41	29	8	3 × 10	8 × 10
W. Bed.....	8220	41	29	8	3 × 10	8 × 10

BOOSTER FANS

Booster fans often may be arranged to operate when gas or oil burners are running and to stop automatically when the burners shut down. The booster equipment is most effective in increasing output at low operating temperatures. According to tests, efficiencies may be advanced from 60 per cent for gravity to 70 per cent with boosters at low operating temperatures, but at high operating temperatures gravity and booster efficiencies are almost identical⁶.

⁶See University of Illinois Eng. Exp. Sta. Bulletin No. 141, p. 79.

PROBLEMS IN PRACTICE

1 ● What may prohibit the use of a gravity warm air system in a large house having several exposed wings?

In a gravity warm air system, excessive vertical distances above the furnace cause little trouble in the design of the wall stacks, but excessive horizontal distances from the furnace should be carefully considered in the design of the leaders. To work effectively, a gravity warm air system should be balanced and leaders over 12 ft in length should be avoided if possible. Long leaders, if used, must be of ample size, well pitched, and well insulated. Large houses having exposed wings may require leaders much longer than 12 ft; infiltration may create severe back-drafts in the exposed wings; and the basement ceiling height may not be sufficient to allow the leaders to have a pitch of more than one inch per foot. These conditions may make the exposed wings very difficult to heat with a gravity system because of its low air head differentials.

- 2 ● A third-story bedroom has a calculated heat loss of 12,000 Btu per hour.
- What size leader pipe should be used for a 175 F register air temperature?
 - What size stack?
 - What size register?

a. Leader area = $\frac{12,000}{200} = 60$ sq in. Use leader with diameter of 9 in.

b. Stack area = $0.7 \times 60 = 42$ sq in. Use stack 3½ in. by 12 in.

c. Register gross area = $\frac{60}{0.7} = 85.7$ sq in. Use register 8 in. by 12 in.

- 3 ● The calculated heat loss of a house is 130,000 Btu per hour. Find the grate area required for the furnace under the following conditions:

Heating value of coal = 12,500 Btu per lb.

Furnace efficiency = 55 per cent.

Combustion rate = 7.5 lb per sq ft per hour.

Ratio of heating surface to grate area of furnace = 20 to 1.

Register temperature = 175 F.

Loss between furnace and registers = 20 per cent.

Grate area = $\frac{1.2 \times 144 \times 130,000}{0.60 \times 12,500 \times 7.5} = 399.5$ sq in.

Grate diameter = 22.6 in.

Use grate with diameter of 23 in.

- 4 ● If in Question 3 the conditions were the same except that the ratio of heating surface to grate area of furnace was 24 to 1, what size grate would be required for the furnace?

Grate area = $\frac{1.2 \times 144 \times 130,000}{0.60 \times 12,500 \times 7.5} \times \frac{1}{1 + 0.02(24 - 20)} = \frac{399.5}{1.08} = 370$ sq in.

Grate diameter = 21.7 sq in.

Select grate with diameter of 22 in.

- 5 ● Name the items involved in the design of a furnace heating system.

- Heat loss from each room, Btu.
- Area and dimensions of warm-air pipes in basement, inches.
- Area and dimensions of vertical pipes, inches.
- Free and gross area and dimensions of warm-air registers, inches.
- Area and dimensions of recirculating or outside air ducts, inches.
- Free and gross area and dimensions of recirculating registers, inches.
- Size of furnace necessary to supply the warm air to overcome the heat loss.
- Area and dimension of chimney and smoke pipe, inches.

- 6 ● Discuss the design features of recirculating ducts.

- Their area should be equal to or greater than that of the supply ducts.
- They should be streamlined, and have a minimum number of turns.
- All runs should be as short as possible.
- Account should be taken of all cold walls and window areas in determining sizes and positions of return air inlets.
- The return line should be pitched downward toward the furnace. It should be designed to minimize friction.
- The top of the shoe or boot should never be above the grate level.

7 • Discuss the use of a booster fan. What effect has a booster fan at low operating temperatures? At high ones?

A booster fan is useful in accelerating the air flow past the surface of a low temperature furnace, where only a small weight differential in the air is created, and in unbalancing a gravity system so flow is established. The first use involves the entire plant, and increases efficiency about 10 per cent with low temperature operation; the second involves only the leaders in which air flow is accelerated. At high operating temperatures the difference in weight between warm outgoing air and cool incoming air is great enough to make a booster unnecessary with ordinary gravity systems.

Chapter 25

BOILERS

Cast-Iron Boilers, Steel Boilers, Special Heating Boilers, Gas-Fired Boilers, Hot Water Supply Boilers, Furnace Design, Heating Surface, Testing and Rating Codes, Output, Efficiency, Selection of Boilers, Connections and Fittings, Erection, Operation and Maintenance, Boiler Insulation

STEAM and hot water boilers for low pressure heating work are built in a wide variety of types, many of which are illustrated in the *Catalog Data Section*, and are classified as (1) cast-iron sectional, (2) steel fire tube, (3) steel water tube, and (4) special.

CAST-IRON BOILERS

Cast-iron boilers may be of round pattern with circular grate and horizontal pancake sections joined by push nipples and tie rods, or of rectangular pattern with vertical sections. The latter type may be either of outside header construction where each section is independent of the other and the water and steam connections are made externally through these headers, or assembled with push nipples and tie rods, in which case the water and steam connections are internal.

Cast-iron boilers usually are shipped knocked down to facilitate handling at the place of installation where assembly is made. One of the chief advantages of cast-iron boilers is that the separate sections can be taken into or out of basements and other places more or less inaccessible after the building is constructed. This feature is of importance in making repairs to or replacing a damaged or worn out boiler and should be given consideration in the original selection. Sufficient space should be provided in the boiler room for assembling the boiler and for disassembling it conveniently if repairs are needed. With the outside header type of boiler a damaged section in the middle of the boiler can be removed without disturbing the other sections and sufficient side clearance should be provided for this contingency.

Capacities of cast-iron boilers range from that required for small residences up to about 18,000 sq ft of steam radiation. For larger loads, cast-iron boilers must be installed in multiple, or a steel boiler must be used. In most cases cast-iron boilers are limited to working pressures of 15 lb for steam and 30 lb for water. Special types are built for hot water supply which will withstand higher local water pressures.

STEEL BOILERS

Two general classifications may be applied to steel boilers: *first*, with regard to the relative position of water and hot gases, distinguished as fire

tube or water tube; *second*, with regard to arrangement of furnace and flues, as (1) horizontal return tubular (HRT) boilers, (2) portable (self-contained) firebox boilers with either water or fire tubes, and (3) water tube boilers of the power type.

Fire tube boilers are constructed so that the water available to produce steam is contained in comparatively large bodies distributed outside of the boiler tubes, the hot gases passing within the tubes. In *water tube* boilers, the water is circulated within the boiler tubes, heat being applied externally to them.

The *HRT boiler* is the oldest type and consists of a horizontal cylindrical shell with fire tubes, enclosed in brickwork to form the furnace and

TABLE 1. PRACTICAL COMBUSTION RATES FOR SMALL COAL-FIRED HEATING BOILERS OPERATING ON NATURAL DRAFT OF FROM $\frac{1}{8}$ IN. TO $\frac{1}{2}$ 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

^aSteel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

combustion chamber. All heating surfaces and the interior of the boiler are accessible for both cleaning and inspection. Horizontal return tubular boilers, especially the larger sizes, should be suspended from structural columns and beams independent of the brick setting. Small HRT boilers sometimes are supported by brackets resting on the brick setting.

Portable firebox boilers are the more generally used type of steel heating boilers, their outstanding characteristic being the water-jacketed firebox which eliminates virtually all brickwork. They are shipped in one piece from the factory and come to the job ready for immediate hook-up to piping. They may be of welded or riveted construction and have either water or fire tubes. Manufacturers' catalogs usually list heating surface as well as grate area. The elimination of brickwork also makes this type the most compact of steel boilers as well as the lowest in first cost.

Water tube boilers. For large heating loads water tube boilers are quite frequently used. They usually require more head room than other types of boilers but require considerably less floor space and make possible a

much higher rate of evaporation per square foot of heating surface, with proper setting, baffling and draft. Water tube boilers used for heating purposes are brick set, supported on structural steel columns and have the brick setting encased in an insulated steel housing to prevent air infiltration and to minimize heat losses. For large heating loads at a high rate of evaporation, such boilers should be operated at pressures above 15 lb per square inch with a pressure-reducing valve on the connection to the heating main.

SPECIAL HEATING BOILERS

A special type of boiler, known as the *magazine feed boiler*, has been developed for the burning of small sizes of anthracite and coke. These are built of both cast-iron and steel, and have a large fuel-carrying capacity which results in longer firing periods than would be the case with the standard types using buckwheat sizes of coal. Special attention must be given to insure adequate draft and proper chimney sizes and connections.

Oil-burner boiler units, in which a special boiler has been designed with a furnace shaped to meet the general requirements of oil burners or are specially adapted to one particular burner have been developed by a number of manufacturers. These usually are compact units with the burner and all controls enclosed within an insulated steel jacket. Ample furnace volume is provided for efficient combustion, and the heating surfaces are proportioned for effective heat transfer. Consequently, higher efficiencies are obtainable than with the ordinary coal fired boiler converted to oil firing.

GAS-FIRED BOILERS

Gas boilers have assumed a well-defined individuality. The usual boiler is sectional in construction with a number of independent burners placed beneath the sections. In most boilers each section has its own burner. In all cases the sections are placed quite closely together, much closer than would be possible when burning a soot-forming fuel. The effort of the designer is always to break the hot gas up into thin streams, so that all particles of the heat-carrying gases can come as close as possible to the heat-absorbing surfaces. Because there is no fuel bed resistance and because the gas company supplies the motive power to draw in the air necessary for combustion (in the form of the initial gas pressure), draft losses through gas boilers are low.

HOT WATER SUPPLY BOILERS

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-making properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. 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 heating system is installed, the domestic hot water usually is obtained from an indirect heater placed below the water line of the boiler.

FURNACE DESIGN

Good efficiency and proper boiler performance are dependent on correct furnace design embodying sufficient volume for burning the particular fuel at hand, which requires thorough mixing of air and gases at a high temperature with a velocity low enough to permit complete combustion of all the volatiles. On account of the small amount of volatiles contained in coke, anthracite, and semi-bituminous coal, these fuels can be burned efficiently with less furnace volume than is required for bituminous coal, the combustion space being proportioned according to the amount of volatiles present.

Combustion should take place before the gases are cooled by the boiler heating surface, and the volume of the furnace must be sufficient for this purpose. The furnace temperature must be maintained sufficiently high to produce complete combustion, thus resulting in a higher CO_2 content and the absence of CO . Hydrocarbon gases ignite at temperatures varying from 1000 to 1500 F.

The question of furnace proportions, particularly in regard to mechanical stoker installations, has been given some consideration by various manufacturers' associations. Arbitrary values have been recommended for minimum dimensions. A customary rule-of-thumb method of figuring furnace volumes is to allow 1 cu ft of space for a maximum heat release of 50,000 Btu per hour. This value is equivalent to allowing approximately 1 cu ft for each developed horsepower, and it is approved by most smoke prevention organizations.

The setting height will vary with the type of stoker. In an overfeed stoker, for instance, all the volatiles must be burned in the combustion chamber and, therefore, a greater distance should be allowed than for an underfeed stoker where a considerable portion of the gas is burned while passing through the incandescent fuel bed. The design of the boiler also may affect the setting height, since in certain types the gas enters the tubes immediately after leaving the combustion chamber, while in others it passes over a bridge wall and toward the rear, thus giving a better opportunity for combustion by obtaining a longer travel before entering the tubes.

To secure suitable furnace volume, especially for mechanical stokers or oil burners, it often is necessary either to pit the stoker or oil burner, or

where water line conditions and headroom permit, to raise the boiler on a brick foundation setting.

Smokeless combustion of the more volatile bituminous coals is furthered by the use of mechanical stokers. (See Chapter 28.) Smokeless combustion in hand-fired boilers burning high volatile solid fuel is aided (1) by the use of double grates with down-draft through the upper grate, (2) by the use of a curtain section through which preheated auxiliary air is introduced over the fire toward the rear of the boiler, and (3) by the introduction of preheated air through passages at the front of the boiler. All three methods depend largely on mixing secondary air with the partially burned volatiles and causing this mixture to pass over an incandescent fuel bed, thus tending to secure more complete combustion than is possible in boilers without such provision.

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 heat transfer capacity of a radiant heating surface may be as high as 6 to 8 times that of an indirect surface. This is one of the reasons why the water legs of some boilers have been extended, especially in the case of stoker firing where the extra amount of combustion chamber secured by an extension of the water legs is important. For the same reason, care should be exercised in building a refractory combustion chamber in an oil-burning boiler so as not to screen any more of this valuable surface with refractories than is necessary for good combustion.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. Investigations¹ by the U. S. Bureau of Mines show that:

1. A boiler in which the heating surface is arranged to give long gas passages of small cross-section will be more efficient than a boiler in which the gas passages are short and of larger cross-section.
2. The efficiency of a water tube boiler increases as the free area between individual tubes decreases and as the length of the gas pass increases.
3. By inserting baffles so that the heating surface is arranged in series with respect to the gas flow, the boiler efficiency will be increased.

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.

Heat Transfer Rates

Practical rates of heat transfer in heating boilers will average about

¹See U. S. Bureau of Mines Bulletin No. 18. The Transmission of Heat into Steam Boilers.

3300 Btu per sq ft per hour for hand-fired boilers and 4000 Btu per sq ft per hour for mechanically fired boilers when operating at *design load*². When operating at *maximum load*³ these values will run between 5000 and 6000 Btu per sq ft per hour. Boilers operating under favorable conditions at the above heat transfer rates will give exit gas temperatures that are considered consistent with good practice.

TESTING AND RATING CODES

The Society has adopted three solid fuel testing codes, a solid fuel rating code and an oil fuel testing code. A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June 1929)⁴, are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics. A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)⁵ is intended for use with A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers⁶. 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 A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel⁶ 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. The *Steel Heating Boiler Institute* suggests a single number dimensional rating in the *S.H.B.I.* Code for the Rating of Low-Pressure Heating Boilers by Their Physical Characteristics⁶.

BOILER OUTPUT

Boiler output as defined in A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) is the quantity of heat available at the boiler nozzle with the boiler normally insulated. It should be based on actual tests conducted in accordance with this code. This output is usually stated in Btu and in square feet of equivalent heating surface (radiation). According to the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers, the performance data should be given in tabular or curve form on the following items for at least five outputs ranging from maximum down to 35 per cent of maximum: (1) fuel available, (2) combustion rate, (3) efficiency, (4) draft tension, (5) flue gas temperature. The only definite restriction placed on setting the maximum output is that priming shall not exceed 2 per cent. These curves provide complete data regarding the performance of the boiler under test conditions. Certain other pertinent information, such as grate area, heating surface and chimney dimensions is desirable also in forming an opinion of how the boiler will perform in actual service.

The output of large heating boilers is frequently stated in terms of

²For definitions of design load and maximum load see pages 439 and 440.

³See A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929. Also Chapter 44.

⁴See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930. Also Chapter 44.

⁵See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931. Also Chapter 44.

⁶See Rating of Heating Boilers by Their Physical Characteristics, by C. E. Bronson (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

boiler horsepower instead of in Btu per hour or square feet of equivalent radiation.

Boiler Horsepower: The evaporation of 34.5 lb of water per hour from and at 212 F which is equivalent to a heat output of $970.2 \times 34.5 = 33,471.9$ Btu per hour.

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 this same temperature and at atmospheric pressure.

It is usually considered that 10 sq ft of boiler heating surface will produce a rated boiler horsepower. A rated boiler horsepower in turn can carry a design load of from 100 to 140 sq ft of equivalent radiation. It is apparent, therefore, that 1 sq ft of boiler heating surface can carry a design load of from 10 to 14 sq ft of equivalent radiation, or somewhat more if the boiler is forced above rating. The application of these values is discussed under the heading Selection of Boilers.

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 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 of fuel to the calorific value of 1 lb of fuel.

Solid fuel boilers usually show an efficiency of 50 to 75 per cent when operated under favorable conditions at their rated capacities. Information on the combined efficiencies of boiler, furnace and burner has resulted from research conducted at Yale University in cooperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*⁷. For general information on heating efficiencies see Chapter 29.

SELECTION OF BOILERS

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932).

The estimated design load is the sum of the following three items⁸:

1. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect or central fan) to be installed.

⁷See A.S.H.V.E. research papers entitled Study of the Characteristics of Oil Burners and Heating Boilers, by L. E. Seely and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931), and A Study of Intermittent Operation of Oil Burners, by L. E. Seely and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

⁸See A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

2. The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler.
3. The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or the equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932.)

The estimated maximum load is given by⁸:

4. The estimated increase in the normal load in Btu per hour due to starting up cold radiation. This percentage of increase is to be based on the sum of Items 1, 2 and 3 and the heating-up factors given in Table 2.

TABLE 2. WARMING-UP ALLOWANCES FOR LOW PRESSURE STEAM AND HOT WATER HEATING BOILERS^{a, b, c}

DESIGN LOAD (REPRESENTING SUMMATION OF ITEMS 1, 2, AND 3, ^d)		PERCENTAGE CAPACITY TO ADD FOR WARMING UP
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

^aThis 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.

^bSee also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggart (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913); 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); Selecting the Right Size Heating Boiler, by Sabin Crocker (Heating, Piping and Air Conditioning, March, 1932).

^cThis table refers to hand-fired solid fuel boilers. A factor of 25 per cent over design load is adequate when oil or gas are used as fuels.

^d240 Btu per square foot.

Other things to be considered are:

5. Efficiency with hard or soft coal, gas, or oil firing, as the case may be.
6. Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas.
7. Combustion space in the furnace.
8. Type of heat liberation, whether continuous or intermittent, or a combination of both.
9. Miscellaneous items consisting of draft available, character of attendance, possibility of future extension, possibility of breakdown, headroom in the boiler room.

Radiation Load

The connected radiation (Item 1) is determined by calculating the heat losses in accordance with data given in Chapters 5, 6 and 7, and dividing by 240 to change to square feet of equivalent radiation as explained in Chapter 30. For hot water, the emission commonly used is 150 Btu per square foot, but the actual emission depends on the temperature of the medium in the heating units and of the surrounding air. (See Chapter 30.)

Although it is customary to use the actual connected load in equivalent square feet of radiation for selecting the size of boiler, this connected load usually represents a reserve in heating capacity to provide for infiltration in the various spaces of the building to be heated, which reserve, however,

is not in use at all places at the same time, or in any one place at all times. For a further discussion of this subject see Chapter 6.

Hot Water Supply Load

When the hot water supply (Item 2) is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. The

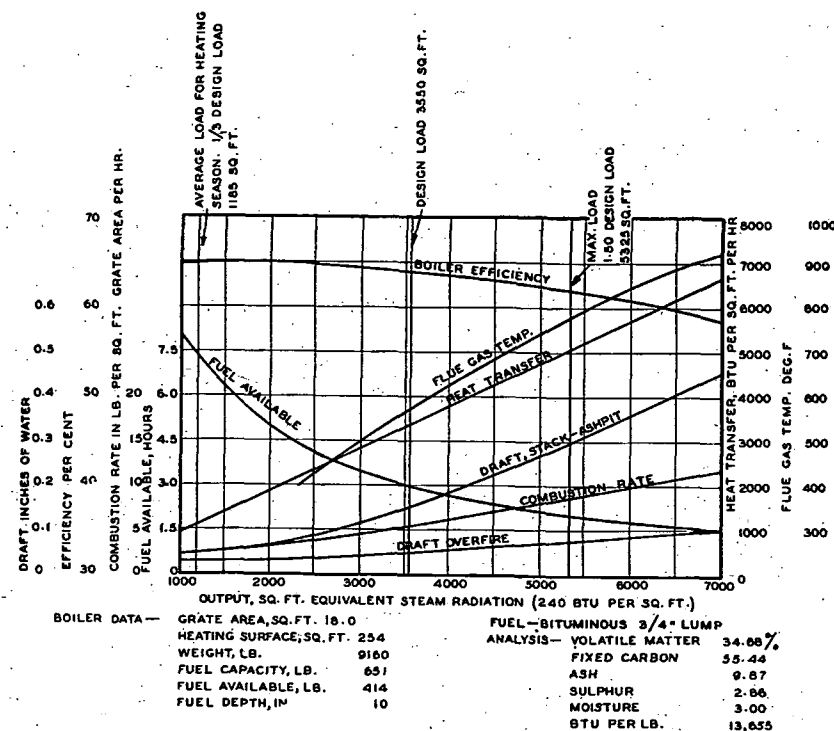


FIG. 1. TYPICAL PERFORMANCE CURVES FOR A 36-IN. CAST-IRON SECTIONAL STEAM HEATING BOILER, BASED ON THE A.S.H.V.E. CODE FOR RATING STEAM HEATING SOLID-FUEL HAND-FIRED BOILERS

allowance to be made will depend on the amount of water heated and its temperature rise. A good approximation is to add 4 sq ft of equivalent radiation for each gallon of water heated per hour through a temperature range of 100 F. For more specific information, see Chapter 35.

Piping Tax (Item 3)

It is common practice to add a flat percentage allowance to the equivalent connected radiation to provide for the heat loss from bare and covered pipe in the supply and return lines. The use of a flat allowance of 25 per cent for steam systems and 35 per cent for hot water systems is preferable to ignoring entirely the load due to heat loss from the supply

and return lines, but better practice, especially when there is much bare pipe, is to compute the emission from both bare and covered pipe surface in accordance with data in Chapter 36. With direct radiation served by bare supply and return piping the percentages may be higher than those stated, while in the case of unit heaters where the output is concentrated in a few locations, the piping tax may be 10 per cent or less.

Warming-Up Allowance

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature and heating up cold radiation and piping. (See Item 4.) The factors to be used for determining the allowance to be made should be selected from Table 2 and should be applied to the estimated design load as determined by Items 1, 2 and 3.

Performance Curves for Boiler Selection

In the selection of a boiler to meet the estimated load, the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers recommends the use of performance curves based on actual tests conducted in accordance with the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3), similar to the typical curves shown in Fig. 1. It should be understood that performance data apply to test conditions and that a reasonable allowance should be made for decreased output resulting from soot deposit, poor fuel or inefficient attention.

Selection Based on Heating Surface and Grate Area

Where performance curves are not 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 consisting of connected radiation, piping tax and domestic water heating load. As stated in the section on Boiler Output, 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.

Due to the wide variation 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:

$$G = \frac{H}{C \times F \times E} \quad (1)$$

where

G = grate area, square feet.

H = required total heat output of the boiler, Btu per hour (see Selection of Boilers, p. 439).

C = combustion rate in pounds of dry coal per square foot of grate area per hour, depending on the kind of fuel and size of boiler as given in Table 1.

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 heat 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 Formula 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

Gas-heating appliances should be selected in accordance with factors given in Table 1, Chapter 28, which include an allowance for heating up cold radiation, and for the piping tax. These factors are for thermostatically-controlled systems; in case manual operation is desired, a warming-up allowance of 100 per cent is recommended by the A.G.A. A gas boiler selected by the use of the A.G.A. factors will be the minimum size boiler which can carry the load. From a fuel economy standpoint, it may be advisable to select a somewhat larger boiler and then throttle the gas and air adjustments as required. This will tend to give a low stack temperature with high efficiency and at the same time provide reserve capacity in case the load is underestimated or more is added in the future.

Conversions

The conversion of a coal or oil boiler to gas burning is simpler than the reverse since little furnace volume need be provided for the proper combustion of gas. When a solid fuel boiler of 500 sq ft or less capacity is converted to gas burning, the necessary gas heat units should be approximately double the connected load. The presumption for a conversion job is that the boiler is installed and probably will not be made larger; therefore, it is a matter of setting a gas-burning rate to obtain best results with the available surface. Assuming a combustion efficiency of 75 per cent for a conversion installation the boiler output would be $2 \times 0.75 = 1.5$ times the connected load, which allows 50 per cent for piping tax and pickup. In converting large boilers, the determination of the required Btu input should not be done by an arbitrary figure or factor but should be based on a detailed consideration of the requirements and characteristics of the connected load.

An efficient conversion installation depends upon the proper size of flue connection. Often the original smoke breeching between the boiler and chimney is too large for gas firing, and in this case, flue orifices can be used, which are discs provided with an opening of the size for the gas input used in this boiler. The size should be based on 1 sq in. of flue area for each 7500 hourly Btu input.

If dampers are found in the breeching they should be locked in position so that they will not interfere with the normal operation of the gas burners at maximum flow. In the case of large boiler conversions, automatic damper regulators proportion the position of the flue dampers to the amount of gas flowing and may be substituted for existing dampers. Generally in residence conversions automatic dampers are not of the

proportioning type but close the flue during the off periods of the gas burners. Automatic shutoff dampers should be located between the backdraft diverter and the chimney flue.

Other Considerations in Selection of Boilers

As it will usually be found that several boilers will meet the specifications, the final selection of the boiler may be influenced by other considerations, some of which are:

1. Dimensions of boiler.
2. Durability under service.
3. Convenience in firing and cleaning.
4. Adaptability to changes in fuel and kind of attention.
5. Height of water line.

In large installations, the use of several smaller boiler units instead of one larger one will obtain greater flexibility and economy by permitting the operation, at the best efficiency, of the required number of units according to the heat requirements.

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 flues, and should be at least 3 ft greater than the length of the boiler firebox.

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, and 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

The velocity of flow through the outlets of low pressure steam heating boilers should not exceed 15 to 25 fps if fluctuation of the water line and undue entrainment of moisture are to be avoided. Steam or water outlet connections preferably should be the full size of the manufacturers' tapping and should extend vertically to the maximum height available above the boiler. For gravity circulating steam heating systems, it is recommended that a Hartford Loop, described in Chapter 32, be utilized in making the return connection.

Particular attention should be given to *fitting connections* to secure conformity with the A.S.M.E. Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Steam gages should be fitted with a water seal and a shut-off consisting

of a cock with either a tee or lever handle which is parallel to the pipe when the cock is open. *Steam gage connections* should be of copper or brass when smaller than 1 in. I.P.S.,⁹ if the gage is more than 5 ft from the boiler connection, and also in any case where the connection is less than 1/2 in. I.P.S.

Each steam or vapor boiler should have at least one *water gage* glass and two or more *gage cocks* located within the range of the visible length of the glass. The water gage fittings or gage cocks may be direct connected to the boiler, if so located by the manufacturer, or may be mounted on a separate water column. No connections, except for combustion regulators, drains or steam gages, should be placed on the pipes connecting the water column and the boiler. If the water column or gage glass is connected to the boiler by pipe and fittings, a cross, tee or equivalent, in which a cleanout plug or a drain valve and piping may be attached, should be placed in the water connection at every right-angle turn to facilitate cleaning. The water line in steam boilers should be carried at the level specified by the boiler manufacturer.

Safety valves should be capable of discharging all the steam that can be generated by the boiler without allowing the pressure to rise more than 5 lb above the maximum allowable working pressure of the boiler. This should be borne in mind particularly in the case of boilers equipped with mechanical stokers or oil burners where the amount of grate area has little significance as to the steam generating capacity of the boiler.

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 Chapters 32 and 33 and the A.S.M.E. 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 up from the inside of the chimney. A thimble or sleeve grout usually is provided where the breeching enters a brick chimney.

⁹A.S.M.E. Code, Identification of Piping Systems.

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 always should 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.

The following precautions should be taken in all installations to prevent damage to the boiler:

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 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 boilers mechanically fired.

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; and (h) insufficient radiation installed.
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; (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; (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; (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 due to air leaks into chimney or breeching; (b) inferior fuel; (c) inferior attention; (d) accumulation of clinker on grate; (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; (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; (e) improper reduction in breeching size.

Cleaning Steam 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 have a tendency 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.

It is common practice when starting new installations to discharge heating returns to the sewer during the first week of operation. This prevents the passage of grease, dirt or other foreign matter into the boiler and consequently may avoid the necessity of cleaning the boiler. During the time the returns are being passed to the sewer, the feed valve should be cracked sufficiently to maintain the proper water level in the boiler.

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 of some one inadvertently building a fire in a dry boiler, however, it is safer to keep the boiler filled with water. 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.

BOILER INSULATION

Insulation for cast-iron boilers is of two general types: (1) plastic material or blocks wired on, cemented and covered with canvas or duck; and (2) blocks, sheets or plastic material covered with a metal jacket furnished by the boiler manufacturer. Self-contained steel firebox boilers usually are insulated with blocks, cement and canvas, or rock wool blankets; HRT boilers are brick set and do not require insulation beyond that provided in the setting. It is essential that the insulation on a boiler and adjacent piping be of non-combustible material as even slow-burning insulation constitutes a dangerous fire hazard in case of low water in the boiler.

REFERENCES

- A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings.
- A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2).
- A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3).
- A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers. Heating, Ventilating and Air Conditioning, by Harding and Willard, Revised Edition, 1932.
- A.S.M.E. Boiler Construction Code for Low Pressure Heating Boilers.
- Heating, Piping and Air Conditioning. Contractors National Association Standards (boiler selection tables).
- House-Heating, published by American Gas Association.
- Handbook of Oil Burning, published by American Oil Burner Association.
- Heating and Air Conditioning, by Allen and Walker (fourth edition).
- Selecting the Right Size Boiler, by Sabin Crocker (Heating, Piping and Air Conditioning, February, March, April, 1932).

PROBLEMS IN PRACTICE

- 1 ● What basic requirements of boiler design are to be accomplished with a combination boiler and oil burner unit?

Combination units vary widely but in general, the basic requirements of design depends upon a combustion chamber of proper design and arranged for the flame shape with adequate heating surface for the complete combustion of the fuel.

- 2 ● Name the construction materials that distinguish two types of low pressure heating boilers.

- a. Cast-iron.
- b. Steel.

- 3 ● What is the normal rating range of each type of boiler?

- a. Cast-iron boilers are rated at from 200 to 18,000 sq ft EDR.
- b. Steel boilers are rated at from 300 to 50,000 sq ft EDR.

- 4 ● a. What is meant by direct boiler heating surface?

- b. What is meant by indirect boiler heating surface?

- a. Direct boiler heating surface is that boiler surface upon which the fire shines, namely, the walls of the firebox and the crown sheet.
- b. Indirect boiler heating surface is that boiler surface not exposed to the direct rays of the fire and over which heated gases pass after they have been in contact with the direct surface. Indirect surface is generally known as convective surface.

- 5 ● What is the average heat transmission rate in heating boilers in Btu per sq ft of heating surface per hour?

3500 for coal burning boilers; 4200 for oil burning boilers.

- 6 ● What factors contribute to economical fuel operation in low pressure boilers burning coal or oil?

- a. Proper furnace volume for complete combustion.
- b. Arrangement of heating surfaces in series to create a turbulent and scrubbing contact of gases against the convective surfaces.
- c. Rapid internal water circulation which will remove steam bubbles from the water side of heating surfaces and allow other steam bubbles to be formed. Rapid disengagement of steam bubbles increases the steam generating efficiency of each unit area of heating surface, and thereby lowers flue gas temperatures.

- 7 ● What equipment is usually directly attached to a low pressure heating boiler?

For coal burning steam boilers: water column, water gage, tri-cocks, steam gage, lever pop safety valve, boiler damper regulator.

For coal burning hot water boilers: damper regulator, altitude gage, thermometer, relief valve.

For oil burning boilers, the damper regulators are omitted and the following additional equipment is usually attached: automatic water feeder, low water cutout, a pressure control, and a water temperature control. These are generally furnished by the oil burner manufacturer and do not come with the boiler.

- 8 ● What general precautions regarding the boiler should be taken to make sure a proposed heating installation will work properly?

- a. Select the right size and type of boiler.
- b. Be sure the combustion space is proper for the type of fuel burned.
- c. Allow sufficient space around the boiler for cleaning.
- d. Secure proper height and area of chimney and connecting breeching.
- e. Clean the boiler thoroughly and provide surface blowoff connections and bottom blowoff connections for periodic cleaning after operation is begun.
- f. See that the boiler heating surface is cleaned at regular periods.
- g. Check flue gas temperatures and make a flue gas analysis at least once a month.
- h. Secure information and advice from boiler manufacturer.

- 9 ● Below what temperature should the water in direct water heaters be maintained to reduce scale formation and corrosion?

140 F.

10 ● a. What is the heat equivalent of a boiler horsepower?

b. How many square feet of heating surface are usually required per boiler horsepower?

a. 33,471.9 Btu per hour.

b. 10 sq ft.

11 ● What is meant by equivalent evaporation?

The amount of water that a boiler would evaporate per hour, if the feed water were at 212 F and if the steam were evaporated at that temperature; this is usually spoken of as "from and at 212 F."

12 ● What loads must be considered in determining the boiler capacity required for a given installation?

Radiation load.

Hot water supply load.

Piping tax.

Warming-up allowance.

Load allowance for inefficient firing.

13 ● A boiler has 6 sq ft of grate area with a possible depth of fuel bed of 18 in. The fuel burned is bituminous coal with a heat value of 12,500 Btu per lb. The efficiency is assumed to be 50 per cent. How great a maximum load will this boiler carry if it is to be fired every 8 hours and if 20 per cent of the fuel is to be left over to kindle the next charge?

Volume of fuel bed = $6 \times 1.5 = 9.0$ cu ft.

Available volume = $0.80 \times 9.0 = 7.2$ cu ft.

Weight of available fuel = $40 \times 7.2 = 288.0$ lb.

Fuel burned per hour = $\frac{288.0}{8} = 36.0$ lb.

Heat released = $36.0 \times 12,500 \times 0.50 = 225,000$ Btu per hour.

Maximum load = $\frac{225,000}{240} = 938$ equivalent square feet.

14 ● What are the usual causes of unsteady water line and priming?

Grease and dirt in boiler.

Overload, resulting in insufficient steam liberating area.

Small outlet connections.

15 ● What type of return connection can be used for gravity steam heating systems to make the use of check valves unnecessary?

The Hartford Loop. (See Chapter 32.)

Chapter 26

CHIMNEYS AND DRAFT CALCULATIONS

Natural Draft, Mechanical Draft, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Chimney Construction, Chimneys for Gas Heating

THE design and construction of a chimney is so important a part of the heating engineer's work that a general knowledge of draft characteristics and calculations is essential.

Draft, in general, may be defined as the pressure difference between the atmospheric pressure and that at any part of an installation through which the gases flow. Since a pressure difference implies a head, draft is a static force. While no element of motion is inferred, yet motion in the form of circulation of gases throughout an entire boiler plant installation is the direct result of draft. This motion is due to the pressure difference, or unbalanced pressure, which compels the gases to flow. Draft is often classified into two kinds according to whether it is created thermally or artificially, viz, (1) natural or thermal draft, and (2) artificial or mechanical draft.

Natural Draft

Natural draft is the difference in pressure produced by the difference in weight between the relatively hot gases inside a natural draft chimney and an equivalent column of the cooler outside air, or atmosphere. Natural draft, in other words, is an unbalanced pressure produced thermally by a natural draft chimney as the pressure transformer and a temperature difference. The intensity of natural draft depends, for the most part, upon the height of the chimney above the grate bar level and also the temperature difference between the chimney gases and the atmosphere.

A typical natural draft system consists essentially of a relatively tall chimney built of steel, brick, or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all the heat has not been extracted. Hot gases are an essential element in the operation of a natural draft system, although inherently a heat balance loss.

A natural draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a nuisance.

Natural draft is quite advantageous in installations where the total loss of draft due to resistances is relatively low and also in plants which have practically a constant load and whose boilers are seldom operated above

their normal rating. Natural draft systems have been, and are still being, employed in the operation of large plants during the periods when the boilers are operated only up to their normal rating. When the rate of operation is increased above the normal rating, some form of mechanical draft is employed as an auxiliary to overcome the increased resistances or draft losses. Natural draft systems are used almost exclusively in the smaller size plants where the amount of gases generated is relatively small and it would be expensive to install and operate a mechanical draft system.

The principal advantages of natural draft systems may be summarized as follows: (1) simplicity, (2) reliability, (3) freedom from mechanical

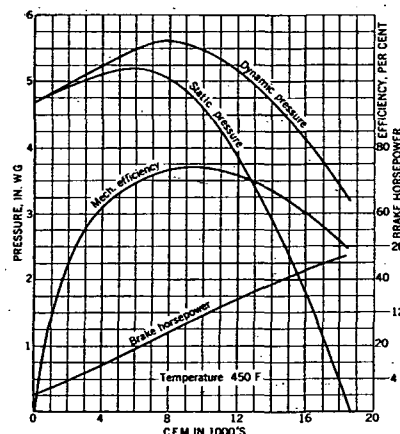


FIG. 1. GENERAL OPERATING CHARACTERISTICS OF TYPICAL INDUCED DRAFT FAN

parts, (4) low cost of maintenance, (5) relatively long life, (6) relatively low depreciation, and (7) no power required to operate. The principal disadvantages are: (1) lack of flexibility, (2) irregularity, (3) affected by surroundings, and (4) affected by temperature changes.

Mechanical Draft

Artificial draft, or mechanical draft, as it is more commonly called, is a difference in pressure produced either directly or indirectly by a forced draft fan, an induced draft fan, or a Venturi chimney as the pressure transformer. The intensity of mechanical draft is dependent for the most part upon the size of the fan and the speed at which it is operated. The element of temperature does not enter into the creation of mechanical draft and therefore its intensity, unlike natural draft, is independent of the temperature of the gases and the atmosphere. Mechanical draft includes the induced and Venturi types of draft systems in which the pressure difference is the result of a suction, and also the forced draft system in which the pressure difference is the result of a blowing. Mechanical draft systems tend to produce a vacuum or a plenum, according as the system used in its production creates a pressure difference below, or above,

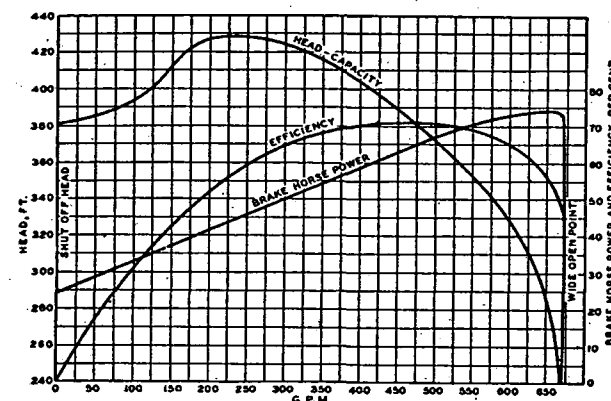


FIG. 2. OPERATING CHARACTERISTICS OF TYPICAL CENTRIFUGAL PUMP

atmospheric pressure, respectively. A mechanical draft system may be used either in conjunction with, or as an adjunct to, a natural draft system.

CHARACTERISTICS OF CHIMNEYS

In order to analyze the performance of a natural draft chimney, it may be advantageous to compare its general operating characteristics with those of a centrifugal pump and also of a centrifugally-induced draft fan, there being a similarity among the three. Figs. 1, 2 and 3 show the general operating characteristics of a typical centrifugally-induced draft fan, a typical centrifugal pump, and a typical natural draft chimney, respectively. The draft-capacity curve of the chimney corresponds to

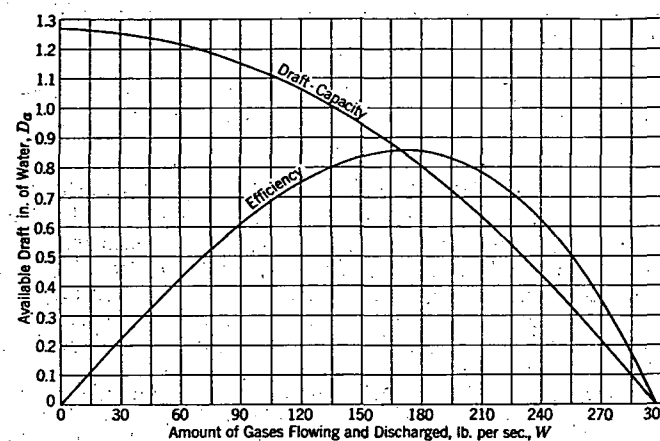


FIG. 3. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

the head-capacity curve of the pump and also to the dynamic-head capacity curve of the fan.

When the gases in the chimney are stationary, the draft created is termed the *theoretical draft*. When the gases are flowing, the theoretical intensity is diminished by the draft loss due to friction, the difference between the two being termed the total *available draft*. The general equation for this net total available draft intensity of a natural draft chimney with a circular section is as follows:

$$D_a = 2.96HB_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) - \frac{0.00126W^2T_c f L}{D^5 B_o W_c} \quad (1)$$

where

D_a = available draft, inches of water.

H = height of chimney above grate bars, feet.

B_o = barometric pressure corresponding to altitude, inches of mercury.

W_o = unit weight of a cubic foot of air at 0 F and sea level atmospheric pressure, pounds per cubic foot.

W_c = unit weight of a cubic foot of chimney gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.

T_o = absolute temperature of atmosphere, degrees Fahrenheit.

T_c = absolute temperature of chimney gases, degrees Fahrenheit.

W = amount of gases generated in the combustion chamber of the boiler and passing through the chimney, pounds per second.

f = coefficient of friction.

L = length of friction duct of the chimney, feet.

D = minimum diameter of chimney, feet.

The first term of the right hand expression of Equation 1 represents the theoretical draft intensity, and the second term, the loss due to friction.

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. of mercury; 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 1 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. 3 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 intensity 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 draft-capacity curve corresponds to the head-capacity curve of centrifugal pump characteristics and the dynamic-head-capacity curve of a fan. 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 1 and then plotting the results in the manner shown in Fig. 3.

In substituting the values for the various factors in Equation 1, care should be exercised that the selections be as near the actual conditions as is practically possible. The following notes will serve as a guide for these selections:

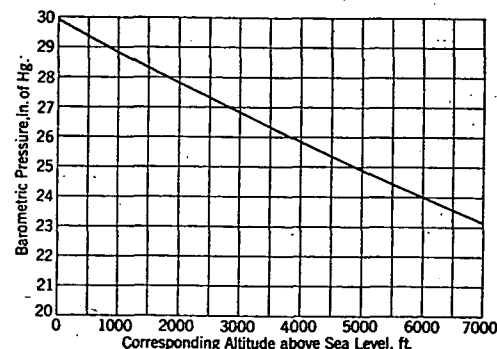


FIG. 4. RELATION BETWEEN BAROMETRIC PRESSURE AND ALTITUDE

1. The *barometric pressure* varies inversely as the altitude of the plant above sea level. Fig. 4 gives the barometric pressure corresponding to various elevations as computed from the equation:

$$E_1 = 62,737 \log_{10} \frac{29.92}{B_o} \quad (2)$$

where

E_1 = altitude of plant above sea level, feet.

In general, the barometric pressure decreases approximately 0.1 in. of mercury 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:

$$W_c = 0.131CO_2 + 0.095O_2 + 0.083N_2 \quad (3)$$

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.

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* does not vary appreciably from the gas temperature as it leaves the breeching and enters the chimney. For average operating conditions, the chimney gas temperature will vary between 500 F and 650 F except in the case when economizers and recuperators are used, when the temperature will vary between 300 F and 450 F. If a chimney has been properly constructed, properly lined and has no air infiltration due to open joints, the temperature of the gases throughout the chimney will not differ appreciably from the foregoing figures. In most up-to-date heating plants, the temperature may be read from instruments or ascertained from a pyrometer. The

analysis of this section is predicated on the assumption of constant gas temperature and no air infiltration throughout the height of the chimney.

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

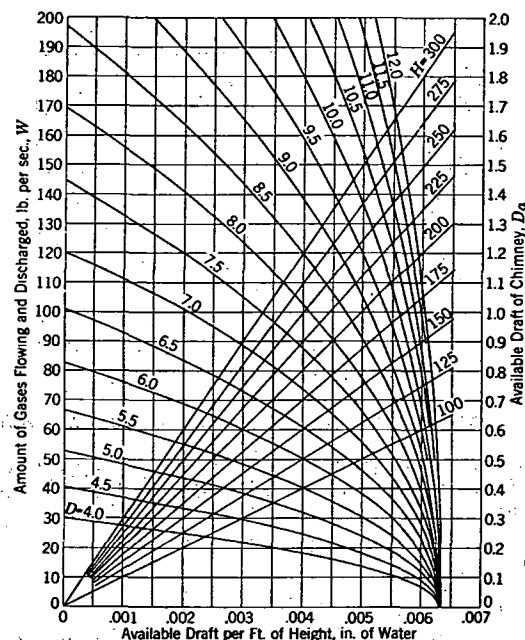


FIG. 5. CHIMNEY PERFORMANCE CHART^a

^aTo 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 reader, this constant value of 0.016 has been employed in the development of the various special equations and charts shown in this chapter.

However, much to be recommended as an alternate method is the practise of separately determining duct friction factors as a function of the flow conditions, specifically as a function of the Reynolds number and the relative duct roughness. The Reynolds criterion is based on the physical properties of the gas, the duct dimensions, and the gas velocity. The gas velocity for a chimney is usually well above the critical velocity. It is likely that this procedure of using a separately determined variable friction factor for chimney flow will give results that are to be preferred over those based on a set constant.

6. The length of the friction duct is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

7. Assuming no air infiltration the amount of gases flowing and being discharged is, of course, equal to the amount of gases generated in the combustion chamber of the boiler. The total products of combustion in pounds per second for a grate fired boiler may be computed from the equation:

$$W = \frac{C_g G W_{tp}}{3600} \quad (4)$$

where

C_g = pounds of fuel burned per square foot of grate surface per hour.

G = total grate surface of boilers, square feet.

$C_g \times G$ = total weight of fuel burned per hour.

W_{tp} = total weight of products of combustion per pound of fuel.

A similar computation may be made in the case of gas, oil, or stoker-fired fuel.

Fig. 5 is a typical chimney performance chart giving the available draft intensities for various amounts of gases flowing 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 operating conditions, a new chart should be constructed from Equation 1.

It has been the usual custom, and still is to a lamentably great extent, to select the required size of a natural draft chimney from a table of chimney sizes based only on boiler horsepower. After the ultimate horsepower of the projected plant had been determined, the chimney size in the table corresponding to this figure was then selected as the proper size required. Generally, no further attempt was made to determine if the height thus selected was sufficient to help create the required draft demanded by the entire installation, or the diameter sufficiently large to enable the chimney quickly, efficiently, and economically to dispose of the gases. Since the operating characteristics of a natural draft chimney are similar in all respects to those of a centrifugal pump, or a centrifugal fan, it is no more possible to select a proper size chimney from such a table, even with correction factors appended, than it is to select the proper size pump from tables based only on the amount of water to be delivered.

DETERMINING CHIMNEY SIZES

The required diameter and height of a natural draft chimney are given by the following equations:

$$H = \frac{D_r}{2.96 B_o \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) - \frac{0.184 f W_c B_o V^2}{T_c D}} \quad (5)$$

$$D = 0.288 \sqrt{\frac{W T_c}{B_o W_c V}} \quad (6)$$

where

H = required height of chimney above grate bar level, feet.

D = required minimum diameter of chimney, feet (constant for entire height).

V = chimney gas velocity, feet per second.
 D_r = total required draft demanded by the entire installation outside of the chimney, inches of water.

Equations 5 and 6 give the required size of a natural draft chimney with all of the operating factors taken into consideration. Values for all of the factors with the exception of the chimney gas velocity may be either observed or computed. It is, of course, necessary to assume an arbitrary value for the velocity in order to arrive at some definite size. For any one set of operating conditions there will be as many sizes of chimneys as there are values of reasonable velocities to assume. Of the number of sizes corresponding to the various assumed velocities, there is one size which will cost least. Since the cost of a chimney structure, regardless of the kind of material used in the construction, varies as the volume of material in the structure, the cost criterion then may be represented by the approximate equation:

$$Q = \pi t H D \quad (7)$$

where

Q = volume of material, cubic feet.
 t = average wall thickness, feet.

For all practical purposes, the value of πt may be taken as a constant regardless of the size of the structure. Hence, in general, the volume, and consequently the cost, of a chimney structure may be based on the factor HD as a criterion. Therefore, the value of the chimney gas velocity which will result in the least value of HD for any one set of operating conditions will produce a structure which will be the most economical to use, because its cost will be least.

The problem at hand is to deduce an equation for the chimney gas velocity which will result in a combination of a height and a diameter whose product HD will be least. The solution is obtained by equating the product of Equations 5 and 6 to HD , differentiating this product with respect to V and equating the resulting expression to zero. This procedure results in the following expression:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{W T_c}{B_o W_c}}}{f W_c} \right)^{2/5} \quad (8)$$

where V_e = economical chimney gas velocity, feet per second.

Equation 8 gives the economical velocity of the chimney gases for any set of operating conditions, and represents the velocity which will result in a chimney the size of which will cost less than that of any other size as determined by any other velocity for the same operating conditions. After the value of the economical velocity has been determined, the corresponding height and diameter can then be determined from Equations 5 and 6, respectively, and the economical size will then be attained. Equations 5, 6 and 8 may be simplified considerably for average operating conditions in an average size steam plant by assuming typical conditions.

Average chimney gas temperature, 500 F..... $T_c = 960$
 Mean atmospheric temperature, 62 F..... $T_o = 522$
 Average coefficient of friction, 0.016..... $f = 0.016$
 Average chimney gas density, 0.09..... $W_c = 0.09$
 Sea level elevation, with barometer of 29.92..... $B_o = 29.92$

Substituting these values in Equations 8, 6 and 5, respectively, and reducing, the results are substantially:

$$V_e = 13.7 W^{1/5} \quad (9)$$

$$D = 1.5 W^{2/5} \quad (10)$$

$$H = 190 D_r \quad (11)$$

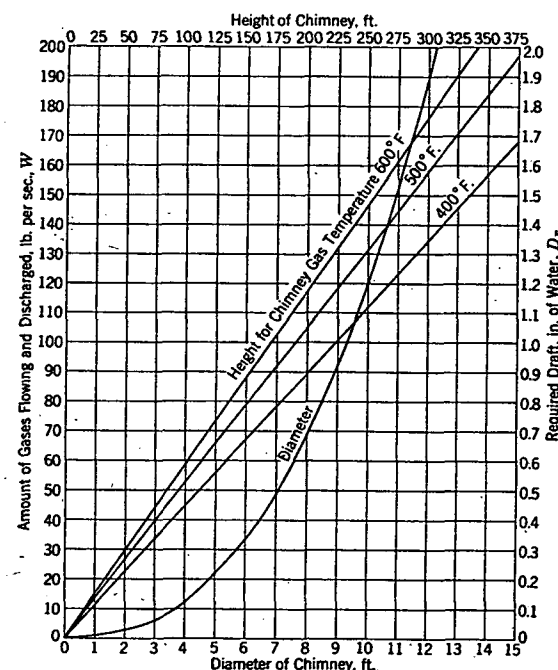


FIG. 6. ECONOMICAL CHIMNEY SIZES^a

^aDiameter values also for gas temperatures of 400, 500 and 600 F.

Fig. 6 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities as computed from Equations 9, 10 and 11. They are based on the operating factors used in reducing Equations 5, 6 and 8 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations 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 5,

6 and 8. The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

GENERAL EQUATION

The general draft equation for a steam producing plant may be stated as follows:

$$D_t - h_f = h_F + h_B + h_{Bd} + h_C + h_{Br} + h_V + h_O + h_E + h_R \quad (12)$$

where

- D_t = theoretical draft intensity created by pressure transformer, inches of water.
 h_f = draft loss due to friction in pressure transformer, inches of water.
 h_F = draft loss through the fuel bed, inches of water.
 h_B = draft loss through the boiler and setting, inches of water.
 h_{Br} = draft loss through the breeching, inches of water.
 h_V = draft loss due to velocity, inches of water.
 h_{Bd} = draft loss due to bends, inches of water.
 h_C = draft loss due to contraction of opening, inches of water.
 h_O = draft loss due to enlargement of opening, inches of water.
 h_E = draft loss through the economizer, inches of water.
 h_R = draft loss through recuperators, regenerators, or air heaters, inches of water.

The left hand member of Equation 12 represents the total amount of available draft created by the pressure transformer, that is, the natural draft chimney, Venturi chimney, or fan, and is equal to the theoretical intensity less the internal losses incidental to operation. The right hand member represents the sum of all of the various losses of draft throughout the entire boiler plant installation outside of the pressure transformer itself. The left hand member expresses the available intensity and is analogous to the head developed by a centrifugal pump in a water works system, while the right hand member expresses the required draft intensity and is analogous to the total dynamic head in a water works system. For a general circulation of gases

$$D_a = D_r \quad (13)$$

where

- D_a = available draft intensity, inches of water.
 D_r = required draft, inches of water.

The draft loss through the fuel bed (h_F), or the amount of draft required to effect a given or required rate of combustion, varies between wide limits and represents the greater portion of the required draft. In coal-fired installations, the draft loss through the fuel bed is dependent upon the following factors: (1) character and condition of the fuel, clean or dirty; (2) percentage of ash in the fuel; (3) volume of interstices in the fuel bed, coarseness of fuel; (4) thickness of the fuel bed, rate of combustion; (5) type of grate or stoker used; (6) efficiency of combustion.

There is a certain intensity of draft with which the best results will be obtained for every kind of coal and rate of combustion. Fig. 7 gives the intensity of draft, or the vacuum in the combustion chamber required to burn various kinds of coal at various rates of combustion. Expressed in

TABLE 1. RECOMMENDED MINIMUM CHIMNEY SIZES FOR HEATING BOILERS AND FURNACES^a

WARM AIR FURNACE CAPACITY IN SQ IN. OF LEADER PIPE	STEAM BOILER CAPACITY SQ FT. OF RADI- ATION	HOT WATER HEATER CAPACITY SQ FT. OF RADI- ATION	NOMINAL DIMEN- SIONS OF FIRE CLAY LINING IN INCHES	RECTANGULAR FLUE		ROUND FLUE		HEIGHT IN FT. ABOVE GRATE
				Actual Inside Dimensions of Fire Clay Lining in Inches	Actual Area Sq. In.	Inside Diam- eter of Lining in Inches	Actual Area Sq. In.	
790	590	973	8½ x 13	7 x 11½	81	10	79	35
1000	690	1,140	13 x 13	11¼ x 11¼	127			
	900	1,490	8½ x 18	6¾ x 16¼	110	12	113	40
	1,100	1,820	13 x 18	11¼ x 16¼	183	15	177	
	1,700	2,800	18 x 18	15¾ x 15¾	248			45
	1,940	3,200	20 x 20	17¼ x 17¼	298	18	254	50
	2,130	3,520				20	314	
	2,480	4,090	20 x 24	17 x 21	357			55
	3,150	5,200	24 x 24	21 x 21	441			60
	4,300	7,100		24 x 24 ^b	576	22	380	
	4,600	7,590				24	452	65
	5,000	8,250						
	5,570	9,190				27	573	
	5,580	9,200						
	6,980	11,500						
	7,270	12,000		24 x 28 ^b	672			
	8,700	14,400		28 x 28 ^b	784			
	9,380	15,500						
	10,150	16,750		30 x 30 ^b	900			
	10,470	17,250		28 x 32 ^b	896			

^aThis table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

^bDimensions are for unlined rectangular flues.

other words, these curves represent the amount of draft required to force the necessary amount of air through the fuel bed in order to effect various rates of combustion. It will be noted that the amount of draft increases as the percentage of volatile matter diminishes, being comparatively low for the lower grades of bituminous coals and highest for the high grades and small sizes of anthracites. Also, when the interstices of the coal are large and the particles are not well broken up, as with bituminous coals, much less draft is required than when the particles are small and are well broken up, as with bituminous slack and the small sizes of anthracites. In general, the draft loss through the fuel bed increases as: (1) the percentage of volatile matter diminishes; (2) the percentage of fixed carbon increases; (3) the thickness of the bed increases; (4) the percentage of ash increases; (5) the volume of the interstices diminishes.

In making the preliminary assumptions for the draft loss through the fuel bed, due allowances should be made for a possible future change in the grade of fuel to be burned and also in the rate of combustion. A value should be selected for this loss which will represent not only the highest rate of combustion which will be encountered, but also the grade of coal which has the greatest resistance through the fuel bed and which may be burned at a later date.

In powdered-fuel and oil-fired installations, there will be no draft loss

through the fuel bed since there is none and, consequently, this factor becomes zero in the general draft equation. All other factors being constant, the height of the chimney in installations of this character will be less than the height in coal-fired installations, and in the case of mechanical draft installations the driving units need not be as large since the head against which the fan is to operate is not as great in the former as in the latter.

The draft loss through the boiler and setting (h_B) also varies between wide limits and, in general, depends upon the following factors:

1. Type of boiler.
2. Size of boiler.
3. Rate of operation.
4. Arrangement of tubes.
5. Arrangement of baffles.
6. Type of grate.
7. Design of brickwork setting.
8. Excess air admitted.
9. Location of entrance into breeching.

Curves showing the draft loss through the boiler are usually based on the load or quantity of gases passing through the boiler, expressed in

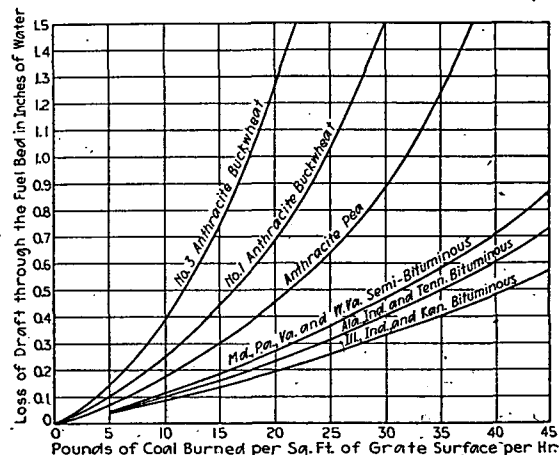


FIG. 7. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

terms of percentage of normal rate of operation. Owing to the great variety of boilers of different designs and the various schemes of baffling, it is impossible to group together a set of curves for the draft loss through the boiler which may even be used generally. It is therefore necessary to secure this information from the manufacturer of the particular type of boiler and baffle arrangement under consideration.

When a boiler is installed and in operation, the draft loss depends upon the amount of gases flowing through it. This, in turn, depends upon the proportion of excess air admitted for combustion. Primarily, the amount of excess air is measured by the CO_2 content; the less the amount of CO_2 , the greater the amount of excess air and hence the greater the draft loss.

The loss of draft through the boiler will vary directly as the size of the boiler and the length of the gas passages within. The loss also varies as the number of tubes high, but not in a direct ratio inasmuch as the loss due to the reversal of flow at the ends of the baffles remains constant regardless of the height of the boiler. The arrangement of the tubes, whether the gases flow parallel to or at right angles to the tubes, has an appreciable effect on the loss. The arrangement of the baffles influences the draft loss greatly, the loss through a boiler with five passes being greater than the loss through one of three or four passes. A poor design and a rough condition of the brickwork will increase the loss greatly, whereas a proper design and a smooth condition will keep the loss at a minimum. The loss through the boiler will be less when the breeching entrance is located at or near the top of the boiler than when it is located at or near the bottom since the gases have a shorter distance to travel in the former instance.

The draft loss through the breeching (h_{Br}) is given by the general equation:

$$h_{Br} = \frac{0.000194W^2T_c f L}{A^2 B_o W_c C_{br}} \quad (14)$$

where

- W = the amount of gases flowing, pounds per second.
 T_c = absolute temperature of breeching gases, degrees Fahrenheit.
 f = coefficient of friction.
 L = length of breeching, feet.
 A = area of breeching, square feet.
 B_o = atmospheric pressure corresponding to altitude, inches of mercury.
 W_c = weight of a cubic foot of breeching gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
 C_{br} = hydraulic radius of breeching section.

It has been the general custom to *lump off* the intensity of the breeching loss at 0.10 in. of water per 100 ft of breeching length regardless of its size or shape or the amount and temperature of the gases flowing through it. This practice is hazardous and has no more foundation in fact than that of determining the friction head in a water works system without taking into consideration the size of the pipe or the amount of water flowing through it. When the length of the breeching is relatively short, any variation in any one of the factors in the equation will have no appreciable effect on the draft loss. However, when the breeching is relatively long, the draft loss is affected greatly by the various factors, particularly by the size and shape as well as by the weight of gases flowing.

The draft loss due to velocity (h_v) is given by the equation

$$h_v = \frac{0.000194W^2T_c}{A^2 B_o W_c} \quad (15)$$

and represents the amount of draft required to accelerate the gases from zero velocity to the velocity at which the gases are flowing, or in other words, from a static gas condition of zero flow to the amount of gases

flowing throughout the installation. This loss corresponds to the velocity head in water works systems.

The *draft loss due to bends* (h_{Bd}) is equivalent to the loss due to the velocity head for a 90-deg bend. In changing direction of flow, the gas velocity decreases to zero with a loss of velocity head and then increases to its proper value at the expense of a loss in pressure head, the net result being a loss in pressure head equal to the velocity head at the bend. This loss is given by the equation:

$$h_{Bd} = \frac{0.000194 W^2 T_c}{A^2 B_o W_c} \quad (16)$$

The friction at a right-angle bend is sometimes expressed as the equivalent of a straight length of flue of a certain length for a certain diameter, similar to the procedure used in estimating the loss due to bends in piping systems conducting water. Most flues, however, particularly breechings, are built square or rectangular in section and no general equation based on the shape of the flue can be conveniently expressed.

The *draft loss due to sudden contraction of an area* (h_c) is given by the equation:

$$h_c = \frac{0.000194 K_c W^2 T_c}{A_s^2 B_o W_c} \quad (17)$$

where

K_c = coefficient of sudden contraction based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section = $0.5 \left(1 - \frac{A_s}{A_1} \right)$

A_s = area of the smaller section.

When the flue or passage through which the gases flow is suddenly contracted, a considerable portion of the static head in the larger section is converted into velocity head and a draft loss of some consequence, particularly in a short breeching, takes place. A sudden contraction should always be avoided where possible. At times, however, due to obstructions or limited head-room, it is necessary to alter the size of the breeching, but a sudden contraction may be avoided by gradually decreasing the area over a length of several feet.

The *draft loss due to a sudden enlargement of an area* (h_o) is given by the equation:

$$h_o = \frac{0.000194 K_o W^2 T_c}{A_s^2 B_o W_c} \quad (18)$$

where

K_o = coefficient of sudden enlargement based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section = $\left(1 - \frac{A_s}{A_1} \right)^2$

When the flue or passage through which the gases flow is suddenly enlarged, a portion of the velocity head is converted into static head in the larger section and, like the loss due to sudden contraction, a loss of some

consequence, particularly in short breechings, takes place. A sudden enlargement in a breeching may be avoided by gradually increasing the area over a length of several feet. In large masonry chimneys, the area of the flue at the region of the breeching entrance is considerably larger than the area of the breeching at the chimney, and a sudden enlargement exists.

The *draft loss through the economizer* (h_E) should be obtained from the manufacturer but for general purposes it may be computed from the following general equation:

$$h_E = \frac{6.6 W_n^2 N T_c}{10^{12}} \quad (19)$$

where

W_n = pounds of gases flowing per hour per linear foot of pipe in each economizer section.

N = number of economizer sections.

An economizer in a steam plant affects the draft in two ways, (1) it offers a resistance to the flow of gases, and (2) it lowers the average chimney gas temperature, thereby decreasing the available intensity. In the case of a natural draft installation, both of these factors result in a relative increase in the height of the chimney and, in the case of a large plant, they may add as much as 20 or 30 ft to the height. The decrease in the temperature of the gases after they have passed through the economizer has an extremely important effect on the performance of a natural draft chimney and also upon the performance of a fan.

CONSTRUCTION DETAILS

For general data on the construction of chimneys reference should be made to the Standard Ordinance for Chimney Construction of the *National Board of Fire Underwriters*. Briefly summarized, these provisions are as follows for heating boilers and furnaces:

The construction, location, height and area of the chimney to which a heating boiler or warm-air furnace is connected affect the operation of the entire heating system. Most residence chimneys are built of brick and may be either lined or unlined, but in either case the *walls must be air-tight* and there should be *only one smoke opening* into the chimney. Cleanout, if provided, must be *absolutely air-tight* when closed.

The walls of brick chimneys shall be not less than $3\frac{3}{4}$ in. thick (width of a standard size brick) and shall be lined with fire-clay flue lining. Fire-clay flue linings shall be manufactured from suitable refractory clay, either natural or compounded, and shall be adapted to withstand high temperatures and the action of flue gases. They shall be of standard commercial thickness, but not less than $\frac{3}{4}$ in. All fire-clay flue linings shall meet the standard specification of the *Eastern Clay Products Association*. The flue sections shall be set in special mortar, and shall have the joints struck smooth on the inside. The masonry shall be built around each section of lining as it is placed, and all spaces between masonry and linings shall be completely filled with mortar. No broken flue lining shall be used. Flue lining shall start at least 4 in. below the bottom of smoke-pipe intakes of flues, and shall be continued the entire heights of the flues and project at least 4 in. above the chimney top to allow for a 2 in. projection of lining. The wash or splay shall be formed of a rich cement mortar. To improve the draft the wash surface should be concave wherever practical.

Flue lining may be omitted in brick chimneys, provided the walls of the chimneys

are not less than 8 in. thick, and that the inner course shall be a refractory clay brick. All brickwork shall be laid in spread mortar, with all joints push-filled. Exposed joints both inside and outside shall be struck smooth. No plaster lining shall be permitted.

Chimneys shall extend at least 3 ft above flat roofs and 2 ft above the ridges of peak roofs when such flat roofs or peaks are within 30 ft of the chimney. The chimney shall be high enough so that the wind from any direction shall not strike the top of the chimney from an angle above the horizontal. The chimney shall be properly capped with stone, terra cotta, concrete, cast-iron, or other approved material; but no such cap or coping shall decrease the flue area.

There shall be but one connection to the flue to which the boiler or furnace smoke-pipe is attached. The boiler or furnace smoke-pipe shall be thoroughly grouted into the chimney and shall not project beyond the inner surface of the flue lining.

The size or area of flue lining or of brick flue for warm-air furnaces depends on height of chimney and capacity of heating system. For chimneys not less than 35 ft in height above grate line, the net internal dimensions of lining should be at least $7 \times 11\frac{1}{2}$ in. for a total leader pipe area up to 790 sq in. Above 790 and up to 1,000 sq in. of leader pipe area the lining should be at least $11\frac{1}{4} \times 11\frac{1}{4}$ in. inside. In case of brick flues not less than 35 ft in height with no linings, the internal dimensions should be at least 8×12 in. up to 790 sq in. of leader area, and at least 12×12 in. for leader capacities up to 1,000 sq in. Chimneys under 35 ft in height are unsatisfactory in operation and hence should be avoided.

CHIMNEYS FOR GAS HEATING

The burning of gas differs from the burning of coal in that the force which supplies the air for combustion of the gas comes largely from the pressure of the gas in the supply pipe, whereas air is supplied to a bed of burning coal by the force of the chimney draft. If, with a coal-burning boiler, the draft is poor, or if the chimney is stopped, the fire is smothered and the combustion rate reduced. In a gas boiler or furnace such a condition would interfere with the combustion of the gas, but the gas would continue to pass to the burners and the resulting incomplete combustion would produce a dangerous condition. In order to prevent incomplete combustion from insufficient draft, all gas-fired boilers and furnaces should have a back-draft diverter in the flue connection to the chimney.

A study of a typical *back-draft diverter* shows that partial or complete chimney stoppage will merely cause some of the products of combustion to be vented out into the boiler room, but will not interfere with combustion. In fact, gas-designed appliances must perform safely under such a condition to be approved by the *American Gas Association* Laboratory. Other functions of the back-draft diverter are to protect the burner and pilot from the effects of down-drafts, and to neutralize the effects of variable chimney drafts, thus maintaining the appliance efficiency at a substantially constant value. Converted boilers or furnaces, as well as gas-designed appliances, should be provided with back-draft diverters.

Since back-draft diverters have a special function to perform in protecting gas burning appliances, it is necessary that they should be built to the proper size as shown in Table 2. Work is now in progress on the development of a horizontal diverter for use where there is not enough room to install a vertical type of diverter. Information on the approved proportion of such equipment may be secured from the *American Gas Association* Testing Laboratory.

As is the case with the complete combustion of almost all fuels, the

TABLE 2. SUGGESTED GENERAL DIMENSIONS FOR VERTICAL BACK-DRAFT DIVERTER

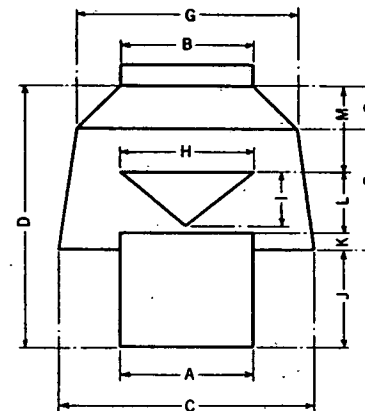


Table of Dimensions (In.)

PIPE SIZE	A	B	C	D	E	F	G	H	I	J	K	L	M
3	3	3	5.5	7.0	3.8	0.7	4.4	3.0	1.5	2.5	0.7	1.5	2.3
4	4	4	7.2	9.5	5.0	1.0	6.0	4.0	2.0	3.5	1.0	2.0	3.0
5	5	5	9.4	10.8	5.3	1.5	8.0	5.0	2.3	4.0	0.9	2.4	3.5
6	6	6	11.5	12.0	5.6	1.9	9.8	6.0	2.5	4.5	0.8	2.7	4.0
7	7	7	13.5	13.9	6.4	2.3	11.6	7.0	2.9	5.3	0.9	3.1	4.6
8	8	8	15.5	15.8	7.1	2.7	13.4	8.0	3.2	6.0	1.0	3.5	5.3
9	9	9	17.5	17.5	7.7	3.1	15.2	9.0	3.5	6.7	1.0	4.0	5.8
10	10	10	19.7	18.8	7.9	3.6	17.2	10.0	3.8	7.3	1.0	4.3	6.2
11	11	11	22.2	20.7	8.4	4.3	19.6	11.0	4.1	8.0	1.5	4.6	6.6
12	12	12	24.7	22.2	8.7	5.0	22.0	12.0	4.4	8.5	1.7	5.0	7.0

products of combustion for gas are carbon dioxide (CO_2) and water vapor with just a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron oxide catalyst. The volume of water vapor in the flue products is about twice the volume of the carbon dioxide when coke oven or natural gas is burned. Because of the large quantity of water vapor which is formed by the burning of gas, it is quite important that all gas-fired central heating plants be connected to a chimney having a good draft. Lack of chimney draft causes stagnation of the products of combustion in the chimney and results in the condensation of a large amount of the water vapor. A good chimney draft draws air into the chimney through the openings in the back-draft diverter, lowers the dew point of the mixture, and reduces the tendency of the water vapor to condense.

The flue connections from a gas-fired boiler or furnace to the chimney should be of a non-corrosive material. In localities where the price of gas requires the use of highly efficient appliances, the material used for the flue connection not only should be resistant to the corrosion of water,

TABLE 3. MINIMUM ROUND CHIMNEY DIAMETERS FOR GAS APPLIANCES (INCHES)

HEIGHT OF CHIMNEY FEET	GAS CONSUMPTION IN THOUSANDS OF BTU PER HOUR								
	100	200	300	400	500	750	1000	1500	2000
20	4.50	5.70	6.60	7.30	8.00	9.40	10.50	12.35	13.85
40	4.25	5.50	6.40	7.10	7.80	9.15	10.25	12.10	13.55
60	4.10	5.35	6.20	6.90	7.60	8.90	10.00	11.85	13.25
80	4.00	5.20	6.00	6.70	7.35	8.65	9.75	11.50	12.85
100	3.90	5.00	5.90	6.50	7.20	8.40	9.40	11.00	12.40

but should resist the corrosion of dilute solutions of sulphur trioxide in water. Sheet aluminum, as well as some other materials, seems to serve this purpose very well.

When condensation in a chimney proves troublesome, it may be necessary to provide a drain to a dry well or sewer. The cause of the excessive condensation should be investigated and remedied if possible. This may be done by raising the flue temperature slightly or increasing the size of the back-draft diverter. 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 provides an excellent protection.

A chimney for a gas-fired boiler or furnace should be constructed in accordance with the principles applicable to other boilers. Where the wall forming a smoke flue is made up of less than an 8-in. thickness of brick, concrete, or stone, a burnt fire clay flue tile lining should be used. Care should be used that the lengths of flue tile meet properly with no openings at the joints. Cement mortar should be used for the entire chimney.

Table 3 gives the minimum cross-sectional diameters of round chimneys (in inches) for various amounts of heat supplied to the appliance, and for various chimney heights. This is in accordance with *American Gas Association* recommendations.

PROBLEMS IN PRACTICE

1 ● What are the principle factors influencing the intensity of natural draft?

The intensity of natural draft depends largely upon the height of chimney above the grate bar level and the temperature difference between the chimney gases and the atmosphere.

2 ● What two kinds of draft need be considered?

Natural draft caused by temperature differences, and artificial draft caused by mechanical forcing.

3 ● What is the effective height of a chimney?

The height from the grate level to the top of the chimney is the effective height in producing natural draft.

4 ● What dual purpose does a tall chimney fulfill?

A tall chimney primarily creates the necessary draft to move the air required for the combustion process and to move the products of combustion, and secondarily it discharges the gases at a high elevation to prevent them from becoming a nuisance.

5 ● a. Name the principal advantages of natural draft.

b. Name the principal disadvantages of natural draft.

- Simplicity, reliability, freedom from mechanical parts, low cost of maintenance, relatively long life, relatively low depreciation, operation with no power requirement.
- Lack of flexibility, irregularity, dependence on surroundings, susceptibility to temperature changes.

6 ● How is mechanical draft created?

By forced draft, by induced-draft fans, or by a Venturi chimney.

7 ● Distinguish between theoretical and available draft.

Theoretical draft is the difference in pressure inside and outside the base of a chimney when it is under operating temperatures but when there are no gases flowing. Available draft is less than theoretical draft by the friction loss due to the flow of gases through the chimney.

8 ● Explain the term efficiency of a natural draft chimney.

The efficiency of a chimney is the ratio of the work it does in moving gases to the theoretical amount of power it generates.

9 ● How is the available draft used in a heating plant?

The available draft at the base of the chimney is used to overcome the loss in pressure through the grate, the fuel bed, the boiler passes, the breeching.

10 ● What are some of the factors that influence the draft loss through the fuel bed?

Uniformity and size of coal, the amount of ash mixed with the fuel on the grate, thickness of fuel bed, rate of combustion, amount of air supply as related to the coal burning rate.

11 ● How does the volatile matter content affect the draft loss through the fuel bed?

The higher the volatile content and the lower the fixed carbon content, the lower the draft loss.

12 ● In what cases will there be no fuel bed draft loss?

In oil, gas, and powdered fuel firing the fuel is mixed and burned in suspension; consequently, no measurable resistance is encountered in the combustion zone.

13 ● Is it possible to state an average value for the draft loss through a boiler and its setting?

No. The draft loss varies widely and depends on many factors such as the size and type of gas passageways. The manufacturer is usually able to supply such information.

14 ● Of what significance is the CO₂ content of stack gases in establishing draft loss?

The CO₂ content of the exit gases is a measure of the completeness of the combustion and the amount of excess air supplied. Low CO₂ indicates a high excess of air and hence a high draft loss.

15 • What two effects does an economizer have on the draft loss?

An economizer offers resistance to the flow of gases over the added surfaces; it lowers the temperature of the gases going to the chimney and therefore decreases the available draft. This decrease often necessitates the addition of forced draft.

16 • What main provisions should be considered in good chimney construction?

Chimneys should be air-tight and connected to only one smoke opening. The chimney top should be high enough above surroundings so the wind will not strike it at any angle above the horizontal. Chimney walls should be not less than one brick in width, and they should be lined with fire clay tile of the size required for the attached heating unit. Tile lining sizes are stated as outside dimensions; therefore, their effective dimensions are less by the thickness of the wall.

17 • What is the purpose of a back draft diverter as used on gas burning units?

Since the fuel is supplied under pressure independent of draft it is necessary to free the unit from the variable chimney draft and to supply air for combustion in direct proportion to the supply of fuel gas. The back draft diverter protects the pilot and burners from down drafts.

Chapter 27

FUELS AND COMBUSTION

Classification of Coal, Air for Combustion, Draft Required, Combustion of Anthracite, Firing Bituminous Coal, Burning Coke, Hand Firing, Classification and Use of Oil, Classification and Use of Gas

THE choice of fuel for heating is a question of economy, cleanliness, fuel availability, operation requirements, and control. The principal fuels to be considered are coal, oil, and gas.

COAL

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 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 *U. S. Bureau of Mines Bulletin 276*.

A classification of coals is given in Table 1, and a brief description of the kinds of fuels 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 very 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 little attention to the fuel beds between firings. It is capable of giving a high efficiency in the common types of hand-fired furnaces.

Semi-anthracite has a higher volatile content than anthracite, it is not as 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 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 ash pit continue to burn. Very little smoke or soot is formed.

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.

TABLE 1. CLASSIFICATION OF COALS BY RANK¹

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-agglutinating*
	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	1. Low volatile bituminous coal	Dry F.C., 77 per cent or more and less than 86 per cent (Dry V.M., 23 per cent or less and more than 14 per cent)	Either agglutinating or non-weathering*
	2. Medium volatile bituminous coal	Dry F.C., 69 per cent or more and less than 77 per cent (Dry V.M., 31 per cent or less and more than 23 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 ² Btu, 14,000 ³ or more	
	4. High volatile B bituminous coal	Moist ² Btu, 13,000 or more and less than 14,000 ³	
	5. High volatile C bituminous coal	Moist Btu, 11,000 or more and less than 13,000 ³	
III. Sub-bituminous	1. Sub-bituminous A coal	Moist Btu, 11,000 or more and less than 13,000 ³	Both weathering and non-agglutinating
	2. Sub-bituminous B coal	Moist Btu, 9500 or more and less than 11,000 ³	
	3. Sub-bituminous C coal	Moist Btu 8300 or more and less than 9500 ³	
IV. Lignite	1. Lignite	Moist Btu less than 8300	Consolidated Unconsolidated
	2. Brown coal	Moist Btu less than 8300	

*If agglutinating, classify in low-volatile group of the bituminous class.

²Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

³Pending the report of the Subcommittee on Origin and Composition and Methods of Analysis, it is recognized that there may be non-caking varieties in each group of the bituminous class.

⁴Coals 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.

⁵There are three varieties of coal in the High-volatile C bituminous coal group, namely, Variety 1, agglutinating and non-weathering; Variety 2, agglutinating and weathering; Variety 3, non-agglutinating and non-weathering.

⁶Adapted from A.S.T.M. Standards on Coal and Coke, p. 68, American Society for Testing Materials, Philadelphia, 1934.

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.

The sale of *petroleum cokes* for domestic furnaces has been small and is generally confined to the Middle West. They 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.

In order to obtain perfect combustion a definite amount of air is required for each pound of fuel fired. 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.

Total Air Required. The theoretical amount of air required per pound of fuel for perfect combustion is dependent upon the analysis of the fuel;

TABLE 2. POUNDS OF AIR PER POUND OF FUEL AS FIRED

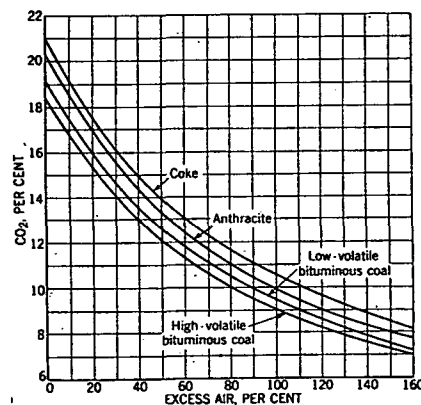
ANTHRACITE	COKE	SEMI-BITUMINOUS	BITUMINOUS	LIGNITE
9.6	11.2	11.2	10.3	6.2

however, for estimating purposes the theoretical air required for different grades of fuel may roughly be taken from Table 2. An excess of about 50 per cent over the theoretical amount is considered good practice under usual operating conditions.

The amount of excess air, based upon the laws of combustion, can be determined by its relation to the percentage of CO_2 (carbon dioxide) in the products of combustion. This relationship is shown by the curves (Fig. 1) for high and low volatile coals and for coke. 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.

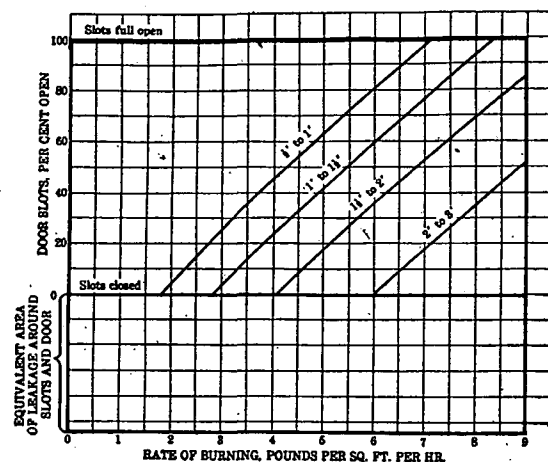
Secondary Air. The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size of fuel, depth of fuel bed, and diameter of fire pot. The ratio of the secondary to the primary air increases with decrease in the size of the fuel pieces, with increase in the depth of the fuel bed, and with increase in the area of the fire pot; the ratio also increases with increase in rate of burning.

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

FIG. 1. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

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. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets 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



From U. S. Bureau of Mines.

FIG. 2. RELATIVE AMOUNT OF FIRE DOOR SLOT OPENING REQUIRED IN A GIVEN FURNACE TO GIVE EQUALLY GOOD COMBUSTION FOR HIGH TEMPERATURE COKE OF VARIOUS SIZES WHEN BURNED AT VARIOUS RATES

size of slot opening is dependent on whether the ashpit damper is open or closed. It is better to have too much than too little secondary air; the opening is too small if there is a puff of flame when the firing door is opened.

Fig. 2, taken from the *U. S. Bureau of Mines Report of Investigations No. 2980*, shows the relationship of the slot opening, for a domestic furnace, to the size of coke and the rate of burning; these openings are with the ashpit damper wide open, and would be less if the available draft permits of its being partly closed. The same openings are satisfactory for anthracite.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing in order to consume 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. The following suggestions will be helpful:

1. In cold weather, with high combustion rates, the secondary air damper should be half open all the time.
2. In very mild weather, with a very low combustion rate, the secondary air damper should be closed all the time.
3. For temperatures between very mild and very cold, the secondary air damper should be in an intermediate position.
4. For ordinary house operation, secondary air is needed after each firing for about one hour.

Draft Required

The draft required to effect a given rate of burning the fuel as measured at the smokehood is dependent on the following factors:

1. Kind and size of fuel.
2. Combustion rate per square foot of grate area per hour.
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 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, and it is often possible to produce a higher rate by increasing the thickness of the fuel bed.

Combustion of Anthracite¹

An anthracite fire should never be poked, as this serves to bring ash to the surface of the fuel bed where it melts into clinker.

Egg size is suitable for large firepots (grates 24 in. and over) if the fuel

¹See reports published by *The Anthracite Institute Laboratory*, Primos, Pennsylvania.

can be fired at least 16 in. deep. The air spaces between the pieces of coal are large, and 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 only instructions needed for burning this type of fuel are that the grate should be shaken daily, the fire should never be poked or disturbed, and the fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for firepots 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. It is relatively low in price. When fired carefully, pea coal can be burned on standard grates. It is well to have a small amount of a larger fuel on hand when building new fires, or when filling holes in the fuel bed. 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. This keeps a bed of ignited coal in readiness against the time when a sudden demand for heat shall be made on the heater.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open, the cold-air check closed, and by controlling the fire with the air-inlet damper only. Pea size can also be fired in layers with stove or egg size anthracite and its use in this manner will reduce the fuel costs and attention required.

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 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 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 an explosion of gas within the firepot, 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. A good draft is required and consequently the fire is best controlled by the air-inlet damper only. 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 special grates or retorts frequently are used with this fuel for best results.

No. 2 buckwheat anthracite, or rice size, is used only with forced draft equipment on mechanical stokers. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

Firing Bituminous Coal

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.

Air should be admitted over the fire through a special secondary air device, or through a slide in the fire door or by opening the fire door slightly. If the quantity of air admitted is too great the gases will be cooled below the ignition temperature and will fail to burn. The fireman can judge the quantity of air to admit by noting when the air supplied is just sufficient to make the gases burn rapidly and smokelessly above the fuel bed.

The red fuel in the firebox, before firing, excepting only a shallow layer of coke on the grate, should be pushed to one side or forward or backward to form a hollow in which to throw the fresh fuel. Some manufacturers recommend that all red fuel be pushed to the rear of the firebox and that the fresh fuel be fired directly on the grate and allowed to ignite from the top. The object of this is to reduce the early rapid distillation of gases and to reduce the quantity of secondary air required for smokeless combustion.

It is well to have the bright fuel in the firebox so placed that the gases from the freshly fired fuel, mixed with the air over the fuel bed, pass over the bed of bright fuel on the way to the flues. The bed of bright fuel then supplies the heat to raise the mixture of air and gas to the ignition temperature, thereby causing the gaseous matter to burn and preventing the formation of smoke.

The fuel bed should be carried as deep as the size of fuel and the available draft permit, in order to have as much coked fuel as possible for pushing to the rear of the firebox at the time of firing. A deep fuel bed allows the longest firing intervals.

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 firebox. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

The output obtained from any heater with bituminous coal will usually exceed that obtainable with anthracite, since soft coal burns more rapidly than hard coal and with less draft. Soft coal, however, will require frequent attention to the fuel bed, because it burns unevenly, even though the fuel bed may be level, forming holes in the fire which admit too much air, chilling the gases over the fuel bed and reducing the available draft.

Semi-bituminous coal is fired as bituminous coal, and because of its caking characteristics it requires practically the same attention. 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.

Burning Coke

Coke is a very desirable fuel and usually will give satisfaction as soon as the user learns how to control the fire. Coke ignites and burns very rapidly with less draft than anthracite coal. 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 more rapidly than an anthracite fire 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. 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. Since coke weighs only about half as much as anthracite per cubic foot only about half as much can be put in the firepot, so it will be necessary to fire oftener. The best size of coke for general use, for small firepots 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 firepots 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.

Dustless Coal

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with a solution of calcium chloride or a mixture of calcium and magnesium chlorides. Both these salts are very hygroscopic and their moisture under normal atmospheric conditions keeps the surface of the coal damp, thus reducing the dust during delivery and in the cellar, and obviating the necessity of sprinkling the coal in the bin.

The coal is sometimes treated at the mine, but more usually by the local distributor just before delivery. The solution is 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.

Pulverized Coal

Installations of pulverized coal burning plants in heating boilers are of the unit type, in which the pulverized coal is delivered into the furnace immediately after grinding, together with the proper amount of preheated

air. With this apparatus, where the necessary furnace volume is obtainable, high efficiencies can be obtained.

A 150-hp boiler has generally been considered the smallest size for which pulverized fuel is feasible. Complications are introduced if an installation with a single boiler has to take care of very light loads.

Hand Firing

Hand firing is the oldest and the most widely used method of burning coal for heating purposes. To keep the fuel bed in proper condition where hand firing is used, the following general rules should be observed:

1. Remove ash from fuel bed by shaking the grates whenever fresh fuel is fired. This removes ash from the fire, enables the air to reach the fuel, and does away with the formation of clinker which is melted ash.
2. Supply the boiler with a deep bed of fuel. Nothing is gained by attempting to fire a small amount of fuel. A deep bed of fuel secures the most economical results.
3. Remove ash from ashpit at least once daily. Never allow ash to accumulate up to the grates. If the ash prevents the air from passing through, the grate bars will burn out and much clinker trouble will be experienced.

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and 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.

OIL

Uniform oil specifications were prepared in 1929 by the *American Oil Burner Association*, in cooperation with the *American Petroleum Institute*, the U. S. Bureau of Standards, the *American Society for Testing Materials* and other interested organizations. Oil fuels were classified into six groups, as indicated by Table 3. When these specifications were prepared, it was generally accepted that the first three grades were adapted to domestic use, while the last three were suitable only for commercial and industrial burners.

Today domestic installations may use No. 4 oil of the so-called heavy oil group, when and if said oil very closely follows the specifications of No. 3. Up-to-date listing by the *Underwriter's Laboratories* should be referred to before a No. 4 grade of fuel is used which merely meets *Commercial Standards CS 12-35*.

Since the specifications as originally drawn provide for maximum limits only for the several grades, this differentiation has not proved stable. Realizing how unsatisfactory it is to have specifications which permit the substitution of one grade for another, the U. S. Bureau of Standards in cooperation with the *American Society for Testing Materials* is figuring

TABLE 3. COMMERCIAL STANDARD FUEL OIL SPECIFICATIONS^a

A. Detailed Requirements for Domestic Fuel Oils

GRADE OF OIL	APPROX. BTU PER GAL. ^b	FLASH POINT		WATER AND SEDIMENT, MAXIMUM	POUR POINT, ^c MAXIMUM	DISTILLATION TEST		VISCOSITY MAXIMUM
		Min.	Max.					
No. 1 Domestic Fuel Oil A light distillate oil for use in burners requiring a high grade fuel.	139,000	110 F or legal	165 F	0.05%	15 F	10% point, maximum 420 F	End point, maximum 600 F	
No. 2 Domestic Fuel Oil A medium distillate oil for use in burners requiring a high grade fuel.	141,000	125 F or legal	190 F	0.05%	15 F	10% point, maximum 440 F	90% point, maximum 620 F	
No. 3 Domestic Fuel Oil A distillate fuel oil for use in burners where a low viscosity oil is required.	143,400	150 F or legal	200 F	0.1%	15 F	10% point, maximum 460 F	90% point, maximum 675 F	Saybolt Universal at 100 F 55 seconds

B. Detailed Requirements for Industrial Fuel Oils

GRADE OF OIL	APPROX. BTU PER GAL. ^b	FLASH POINT, MIN. MAX.	WATER AND SEDIMENT, MAXIMUM	POUR POINT, ^c MAXIMUM	VISCOSITY, MAXIMUM
No. 4 Industrial Fuel Oil An oil known to the trade as a light fuel oil for use in burners where a low viscosity industrial fuel oil is required.	144,500	150 F. See Note ^d	1.0%	See Note ^e	Saybolt Universal at 100 F 125 seconds
No. 5 Industrial Fuel Oil Same as Federal Specifications Board specification for bunker oil "B" for burners adapted to the use of industrial fuel oil of medium viscosity.	146,000	150 F	1.0%		Saybolt Furol at 122 F 100 seconds
No. 6 Industrial Fuel Oil Same as Federal Specifications Board specification for bunker oil "C" for burners adapted to oil of high viscosity.	150,000	150 F	Water sediment 1.75% 0.25%		Saybolt Furol at 122 F 300 seconds

^aAdapted from "Fuel Oils," p. 2, U. S. Department of Commerce, Bureau of Standards, Commercial Standard CS12-33, Washington, 1933.

^bGovernment specifications do not give Btu per gallon, but they are noted here for information only.

^cLower or higher pour points may be specified whenever required by conditions of storage and use. However, these specifications shall not require a pour point less than 0 F under any conditions.

^dWhenever required, as for example in burners with automatic ignition, a maximum flash point may be specified. However, these specifications shall not require a flash point less than 250 F under any conditions.

^ePour point may be specified whenever required by conditions of storage and use. However, these specifications shall not require a pour point less than 15 F under any conditions.

on a new set of specifications providing for definite limits for each grade. When these specifications are adopted, it is expected that the *National Board of Fire Underwriters* will retest all burners using oils of the maximum specifications for the grade so that if a burner is approved for a certain grade it will burn any oil meeting the specifications for that particular grade.

Several burners adapted to industrial use have recently been listed for automatic operation with No. 5 oil. Usually oils No. 5 or 6 require preheating for proper operation, but where conditions are favorable, No. 5 can be used without the equipment that this entails.

There are two reasons for the trend to lower grades of oil. While the lighter oils contain slightly more heat units per pound, the weight per gallon increases more rapidly than the decrease in heat units per pound, and oil is bought by the gallon. As a consequence, while a No. 1 oil may contain 139,000 Btu per gallon, oil No. 5 may test 146,000 Btu per gallon, or 6 per cent more. Usually there is a differential of 3 to 4 cents between the No. 1 and No. 5 oils, so that the economy of buying the heavier fuels is apparent; there remains the economic utilization of the heat content of the heavier oils.

The cost of oil fuel is dependent also upon the amount that can be delivered at one time, and the method of delivery. Common practice has split the tank of the truck delivering oils for domestic use into compartments of 150 to 500-gal capacity, and these *unit dumps* are made the basis of price. Where a truck can be connected to a storage-tank *fill* and quickly discharge its oil by pump, the price obviously can be less than where a smaller quantity must be drawn off in 5-gal cans and poured. For similar reasons an installation that can be supplied from a tank car on a siding provides for a lower unit fuel cost than one where the oil must be trucked, even in the large trucks holding 2,000 gal or more that are used for distributing the heavier oils.

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, and is often designated as *city gas*.

When gas is burned a large amount of water vapor is produced as one of the products of combustion. This ordinarily escapes up the chimney, carrying away with it a certain amount of heat. However, when the heat value of gas is determined in an ordinary calorimeter, this water vapor is condensed and the latent heat of vaporization that is given up during the condensation is reported as a portion of the heat value of the gas. The heat value so determined is termed the *gross* or *higher* heat value and this is what is ordinarily meant when the heat value of gas is specified. The heat that is reclaimed by the condensation of the water vapor amounts to about 10 per cent of the total heat value. It is impractical to utilize the entire higher heat value of the gas in any house-heating appliance, because to do so it would be necessary to cool the products of

combustion down below their dew point, which is ordinarily in the neighborhood of 130 F.

The actual dew point in the chimney is different from the theoretical value because excess air is admitted not only at the burner but also at the backdraft diverter which lowers the dew point.

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 700 to 1,500 Btu per cubic foot, the majority of natural gases averaging about 1,000 Btu per cubic foot. Table 4 shows typical values for the three main oil fields, although values from any one field vary materially.

Table 4 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. There are limits to the

TABLE 4. 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, (DEG FAHR)
	High (Gross)	Low (Net)			Cubic Feet			ULTIMATE CO ₂ DRY BASIS	
					CO ₂	H ₂ O	Total with N ₂		
Natural gas—California	1200	1087	0.67	11.26	1.24	2.24	12.4	12.2	3610
Natural gas—Mid-Continental	967	873	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	1232	1120	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	575	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	588	521	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carburetted water gas	536	496	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	308	281	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	134	124	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

variation allowable, because the specific gravity of the gas depends on its composition, and too great a change in the specific gravity necessitates a change in the adjustment of the burners of small appliances.

Table 4 shows that a large proportion of the products of combustion when gas is burned may consist of water vapor, and that the greater the proportion of water vapor, the lower the maximum attainable CO_2 by gas analysis. The table also shows that a low calorific value does not necessarily mean a low flame temperature since, for example, natural gas has a theoretical flame temperature of 3600 F and blue water gas of 3800 F, although it has a calorific value less than one third that of natural gas.

The quantity of air given in Table 4 is that required for theoretical combustion, but with a properly designed and installed burner the excess air can be kept low. The division of the air into primary and secondary is a matter of burner design and the pressure of gas available, and also of the type of flame desired.

PROBLEMS IN PRACTICE

1 • Differentiate between the general characteristics of hard and soft coals.

Hard coals contain fixed carbon in large proportions and in addition more ash is present especially in the smaller sizes. Soft coals have an increasing percentage of carbon in combination with hydrogen which is volatile and will distill off under high temperature, producing smoke.

2 • Name several important properties of coal from a utilization standpoint.

- Caking tendency, whether none, weak, or strong.
- Quantity of volatile matter.
- Friability.
- Fusibility of the ash.

3 • What are the main data commonly available that fix the qualities of coal, and do these tell the whole story?

- Calorific value, Btu per pound.
- Proximate analysis giving percentages of moisture, volatile matter, fixed carbon, ash, and sulphur.
- Temperature at which the ash softens.
- Screen sizes.

Other important qualities not usually given are the friability of the coal, its caking tendency, and the qualities of the volatile matter. The percentage of ash and its fusion temperature do not tell how the ash is distributed or how much of it is less fusible lumps of slate or shale.

4 • Are there available complete and sufficient data on gas and oils to fix their burning properties and furnace requirements?

Yes. Because gas and oils are of simple and uniform composition, data are available to fix their burning properties and furnace requirements, but the ability to control their combustion is somewhat less determinable.

5 • What effect does moisture in fuels have on their efficiency?

With any solid fuel, latent and sensible heat are lost at the stack when moisture is dried out of the fuel in burning, and when its hydrogen is burned. Therefore, such fuels as sub-bituminous coal and lignite, which are high in moisture content, have a low efficiency. However, these efficiencies may be improved if the stack gases are cooled to room temperature, by heating the feed water, for example.

6 ● What are the advantages of a sized fuel for heating furnaces?

Because a sized fuel encourages a more uniform flow of air through the bed, the burning will be more uniform, and the bed will be less liable to develop holes and will require less attention. Uniformity of fuel size is more desirable as the area of the bed becomes smaller; it is less important with fuels that cake, but with sized fuels the caking will be more uniform and the air flow through the bed will be steadier. In addition, ash and pieces of slate are less likely to be segregated and to form lumps of clinker.

7 ● Does the size of a fuel affect the quantity of air required to burn it at a given rate?

The total air required to give the same gas analysis at the stack is independent of the size of the fuel burned, but for non-caking fuels the ratio of the air passing through the fuel bed to the total air entering the burner base decreases, for the same thickness of bed, as the size of the fuel becomes smaller; this decrease is very rapid for sizes less than one inch. For coals that cake, this ratio will depend on the way the caked bed is broken up and on the size of the resulting pieces.

8 ● Is the volatile matter which is given off when coals are burned of the same nature in all coals?

No. The 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-free and moisture-free coal, increasing amounts of oils and tars are given up. 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.

9 ● Is smoke a primary product in the burning of fuels?

Visible smoke may include very small particles of carbon, oil, tar, water (condensed steam), and ash. Of these, the oils, tars, and ash are mainly primary products, and the water is partly primary. The carbon, which usually comprises the greater part of the smoke, results from the breaking up by heat of oils, tars, and such gases as methane, so it may be considered a secondary product.

10 ● Is the sulphur in coals detrimental to combustion?

Not so far as is known, but its complete combustion gives only 25 per cent as much heat as is given by the same weight of carbon. Sulphur is undesirable because it causes corrosion of flues and stacks, and also because its gases pollute the atmosphere, and damage buildings and vegetation.

11 ● How do deposits of soot on the surfaces of a boiler or heater affect the quantity of fuel burned?

There are two effects. The soot 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 shows that the loss of seasonal efficiency is not as great as has been believed and should not be over 6 per cent because the greater part of the heat is transmitted through the fire pot. The soot clogs the passages and reduces the draft; the loss of efficiency from this action may be much more, and also the lack of draft results in unsatisfactory heating.

Chapter 28

AUTOMATIC FUEL BURNING EQUIPMENT

Residential Stokers, Apartment House Stokers, Commercial Stokers, Domestic Oil Burners, Commercial Oil Burners, Gas-Fired Appliances, Gas Boilers, Warm Air Furnaces, Space Heaters, Conversion Burners, Gas Appliances

AUTOMATIC, mechanical equipment for the efficient combustion of coal, oil, and gas is considered in this chapter.

MECHANICAL STOKERS

Coal can be burned more efficiently on a mechanical stoker than by hand firing. The burning of coal involves uniformity of stoking, proper distribution over the fuel bed, admission of air as required to the fuel bed, and means for removing ash. The proper burning of the fuel on the grate is the function of the stoker and depends upon the stoker design.

The burning of the volatile gases above the fuel bed is a matter of furnace design. The requirements are the same regardless of the type of stoker. Proper care should be taken to provide furnaces sufficiently liberal in volume and with grates at a sufficient distance from the heating surface in order to permit proper combustion of gases. The standards that have been most universally adopted for the proportioning of furnaces are those of the *Midwest Stoker Association*.

Stokers may be divided into four types according to their construction, namely, (1) overfeed flat grate, (2) overfeed inclined grate, (3) underfeed side cleaning type, and (4) underfeed rear cleaning type. They may also be classified according to their uses. The following classification has been recommended by the *Stoker Manufacturers Association*.

Class 1. Up to 60 lb coal feed per hour (Household).

Class 2. 60 to 500 lb coal feed per hour (Apartment house and small commercial).

Class 3. 500 to 1200 lb coal feed per hour and less than 36 sq ft of grate area (General commercial heating and small high pressure steam plants).

Class 4. Over 1200 lb coal feed per hour and over 36 sq ft grate area (Large commercial and high pressure steam plants).

Overfeed Flat Grate Stokers

This type is represented by the various chain- or traveling-grate stokers. These stokers receive fuel at the front of the grate in a layer of uniform thickness and move it back horizontally to the rear of the furnace. Air is

supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze and also for bituminous coals, the characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker requires an arch over the front of the stoker to maintain ignition of the incoming fuel. Frequently, a rear combustion arch is required to maintain ignition until the fuel is fully consumed. A typical traveling-grate stoker is illustrated in Fig. 1.

Another and distinct type of overfeed flat-grate stoker is the spreader or sprinkler type in which coal is distributed either mechanically or by air over the entire grate surface. This type of stoker has a wide application

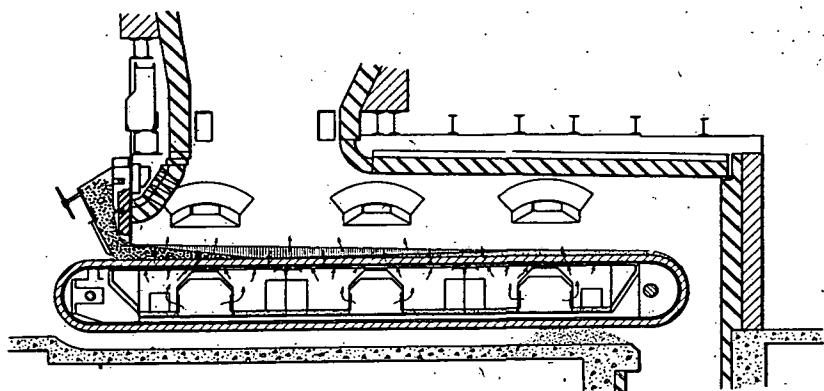


FIG. 1. OVERFEED TRAVELING-GRATE STOKER

on small sized fuels and on certain special fuels such as lignites, high-ash coals, and coke breeze.

Overfeed Inclined Grate Stokers

In general the combustion principle is similar to the flat grate stoker, but this stoker (Fig. 2) is provided with rocking grates set on an incline to advance the fuel during combustion. Also this type is provided with an ash plate where ash is accumulated and from which it is 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 has the tendency to keep the fuel bed well broken up thereby allowing for free passage of air. Because of its agitating effect on the fuel it is not so desirable for badly clinkering coals. Furthermore, it should usually be provided with a front arch to care for the volatile gas.

Underfeed Side Cleaning Stokers

In this type (Fig. 3), the fuel is fed in at the front of the furnace to one or more retorts, 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 coking coals while in the smaller sizes it is suitable for small sizes of anthracites. 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 gas is released, is mixed with air, and passes through the fire

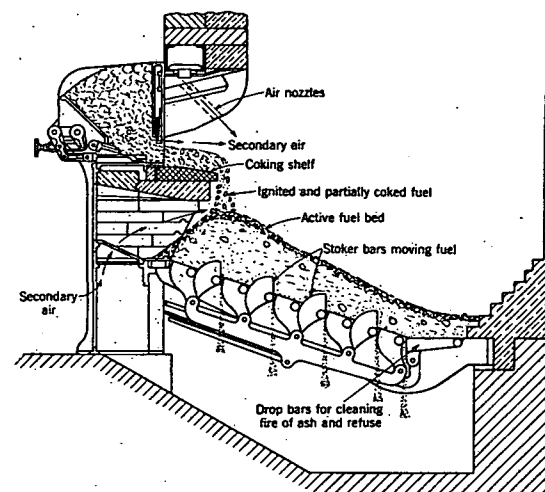


FIG. 2. OVERFEED INCLINED GRATE STOKER

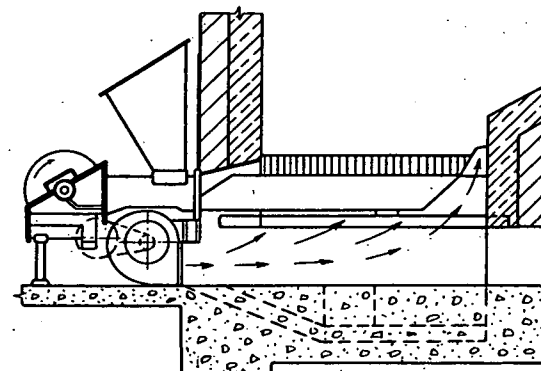


FIG. 3. UNDERFEED PLUNGER TYPE STOKER

where it is burned. The ash may be continuously discharged as in the small stoker or may be accumulated on a dump plate and periodically discharged. This stoker requires no arch as it automatically provides for the combustion of the volatile gas.

Underfeed Rear Cleaning Stokers

This type of stoker carries on 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.

Class 1 Stokers, Household

A common type of stoker in this class consists of a round retort having tuyeres at the top where all of the air for combustion is admitted. Coal is fed from a storage hopper (Fig. 4) outside of the boiler by means of a

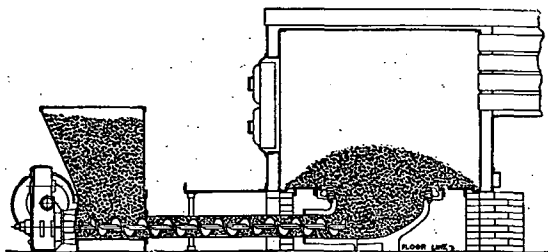


FIG. 4. UNDERFEED SCREW STOKER, HOPPER TYPE

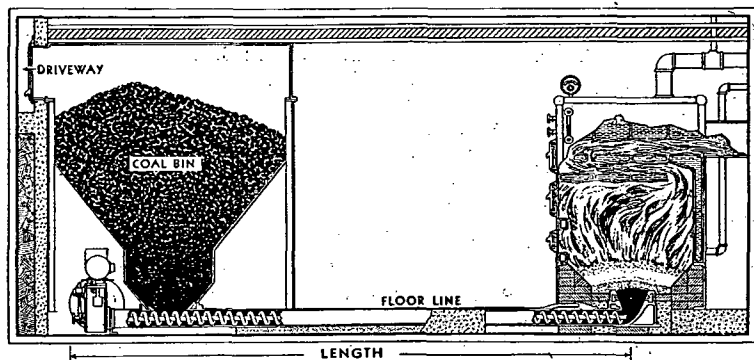


FIG. 5. UNDERFEED SCREW STOKER, BIN TYPE

worm into the bottom of this retort and beneath the fire. The equipment includes a blower which is driven by the same motor that drives the stoker.

Some household stokers are provided with automatic grate-shaking mechanism together with screw conveyers for removing the ash from the ashpit and depositing it in an ash receptacle outside the boiler. Certain types can also be provided with a coal conveyer which takes coal from the storage bin and maintains a full hopper at the stoker. In some cases the stoker hopper functions as the coal bin as shown in Fig. 5, and an extended worm is used to convey the fuel to the boiler. They may feed coal to the furnace either intermittently or with a continuous flow regulated auto-

matically to suit conditions. Where the boiler is provided with indirect coils for heating the domestic hot water, the stoker may be arranged so that it can be used the entire year to maintain a continuous hot water supply.

Household stokers are made for all classes of fuel—anthracite, bituminous and semi-bituminous. The *United States Department of Commerce* has issued commercial standards for household anthracite burners, which may be secured by application. The requirements stated in these standards are described in the following paragraphs.

Operating Requirements

Efficiency. The over-all efficiency of the unit at all points above 50 per cent of maximum coal feed shall be above 50 per cent when installed in a round sectional cast iron boiler having three intermediate sections and 1½ in. of asbestos insulation or its equivalent in good condition of repair, operating at 50 per cent or more of the boiler capacity. The efficiency shall be maintained for any continuous period of 4 hours during any test or observation run.

Ash Loss. Combustible in ash shall not exceed 7.5 per cent of the Btu content of the coal as fired at any rate of coal feed above 50 per cent of maximum. Methods of test according to Code No. 3 of the A.S.H.V.E.¹ are to be followed in all details applicable to stoker testing.

Clinker. Ash removing systems should at all times be capable of disposing of any clinker which may be formed under any conditions of operation with the coals prescribed.

Combustion Rate. A combustion rate of at least 13 lb per square foot of horizontal projected area of ash ring per hour must be continuously maintained for at least 9 hours with the above conditions of efficiency, ash and clinker.

Flue Gas. Flue gas shall be not below 6 per cent in carbon dioxide with a reasonably tight boiler at any rate of operation above 50 per cent of maximum coal feed.

Maximum Rating. The maximum rating, in terms of gross square feet of water or steam radiation which the burner will supply, when intended for installation in the average existing cast-iron boiler, shall be 90 per cent of the maximum steam produced in a round cast-iron boiler in good repair having three intermediate sections and the equivalent of 1½ in. of asbestos insulation. However, in no case shall the maximum rating be greater than 29 sq ft of direct steam radiation for each pound of coal fired, per hour, and in no case shall ratings be based upon efficiency figures below 50 per cent.

The maximum rating as defined in the preceding paragraph shall be based upon combustion of Pennsylvania anthracite having the following approximate analysis:

Volatile matter 3.5 to 9 per cent; ash content not to exceed 15 per cent; sulphur content under 1.5 per cent; ash fusing temperature 2750 F, or above (volatile, ash and sulphur content on dry basis in accordance with A.S.T.M. method D271-33); Btu content 12,000 or above; properly sized as follows: A No. 1 buckwheat should pass through a round mesh screen having ⅜ in. holes and over a similar screen having ⅝ in. holes. The undersizing should not exceed 15 per cent and the oversizing should not exceed 10 per cent. A No. 2 buckwheat (rice) should pass through a round mesh screen having holes ⅜ in. in diameter and over a like screen having holes of ⅝ in. in diameter. The undersizing should not exceed 15 per cent and the oversizing should not exceed 10 per cent.

Coal Storage. It is recommended that the coal bin or closet be constructed so as to be dustproof.

Electrical Consumption. The electrical consumption shall not exceed 18 kw-hr per 2000 lb of coal burned at any rate of coal feed above 50 per cent of the maximum.

Operation Upon Other Sizes of Coal. The foregoing specifications have been drafted for operating with the Nos. 1 and 2 buckwheat sizes of anthracite. In the event that other sizes are recommended, ratings shall be based upon the same efficiency and ash loss requirements.

¹A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3), (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

Banking. The burner shall be so constructed or controlled as to maintain a fire during an indefinite banking period.

Acceleration. When the burner resumes operation after a 12 hour banking period, the time required for the stack temperature to reach a normal maximum shall not exceed 60 min.

Class 2 Stokers, Apartment House, Small Commercial

This class is used extensively for heating plants in apartments and hotels, and also for small industrial plants such as laundries, bakeries, and creameries. The majority of stokers used in this field are of the underfeed type. The principal exception is an overfeed type having step action grates in a horizontal plane and so arranged that they are alternately moving and stationary, and are designed to advance the fuel during combustion to an ash plate at the rear.

All of the stokers are provided with a coal hopper outside of the boiler. In the underfeed types, the coal feed from this hopper to the furnace may be accomplished by a continuously revolving worm or by an intermittent plunger. The drive for the coal feed may be an electric motor, or a steam or hydraulic cylinder. With an electric motor, the connection between the driver and the coal feed may be through a variable speed gear train which provides two or more speeds for the coal feed; or it may be through a simple gear train and a variable speed driver for the change in speed of the coal feed; or a simple gear train with a coal feed having an adjustment for varying the travel of the feeding device. With a steam or hydraulic cylinder, the power piston is connected directly to the coal feeding plunger.

The stokers in this class vary also in their retort design according to the fuels and load conditions. The retort is placed approximately in the middle of the furnace and is provided with tuyere openings at the top on all sides. In the plunger-feed type the retort extends from the inside of the front wall entirely to the rear wall or to within a short distance of the rear wall. This type of retort has tuyeres on the sides and at the rear.

These stokers also differ in the grate surface surrounding the retort. In many of the worm-feed stokers this grate is entirely a dead plate on which the fuel rests while combustion is completed. In the dead-plate type, all of the air for combustion is furnished by the tuyeres at the retort. Because of this, combustion is well advanced over the retort so that it may easily be completed by the air which percolates through the fuel bed. With the dead-plate type of grate the ash is removed through the fire doors and it is therefore desirable that the fuel used shall be one in which the ash is readily reduced to a clinker at the furnace temperature, in order that it may be removed with the least disturbance of the fuel bed.

In other stokers in this class, the grates outside of the retort are air-admitting and some stokers have shaking grates. These grates permit a large part of the ash to be shaken into the ash pit beneath, while the clinkers are removed through the fire doors. With this type of grate, the main air chamber extends only under the retort while the side grates receive air by natural draft from the ash pit.

In still other stokers of this class, the main air chamber extends beyond the retort and is covered with fuel-bearing, air-supplying grates. With this type of grate, the fuel is supplied with air from the main air chamber

throughout combustion. Also with this type of grate, dump plates are provided beyond the grates where the ash accumulates and from which it can be dropped periodically into the ash pit beneath.

Stokers in this class are compactly built in order that they may fit into standard heating boilers and still leave room for sufficient combustion space above the grates. The height of the grate is approximately the same as that of the ordinary grates of boilers, so that it is usually possible to install such stokers with but minor changes in the existing equipment. In some districts, there are statutory regulations governing such settings.

These stokers vary in furnace dimensions from 30 in. square to approximately 66 in. square. The capacity of the stokers is measured by the amount of coal that can be burned per hour. In general, manufacturers recommend that, for continuous operation, the coal burning rate shall not exceed 25 lb of coal per square foot of grate per hour, while for short peaks this rate may be increased to 30 lb per hour. Although these stokers were designed to burn bituminous coal, types are available for the semi-bituminous coals such as Pocahontas and New River. They can also be used to burn the small sizes of anthracite but at a somewhat

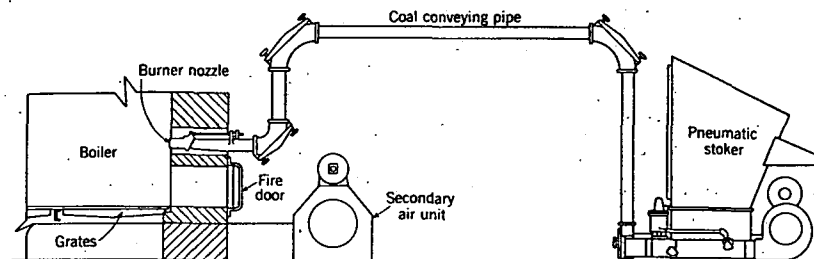


FIG. 6. PNEUMATIC STOKER WITH CONVEYING PIPE

lower rate. It is often customary to have the janitor or some other attendant care for the boiler as one of his duties. Under these conditions the heating plant does not receive the same careful attention as it would if a man devoted his entire attention to the fire. With periodic hand-firing, the boiler is operated inefficiently much of the time. With a stoker, the boiler is operated at the rate that the conditions require so long as there is coal in the hopper. With hand firing, it is customary to use a more expensive size of fuel, while with a stoker the smaller sizes are used at a considerable saving in the cost per ton. Because the stoker responds promptly to automatic regulation, it is possible to maintain a reasonably constant standard. Also because the stoker feeds the fuel regularly and in small quantities without losses due to opening doors, it is more efficient than hand firing. This increase in efficiency depends entirely on conditions, with a minimum of about 10 per cent and a maximum of about 25 per cent.

Another type of stoker which may be used in connection with small size coal is a pneumatic type as shown in Fig. 6. The equipment may be arranged with a pipe conveying the fuel from a storage space directly to the burner nozzle with secondary air supplied from a separate unit located near the boiler.

Class 3 Stokers, General Commercial, Small High Pressure Plants

The general commercial heating and high pressure plant with boilers burning from 500 lbs of coal per hour to 1200 lbs of coal per hour, is a field distinct from the large commercial and industrial field. The prevalent type of stoker employed is the single retort underfeed side-cleaning type. In this field is found the greatest financial return on stoker investments due to the fact that 90 per cent of the fuel is purchased in the retail market where prices are on the average 40 per cent higher for the same fuel than if the coal were purchased in carload quantities. This is, therefore, an extremely important class of stokers. Savings of 40 per cent to 50 per cent are not at all unusual, and many plants show as much as a 50 per cent return on the investment.

Automatic Stoker Controls

The industry developed by stokers in Classes 1, 2 and 3 has been due as much to the application of proper controls as to the stoker itself, as the types of stokers used in these classes are not basically new, while the industry is distinctly new, originating in 1923. The usual controls applied are as follows:

- Thermostats (plain and clock).
- Limit Controls (steam, vapor, hot water, etc.).
- Stack Temperature or Time Controls (for actuating fires periodically).
- Relay (for low voltage controls).
- Safety or Overload Cutout (for protection against overload).

Class 4 Stokers, Large Commercial, High Pressure Steam Plants

This class includes stokers with grate areas above 36 sq ft and with hourly burning rates of over 1200 lbs of coal per hour. The prevalent stokers in this field are:

- Overfeed flat grate stokers.
- Overfeed inclined grate stokers.
- Underfeed side cleaning stokers.
- Underfeed rear cleaning stokers.

Overfeed inclined grate stokers are seldom built in sizes of over 500 hp and are not as extensively used as other types of stokers.

Underfeed side-cleaning stokers are made in sizes up to approximately 500 hp and in this field are extensively used. These stokers are not so varied in design as those in the smaller classes although the principle is much the same. Practically all of them are of the front coal feed type, either power driven or steam driven. Dump plates at the side are manually operated. These stokers are heavily built and designed to operate continuously at high boiler ratings with a minimum amount of attention. Because of the fact that all volatile gas must pass through the fire before reaching the combustion chamber, these stokers will operate smokelessly under ordinary conditions. Also because of the fact that these stokers are always provided with forced draft, they are the most desirable type for fluctuating loads or high boiler ratings.

In the design of the grates for supporting the fuel between the retort and the ash plates, the stokers differ in providing for movement of the

fuel during combustion. Some stokers are designed with fixed grates of sufficient angle to provide for this movement as the bed is agitated by the incoming fuel, while others have alternate moving and stationary bars in this area and provide for this movement mechanically. In either type, with proper operation, all refuse will be deposited at the dump plate. Recent developments in this type of stoker provide for sliding distributor blocks along the bottom of the retorts which give flexibility in providing proper distribution of fuel over the grate area and assist in preventing coke masses when strong coking coals are used. Another difference in these stokers is that some use a single air chamber under the whole grate area thus having the same air pressure under the ignition area as under the rest of the grate, while others have a divided air chamber using the full air pressure under the ignition area and a reduced air pressure under the remainder of the grate. These stokers vary in size from approximately 5 sq ft to a maximum of 8½ sq ft.

The most prevalent type of rear-cleaning underfeed stoker is the multiple retort design. Occasionally double or triple retort side-cleaning, underfeeds are made. The multiple retort underfeed stoker is made for the largest sizes of boilers for large industrial plants and central stations. This stoker has reached a very fine stage of development mechanically and in the matter of air supply and control. In some instances zoned air-control has been applied both longitudinally and transversely to the grate surface. Ash dumps on smaller sizes are sometimes manually operated,

TABLE 1. RECOMMENDED SETTING HEIGHTS FOR HEATING BOILERS EQUIPPED WITH MECHANICAL STOKERS^a

FIREBOX BOILERS									
Actual Load	2500	5000	7500	10000	12500	15000	20000	25000	30000
A	18"	18"	20"	20"	22"	22"	24"	24"	24"
B	42"	48"	54"	60"	66"	72"	78"	84"	84"

A = Distance from bottom of Water Leg to floor.

B = Distance from Crown Sheet to bottom of Water Leg.

COMPACT WELDED BOILERS									
Actual Load	2500	5000	7500	10000	12500	15000	20000	25000	30000
A	18"	18"	20"	20"	22"	22"	24"	24"	24"
B	30"	33"	36"	42"	45"	48"	54"	60"	60"

A = Distance from bottom of Water Leg to floor.

B = Distance from Crown Sheet to bottom of Water Leg.

H. R. T. BOILERS											
Hp.	50	75	100	125	150	175	200	225	250	275	300
A	5'-0"	5'-6"	6'-0"	6'-6"	7'-0"	7'-0"	7'-6"	8'-0"	8'-6"	9'-0"	9'-0"

A = Distance from bottom of shell to floor.

Hp = Installed horsepower.

In the case of the Firebox or Compact Welded type boilers the desired setting height can be obtained by combining A and B dimensions. The load ratings shown for this class of boilers are actual developed loads in square feet of equivalent cast iron steam radiation and are not manufacturers rating.

The setting heights given for H. R. T. boilers may be used for developed loads up to 50 per cent above normal rating.

^aFrom Data Prepared by the Midwest Stoker Association.

while larger sizes are power operated. The number of retorts and dimensions of furnaces are practically unlimited.

The V-type stoker is practically obsolete although many are still in operation. In this stoker, the grates are inclined downward from both sides of the furnace to a low point at the middle where there is either a dump plate for periodic disposal of the ash or a rotary ash grate for continuous discharge of ash. In this stoker, the fuel is fed into a hopper at the top of the grate on each side of the furnace and advanced down the grates to the center where the refuse is accumulated. This stoker is always provided with a combustion arch over the entire furnace for the purpose of assuring thorough combustion of the solid fuel and providing a furnace temperature sufficiently high to burn the volatile gases. Because of this high furnace temperature and because so little of the boiler surface is exposed to the fire to assist in carrying off the heat by radiation, this stoker is characterized by severe clinkering in the ash area. With all types of overfeed stokers, the most desirable installations are in boilers which are operated with comparatively uniform loads and moderate rates of combustion, since, even with good combustion arches, fluctuating loads or high combustion rates result in smoke.

Table 1 gives recommended setting heights for heating boilers equipped with mechanical stokers.

DOMESTIC OIL BURNERS

An oil burner is a mechanical device for producing heat automatically and safely from liquid fuels. This heat is produced in the furnace or firepot of hot water or steam boilers or warm air furnaces and is absorbed by the boiler, and thus made available for distribution to the house through the heating system. Heat production is thus only one of many functions to be performed by a heating system.

Efficient heat production with any kind of fuel requires that all combustible matter in the fuel shall be completely consumed and that it shall be done with no more excess air than necessary. The combustion of oil is a rather rapid chemical reaction. Excess air provides an over supply of oxygen so that all of the oil, composed of carbon and hydrogen, will be completely oxidized and thus produce all the heat possible. The use of unnecessary excess air means no increase or decrease in the heat produced but it does mean that some air is needlessly heated and thrown away. This loss cannot be counteracted by any other part of the heating system and is therefore chargeable to the fuel-burning device.

Oil is a highly concentrated fuel composed exclusively of 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 it means that air must be supplied in carefully regulated quantities. The air and oil vapor must be vigorously mixed to get a rapid and complete chemical reaction. The better the mixing, the less excess air that will be needed. The combustion must take place in a space that maintains the temperatures high so the reaction will not be stopped before completion. When equipped with a means of igniting the oil and safety devices to guard against mishaps, the oil burner possesses all of the elements to be efficient and automatic.

The number of combinations of the characteristic elements of domestic oil burners is rather large and accounts for the variety of burners found in actual practice. Domestic oil burners may be classified as follows:

1. AIR SUPPLY FOR COMBUSTION

- a. *Atmospheric*—by natural chimney draft.
- b. *Mechanical*—electric-motor-driven fan or blower.
- c. *Combination of (a) and (b)*—primary air supply by fan or blower and secondary air supply by natural chimney draft.

2. METHOD OF OIL PREPARATION

- a. *Vaporizing*—oil distills on hot surface or in hot cracking chamber.
- b. *Atomizing*—oil broken up into minute globules.
 - (1) *Centrifugal*—by means of rotating cup or disc.
 - (2) *Pressure*—by means of forcing oil under pressure through a small nozzle or orifice.
 - (3) *Air or steam*—by high velocity air or steam jet in a special type of nozzle.
 - (4) *Combination air and pressure*—by air entrained with oil under pressure and forced through a nozzle.
- c. *Combination of (a) and (b)*.

3. TYPE OF FLAME

- a. *Luminous*—a relatively bright flame. An orange-colored flame is usually best if no smoke is present.
- b. *Non-luminous*—Bunsen-type flame (*i.e.*, blue flame).

4. METHODS OF IGNITION

- a. *Electric*.
 - (1) *Spark*—by transformer producing high-voltage sparks. Usually shielded to avoid radio interference. May take place continuously while the burner is operating or just at the beginning of operation.
 - (2) *Resistance*—by means of hot wires or plates.
- b. *Gas*.
 - (1) *Continuous*—pilot light of constant size.
 - (2) *Expanding*—size of pilot light expanded temporarily at the beginning of burner operation.
- c. *Combination*—electric sparks light the gas and the gas flame ignites the oil.
- d. *Manual*—by manually-operated gas torch for continuously operating burners.

5. MANNER OF OPERATION

- a. *On and off*—burner operates only a portion of the time (intermittent).
- b. *High and low*—burner operates continuously but varies from a high to a low flame.
- c. *Graduated*—burner operates continuously but flame is graduated according to needs by regulating both air and oil supply.

A trade classification of oil burners consists of the following general types: (a) gun, (b) rotary and (c) pot.

The gun type is characterized by an air tube, usually horizontal, with oil supply pipe centrally located in the tube and so arranged that a spray of atomized oil is introduced and mixed in the firepot with the air stream emerging from the air tube. A variety of patented shapes are 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. The most distinguishing feature of the rotary type is that the oil is discharged to the furnace or fire-pot by a rotating element of special design. The pot type can be identified by the presence of a metal structure, called a pot, in which combustion takes place. While fire brick linings in the boiler are necessary with the gun and rotary type, they are not needed with the pot type.

The oil burners are operated by a small electric motor which pumps the oil and some or all of the air required. The smallest sizes can generally burn not much less than $1\frac{1}{2}$ gal of oil per hour. The grade of oil burned ranges from No. 1 to No. 4 (see pg. 479 to 481). No. 4 oil is the heaviest and most viscous of the various grades mentioned. An oil burner satisfactory for No. 4 oil can burn any of the lighter grades easily but an oil burner recommended for No. 2 oil should never be supplied with the heavier grades. It has been found that while the heavier grades of oil have a smaller heat value per pound, they have, due to greater density, a larger heat value per gallon. The relative economy of the various grades must be based upon price and the amount of excess air required for clean and efficient combustion.

The Combustion Process

Efficient combustion as previously indicated must produce a clean flame and must use relatively small excess of air (*i.e.*, between 25 and 50 per cent). This can be done only by vaporizing the oil quickly, completely, and mixing it vigorously with air in a firepot hot enough to support the combustion. A vaporizing burner (*i.e.*, pot type) prepares the oil vapor before it mixes with air to any extent. If air and oil vapor temperatures are high and the firepot hot, a clear blue flame is produced. There may be a deficiency of air as shown by the presence of carbon monoxide (CO) or an excessive supply of air, depending upon burner adjustment, without altering the clean, blue appearance of the flame. An atomizing burner (*i.e.*, gun and rotary types) is so named because the oil in one way or another is mechanically separated into very fine particles so the surface exposure of the liquid to the radiant heat of the firepot is vastly increased and vaporization proceeds quickly. Since the air enters the firepot with the liquid fuel particles, it follows that mixing, vaporization and burning are all occurring at once in the same space. This produces a luminous instead of a blue or non-luminous flame. In this case a deficient amount of air is indicated by a dull red or dark orange flame with smoky flame tips.

An excessive supply of air may produce a brilliant white flame in some cases or, in others, a short ragged flame with incandescent sparks flashing through the combustion space. While extreme cases may be easily detected, it is generally not possible to distinguish, by the eye alone, the finer adjustments which competent installation requires.

Certain tests indicate that there is no difference in economy between a blue flame and a luminous flame if the position, shape and the per cent of excess air of both flames are about the same.

Furnace or Firepot Design

It is evident that the atomizing burner is dependent upon the surrounding heated refractory or firebrick surfaces to vaporize the oil and

support combustion. While the importance of the firepot is obvious, its design has been troublesome. Unsatisfactory combustion may be due to inadequate atomization and mixing. A firepot 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. Fundamentally, the firepot 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 firepot 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 firepot thus 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 quite hot. This insures quick vaporization, rapid combustion and better mixing by eliminating dead or inactive spaces in the firepot. An overhanging arch at the back of the firepot 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, firepot design becomes a less critical matter. Where secondary air is used, firepot design is quite important. Manufacturers generally provide careful directions and in some instances provide special firebrick shapes suited to their burners.

Oil and Air Adjustments

Where adjustments of oil and air have been made which give efficient combustion, the problem of maintaining the adjustments constant becomes an important one. Particularly is this true when the change causes the per cent of excess air to decrease below allowable limits of the burner. 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 (*i.e.*, 25 per cent excess air) the more critical it will be of variations. The oil and air supply rates must remain constant.

The following factors may influence the oil delivery rate: (a) changes in oil viscosity due to temperature change or variations in grade of oil delivered, (b) erosion of atomizing nozzle, (c) fluctuations in by-pass relief pressures and (d) possible variations in methods 2b (3) and 2b (4) listed in the previous classification table. Note that 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 but usually no soot will form.

The following factors may influence the air supply: (a) changes in firepot 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), (b) changes in air inlet adjustments to the fan—collection of lint and dirt on the inlet grille may be enough in some cases.

Measurement of the Efficiency of Combustion

Efficient combustion being based upon a clean flame and certain proportions of oil and air employed, it is possible to determine the results by analyzing the gases formed by the combustion process. An Orsat apparatus is a device which measures the volume of carbon dioxide (CO_2), oxygen (O_2) and carbon monoxide (CO) in the flue gases. Except in the case of a non-luminous flame 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 at the present time show from 8 to 10 per cent CO_2 . Taking into account the potential hazard of oil or air fluctuations with low excess air (high CO_2) a setting to give 10 per cent CO_2 constitutes a reasonable standard for the majority of oil burners.

Additional Design Considerations

Efficient combustion is found to be a question of good design plus a competent installation but it is not the sole feature of a completely satisfactory job. The following items should all be considered: (a) clean and efficient combustion, (b) a setting of the oil-burning rate to give the proper gross load, (c) a boiler to absorb the heat efficiently, (d) quiet operation, and (e) suitable control and safety devices.

The Gross Load

See Chapter 25 for general material on allowances, etc. To the design or heat loss of the house (Chapter 7), it is customary to add an allowance for piping and pick-up. The most common value for piping is 25 per cent and for pick-up 20 per cent. To these allowances should be added an allowance where domestic hot water is heated by the boiler. (See Chapter 35 for information on Domestic Hot Water). The design load plus the allowances indicates the gross load that should be produced by the boiler. While in every case the gross load will exceed the design load if adequate heating response is to be achieved, there is, however, no object whatever in over-estimating the allowances. The only effect would be to reduce the time of pick-up by a few minutes. Otherwise, it might mean forcing the boiler unduly and increasing the cost of operation.

Setting the Rate of Oil Burning

The rate of oil burning to get the gross output depends upon the combustion (*i.e.*, per cent CO_2) and the efficiency of the boiler in absorbing heat. The oil burning rate in conversion jobs—where a burner is placed in existing equipment—is troublesome to adjust accurately because the boiler efficiency is usually a matter of conjecture rather than actual knowledge. In general, each gallon of oil burned per hour will produce a boiler output of from 300 to 450 ft of steam radiation (see Fig. 7). The number of possible burners, adjustments, and boilers makes each installation a separate problem. Obviously, a satisfactory setting depends upon the knowledge, experience and judgment of the individual installer. It seems wise to set the oil-burning rate on the low rather than high side. The tendency has always been the other way. The oil-burning rate can easily be increased. If properly explained to the owner, he will appreciate the situation if subsequent adjustments are necessary.

Boiler-Burner Units

Boilers especially designed for oil burners are gradually becoming available to the purchaser of this type of equipment. They are used for replacements as well as for new installations. These boilers have more heating surface than the older coal-burning designs. Flue proportions and gas travel have been changed with beneficial results. All questions of firepot design, capacities, efficiencies, etc., have been determined. The selection of the proper size of unit should be a simple process.

Controls

Oil burner controls may be divided into two parts: (a) devices to regulate burner operation so the desired house heating result may be obtained and (b) devices for the safety and protection of the boiler and

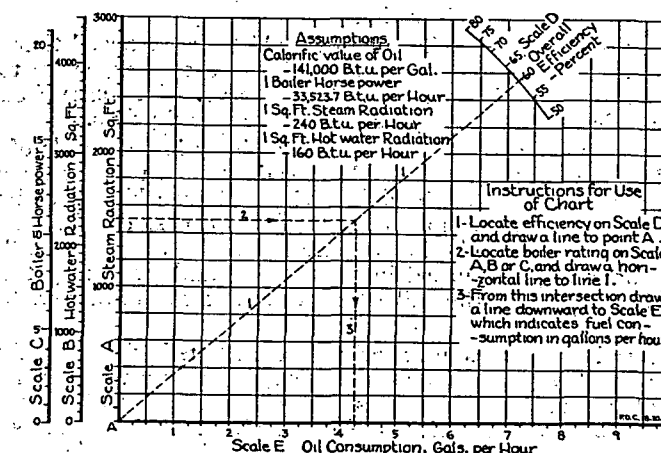


FIG. 7. FULL LOAD RATE OF OIL CONSUMPTION FOR HEATING BOILERS

burner. For control devices generally consult Chapter 14. The room thermostat has recently been improved to provide more frequent burner operation and greater uniformity of room temperature. Class (b) controls comprises a device to shut off the burner if the oil fails to ignite or if the flame should cease due to lack of oil; a device actuated by steam boiler pressure to shut off the burner when the pressure reaches some predetermined value; a device on the boiler to shut off the burner if the water level acts too low for safety or one which automatically feeds additional water to the boiler; a device on warm air furnaces to shut off the burner if the air temperature gets too high; a valve in the oil supply line which automatically closes in the event of fire in or near the cellar; and a device to keep the temperature of the boiler water within certain limits when it is being used to heat domestic hot water. These devices are all tested and approved by the Underwriters' Laboratory of Chicago, Ill., before they are offered to the purchaser. The selection of class (b) devices is made by the oil burner manufacturer.

Domestic Hot Water Supply

Provisions for heating domestic hot water in connection with automatic fuel-burning devices through heat exchangers attached to the boiler are fully described in Chapter 35.

COMMERCIAL OIL BURNERS

Liquid fuels are used for heating apartment buildings, hotels, public and office buildings, schools, churches, hospitals, department stores, as well as industrial plants of all kinds. Contrary to domestic heating, convenience seldom is a dominating factor, the actual net cost of heat production usually controlling the selection of fuel. Some of the largest office buildings have been using oil for many years. Many department stores have found that floor space in basements and sub-basements can be used to better advantage for merchandising wares, and credit the heat producing department with this saving.

Wherever possible, the boiler plant should be so arranged that either oil or solid fuel can be used at will, permitting the management to take advantage of changes in fuel costs if any occur. Each case should be considered solely in the light of local conditions and prices.

Burners for commercial heating may be either large models of types used in domestic heating, or special types developed to meet the conditions imposed by the boilers involved. Generally speaking, such burners are of the mechanical or pressure atomizing types, the former using rotating cups producing a horizontal torch-like flame. As much as 350 gal of oil per hour can be burned in these units, and frequently they are arranged in multiple on the boiler face, from two to five burners to each boiler.

The larger installations are nearly always started with a hand torch, and are manually controlled, but the use of automatic control is increasing, and completely automatic burners are now available to burn the two heaviest grades of oil. Nearly all of the smaller installations, in schools, churches, apartment houses and the like, are fully automatic.

Because of the viscosity of the heavier oils, it is customary to heat them before transferring by truck tank. It also has been common practice to preheat the oil between the storage tank and the burner, as an aid to movement of the oil as well as to atomization. This heating is accomplished by heat-transfer coils, using water or steam from the heating boiler, and heating the oil to within 30 deg of its flash point.

Unlike the domestic burner, units for large commercial applications frequently consist of atomizing nozzles or cups mounted on the boiler front with the necessary air regulators, the pumps for handling the oil and the blowers for air supply being mounted in sets adjacent to the boilers. In such cases, one pump set can serve several burner units, and common prudence dictates the installation of spare or reserve pump sets. Pre-heaters and other essential auxiliary equipment also should be installed in duplicate.

Boiler Settings

As the volume of space available for combustion is the 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 should 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 lb of oil can properly be burned. This corresponds to a maximum liberation of about 38,000 Btu per cubic foot per hour. There are indications that at times much higher fuel rates may be satisfactory. This in turn suggests that the value of 38,000 Btu per cubic foot per hour might be adjusted according to good engineering judgment. For best results, care should be taken to keep the gas velocity below 40 ft per second. 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 fire-brick 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.

GAS-FIRED APPLIANCES

The increased use of gas for house heating purposes has resulted in the production of such a large number of different types of gas-heating systems and appliances that today there is probably a greater variety of them than there is for any other kind of fuel.

Gas-fired heating systems may be classified as follows:

- I. Gas-Designed Heating Systems.
 - A. Central Heating Plants.
 1. Steam, hot water, and vapor boilers.
 2. Warm air furnaces.
 - B. Unit Heating Systems.
 1. Warm air floor furnaces.
 2. Industrial unit heaters.
 3. Space heaters.
 4. Garage heaters.
- II. Conversion Heating Systems.
 - A. Central Heating Plants.
 1. Steam, hot water and vapor boilers.
 2. Warm air basement furnaces.

The majority of these systems are supplied with either automatic or manual control. Central heating plants, for example, whether gas designed or conversion systems, may be equipped with room temperature control, push button control, or manual control.

Although no exact rules can be prescribed as to the field best covered by each of the foregoing systems, each installation will have problems pointing more or less directly to some particular type of heating equipment.

Gas-Fired Boilers

Information on gas-fired boilers will be found in Chapter 25.

Either snap action or throttling control is available for gas boiler operation. This is especially advantageous in straight steam systems because steam pressures can be maintained at desired points, while at the same time complete cut-off of gas is possible when the thermostat calls for it.

Warm Air Furnaces

There are two general classes of gas-fired warm air furnaces, the gravity furnace which depends upon the natural tendency of heated air to rise, providing the proper circulation of heated air into the room, and the mechanical circulation furnace by which the air to be heated is forced through or drawn through the furnace by means of a fan.

Warm air furnaces are variously constructed of cast iron, sheet metal and combinations of the two materials. If sheet metal is used, it must be of such a character that it will have the maximum resistance to the corrosive effect of the products of combustion. With some varieties of manufactured gases, this effect is quite pronounced. Warm air furnaces are obtainable in sizes from those sufficient to heat the largest residence down to sizes applicable to a single room. The practice of installing a number of separate furnaces to heat individual rooms is peculiar to mild climates, such as that of Southern California. Small furnaces, frequently controlled by electrical valves actuated by push-buttons in the room above, are often installed to heat rooms where heat may be desired for an hour or so each day. These furnaces are used also for heating groups of rooms in larger residences. In a system of this type each furnace should supply a group of rooms in which the heating requirements for each room in the group are similar as far as the period of heating and temperature to be maintained are concerned. Bedrooms, living rooms, and dining rooms often present excellent possibilities for this type of furnace.

The same fundamental principle of design, that is followed in the construction of boilers, that is, breaking the hot gas up into fine streams so that all particles are brought as close as possible to the heating surface, is equally applicable to the design of warm air furnaces. The desirability of using an appliance designed for gas, when gas is to be the fuel, applies even more strongly to furnaces than to boilers.

Codes for proportioning warm air heating plants, such as that formulated by the *National Warm Air Heating and Air Conditioning Association* (see note p. 429), are equally applicable to gas furnaces and coal furnaces. Recirculation should always be practiced with gas-fired warm air furnaces. It not only aids in heating, but is essential to economy. Where fans are used in connection with warm air furnaces for residence heating, it is well to have the control of the fan and of the gas so coordinated that there will be sufficient delay between the turning on of the gas and the starting of the fan to prevent blasts of cold air being blown into the heated rooms. An additional thermostat in the air duct easily may be arranged to accomplish this.

Floor Furnaces

Warm air floor furnaces are well adapted 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 as any number of rooms may be heated without heating the others. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below. Air is taken downward between the two sheets of the double casing and discharged upward over the heating surfaces and into the room. The appliance is controlled from the room to be heated by means of a control lever located near the edge of the register. The handle of the control is removable as a precaution against accidental turning on or off of the gas to the furnace.

Space heaters are generally used for auxiliary heating, but may be, and are in many cases, installed for furnishing heat to entire buildings. Space heaters are quite extensively used for house heating in milder climates such as exist in the South and Southwest. With the exception of wall heaters, they are portable, and can be easily removed and stored during the summer season. Although they should be connected with solid piping it is sometimes desirable to connect them with flexible gas tubing in which case a gas shut-off on the heater is not permitted, and only A.G.A. approved tubing should be used.

Space Heaters

Parlor furnaces or *circulators* are usually constructed to resemble a cabinet radio. They heat the room entirely by convection, *i.e.*, the cold air of the room is drawn in near the base and passes up inside the jacket around a drum or 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 burner or burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass up around baffles within the heating element or drum, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters make admirable auxiliary heating appliances to be used during the occasional cool days at the beginning and end of the heating season when heat is desired in some particular room for an hour or two. The radiant heater gives off a considerable portion of its 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. Some have sheet-iron bodies finished in enamel or brass while others have cast-iron or brass frames with heavy fire clay bodies. An atmospheric burner is supported near the center of the base, usually by set screws at each end. Others have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are supported on legs and are portable; however, there are also types which are encased in a jacket which fits into the wall with a grilled front, similar to the ordinary wall register. Others are encased in frames which fit into fireplaces.

Gas-fired steam and hot water radiators are popular types of room heating appliances. They provide a form of heating apparatus for intermittently heated spaces such as stores, small churches and some types of offices and apartments. They are made in a large variety of shapes and sizes and are

similar in appearance to the ordinary steam or hot water radiator connected to a basement boiler. 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 the steam or hot water radiators. They are usually constructed of pressed steel or sheet metal hollow sections. The hot products of combustion circulate through the sections and are discharged out a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Garage heaters are usually similar in construction to the cabinet circulator space heaters, except that safety screens are provided over all openings into the combustion chamber to prevent any possibility of explosion from gasoline fumes or other gases which might be ignited by an open flame. They are usually provided with automatic room temperature controls and are well suited for heating either residence or commercial garages.

Conversion Burners

Residence heating with gas through the use of conversion burners installed in coal-designed boilers and furnaces represents a common type of gas-fired house heating system, especially in natural gas territories. In many conversion burners radiants or refractories are employed to convert some of the energy in the gas to radiant heat. Others are of the blast type with luminous flames, operating without refractories. In each case an attempt is made to transfer the majority of the heat from the gas to the medium to be heated within the fire pot itself because of the low heat transfer that takes place in the flue passages.

Many conversion units are equipped with sheet metal secondary air ducts which are inserted through the ash-pit door. The duct is equipped with automatic air controls which open when the burners are operating and close when the gas supply is turned off. This prevents a large part of the circulation of cold air through the combustion space of the appliance when not in operation. By means of this duct the air necessary for proper combustion is supplied directly to the burner, thereby making it possible to reduce the amount of excess air passing through the combustion chamber.

Conversion units are made in many sizes both round and rectangular to fit different types and makes of boilers and furnaces. They may be secured with manual, push button, or room temperature control.

Sizing Gas-Fired Heating Plants

While gas-burning equipment can be and usually is so installed as to be completely automatic, maintaining the temperature of rooms at a predetermined and set figure, there are in use installations which are manually controlled. Experience has shown that in order to effectively overcome 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 cast-iron column radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up or starting loads. Consequently, it is possible to use a

much lower selection, or safety, factor. A gas-fired boiler under thermostatic control is so sensitive to variations in room temperatures that in most cases a factor of 25 per cent is sufficient for pick-up load.

The factor to be allowed for loss of heat from piping, however, must vary somewhat, the proportionate amount of piping installed being considerably greater for small installations than for large ones. Consequently, a selection factor for thermostatically controlled boilers must be variable. Table 2 gives liberal selection factors to be added to the installed steam radiation under thermostatic control. They have been established by experience and are recommended by the *American Gas Association*.

The same factors may be used in determining the gas demand for which conversion burners installed in steam or hot water boilers should be set. Multiplying the equivalent direct heating surface (radiation) by 240 and adding the appropriate percentage from Table 2, and then dividing by the heat value of the gas and by the heating efficiency (see discussion of

TABLE 2. SELECTION FACTORS FOR GAS BOILERS

CAST-IRON STEAM RADIATION (EQUIVALENT SQUARE FEET)	SELECTION FACTOR (PER CENT)
500	56.0
800	54.0
1,200	51.0
1,600	48.0
2,000	45.0
3,000	42.5
4,000 and over	40.0

heating efficiencies in Chapter 29), gives the proper hourly rate of gas consumption. However, inadequate boiler heating surface for gas burning, often encountered in coal-designed boilers converted to gas, may necessitate operation at a lesser demand, resulting in much slower pick-up and less margin of safety for piping loss.

Appliances used for heating with gas should bear the approval seal of the *American Gas Association* Testing Laboratory. Installations should be made in accordance with the recommendations shown in the publications of that association.

Ratings for Gas Appliances

Since a gas appliance has a heat-generating capacity that can be predicted accurately to within 1 or 2 per cent, and since this capacity is not affected by such things as condition of fuel bed and soot accumulation, makers of these appliances have an opportunity to rate their product in exact terms. Consequently all makers give their product an hourly Btu output rating. This is the amount of heat that is available at the outlet of a boiler in the form of steam or hot water, or at the bonnet of the furnace in the form of warm air. The output rating is in turn based upon the Btu input rating which has been approved by the *American Gas Association* Testing Laboratory and upon an average efficiency which has been assigned by that association.

In the case of boilers, the rating can be put in terms of square feet of equivalent direct radiation by dividing it by 240 for steam, and 150³ for water. This gives what is called the *American Gas Association* rating, and is the manner in which all appliances approved by the *American Gas Association Laboratory* are rated. To use these ratings it is only necessary to increase the calculated heat loss or the equivalent direct radiation load by an appropriate amount for starting and piping, and to select the boiler or furnace with the proper rating.

The rating given by the *American Gas Association Laboratory* is not only a conservative rating when considered from the standpoint of capacity and efficiency, but is also a safe rating when considered from the standpoint of physical safety to the owner or caretaker. The rating that is placed upon an appliance is limited by the amount of gas that can be burned without the production of harmful amounts of carbon monoxide. This same limitation applies to all classes of gas-consuming heating appliances that are tested and approved by the Laboratory. Gas boilers are available with ratings up to 14,000 sq ft of steam, while furnaces with ratings up to about 500,000 Btu per hour are available. (See Chapter 23.)

Installation Features

One feature of the piping installation that adds to the satisfactory service rendered by gas boilers is provision for adequate and rapid venting of the air from steam heating systems. If air leaks into the steam distribution system during the period that the gas is turned off, and then vents out slowly when the thermostat calls for heat, the result will be a further cooling of the premises between the time that the thermostat calls for heat and the time that steam reaches the radiators. A freely venting steam or vapor system gives maximum economy and minimum temperature variation. When gas boilers are attached to existing heating plants, it is good practice to check the effectiveness of the venting devices and if necessary to replace them with more effective ones that will prevent the return of air into the heating system, and also to check the tightness of the piping.

Frequently when a coal boiler is already installed in a home, it is expedient to leave the coal boiler in place, and to cross-connect the gas boiler with it. Where gas heating is new to the community, it produces a more secure feeling in the customer's mind when putting in gas-fired house-heating equipment, if he knows that he can burn coal at any time he may desire. For steam or vapor installations, it is desirable to have the water line in both boilers at the same level.

REFERENCES

Stoker Information, Bulletin No. 5, Committee of Ten, 307 N. Michigan Avenue, Chicago, Ill.

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).

Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

³A value of 160 for the heat emission of hot water radiators is used by many engineers. The actual heat emission, however, depends on the temperature of the water and of the surrounding air. See Chapters 30 and 33.

Comparison of Oil and Gas Firing in a Heating Boiler, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, October, 1933).

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).

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).

Oil Burning in Residences, by D. W. Nelson (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, July, 1935).

PROBLEMS IN PRACTICE

1 ● What features of furnace design are essential for the proper burning of the volatile coal gases above the fuel bed?

Adequate provisions should be made so that the furnace volume is sufficiently liberal and that the grates are a sufficient distance from the heating surfaces to permit the proper combustion of gasses.

2 ● Classify stokers as to construction and operation.

- a. Overfeed flat grate.
- b. Overfeed inclined grate.
- c. Underfeed side cleaning type.
- d. Underfeed rear cleaning type.

3 ● What classification may be made of stokers as to their use?

Class 1. For residences (Capacity less than 60 lb of coal per hour).

Class 2. For apartment houses and small commercial heating jobs (Capacity 60 to 500 lb of coal per hour).

Class 3. For general commercial heating and small high pressure steam plants (Capacity 500 to 1200 lb of coal per hour).

Class 4. For large commercial and high pressure steam plants (Capacity over 1200 lb of coal per hour and over 36 sq ft grate area).

4 ● What main parts are found in an underfeed residential stoker?

A *hopper* is supplied to hold coal which is fed by a *screw* or *plunger* into a *retort* provided with air openings called *tuyeres*. A *blower* supplies air under pressure for combustion, and a *gear case* provides for changes in coal feeding rates.

5 ● What is a dead-plate?

A dead-plate is a flat surface without air supply openings upon which the fuel rests while combustion of the fixed carbon is completed. Generally the ash is removed from the dead-plate.

6 ● What rate of coal burning is usually recommended for small underfeed stokers?

For continuous operation, 25 lb per square foot of grate surface is recommended; for short duration peaks, 30 lb.

7 ● What methods of oil atomization are used?

1. Throwing the oil from a rotating cup or disc.
2. Forcing the oil under high pressure through a nozzle.
3. Propelling the oil with a high velocity jet of air or steam.
4. Forcing an oil and air mixture through a nozzle.

8 • What is the purpose of atomization?

Atomization is used to increase the surface area of the oil in order to facilitate putting it into a vaporous state so it may burn.

9 • Is the furnace of much importance in oil burning?

In most cases it is very important. It is the function of the oil burner to supply the air and fuel in correct proportions; the furnace must provide heated space for proper mixing and combustion.

10 • Which flame is considered better, the luminous or the non-luminous?

Laboratory tests show that they are equally efficient in the usual installation.

11 • What main precaution is necessary in choosing a boiler for an oil burner?

Since the burner output is usually varied through a wide range under control of the thermostat, a boiler should be provided with enough indirect heating surface to absorb the heat as it is released. The combustion space must be large enough, and have correct proportions for mixing fuel and air at high temperatures. If oil is used inefficiently high heating costs will result.

12 • How should oil burner adjustments be made?

Adjustments should be made by an experienced man who uses a gas analysis apparatus to determine the CO_2 content.

13 • What CO_2 content should be attained in oil burning?

Ten per cent CO_2 is considered good practice, for it indicates the supplying of 50 per cent excess air.

14 • What maximum heat release is considered good practice in oil burning?

A heat release of 38,000 Btu per cubic foot per hour is considered to be the maximum for average large installations. This figure has been greatly exceeded in some cases. The design of the combustion chamber, as to impingement of flame and as to proper mixing at high temperatures, has much to do with the attainable heat release.

15 • Name five types of gas-fired space heaters.

- a. Parlor furnaces or circulators.
- b. Radiant heaters.
- c. Gas-fired steam or hot water radiators.
- d. Warm air radiators.
- e. Garage heaters.

16 • How are gas heating units rated?

Gas-fired units are rated on the basis of output in Btu per hour.

17 • What safety consideration is noted in establishing the ratings of gas-fired units?

The rating is limited by the amount of gas that can be burned without the liberation of harmful amounts of carbon monoxide.

18 • List some factors which might account for possible economies of stoker firing over hand firing.

- a. The regular feed of coal instead of the intermittent feed.
- b. The use of cheaper sizes and grades of fuel.
- c. The absence of door openings for firing purposes, which avoids the admission of cold excess air.
- d. The avoidance of overheating because the stoker responds quickly to automatic equipment controlled by the heat demand.

Chapter 29

HEAT AND FUEL UTILIZATION

Total Heat Loss Requirements, Utilization Factors, Degree-Day Methods, Base Temperature Determinations, Steam Consumption of Buildings, Fuel Consumption, Maximum Demands, Load Factors

THE hourly heat loss (H) is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i) of the rooms or spaces to be heated. The total equivalent heating surface required is equal to $\frac{H}{240}$ sq ft.

In estimating the fuel consumption of a building of more than one room divided by walls or partitions, it is not correct to use the calculated heat loss of the building without making the proper allowances for the fact that the heating load at any time does not involve the sum of the infiltration losses of all the heated spaces of the building but only part of the infiltration losses. This is explained in Chapter 6.

It is sufficiently accurate in most cases to consider only half of the total infiltration losses of a building having interior walls and partitions. The value of H in Equation 1 would, under these conditions, be equal to $H_t + \frac{H_i}{2}$.

In some cases, where the building has no interior walls or partitions, the infiltration losses are calculated by using only half of the total crack. In this case the entire infiltration loss should be considered.

The heat required to warm the cold building and contents is a factor to be considered. Under certain conditions the cooling of the structure and contents will, to some extent, compensate for the heat required to rewarm the building. For example, if the building is under thermostatic control and the day and night temperatures are, 70 F and 50 F respectively, there will be a period during which no heat will be added while the building is cooling to 50 F, and the saving resulting therefrom will correspond to the additional heat required to bring the building and contents back to the daytime temperature.

HEAT AND FUEL CONSUMPTION

There are two methods in use for estimating heat or fuel consumption. One method is theoretical, based on a calculated heat loss and assuming absolute 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, sun effect, poor heating systems, etc.

The second method is based on steam consumption data which have been taken from a group of buildings in operation, and the results computed on a degree-day basis. While this method may not be as theoretically correct as the first mentioned method, it is of more value for practical use. Calculations of heat consumption made by the second method will invariably be higher than calculations made by the first method.

Method 1

To predict the amount of fuel likely to be consumed in heating a building during a normal heating season, it is necessary to know the total heat requirements of the building and the utilization factor of the fuel. The accuracy of the estimate will depend on the ability to select these values and on the care taken in making allowances for other variable factors.

Heat requirements are given by the following general formula:

$$M = \frac{H(t - t_a)N}{t_d - t_o} \quad (1)$$

Steam requirements are determined by dividing the above by 1000, thus:

$$S = \frac{H(t - t_a)N}{(t_d - t_o)1000} \quad (2)$$

Fuel requirements may be determined by the following formula:

$$F = \frac{M}{C \times E} \quad (3)$$

where

- t = inside temperature, degrees Fahrenheit.
- t_d = inside design temperature, degrees Fahrenheit.
- t_a = average outside temperature, degrees Fahrenheit (Table 2, Chapter 7).
- t_o = outside design temperature, degrees Fahrenheit.
- H = calculated heat loss of building based on outside temperature (t_o), Btu per hour.
- N = number of heating hours per season; 5088 from October 1 to May 1¹.
- M = heat loss, Btu per season.
- S = steam required to supply M Btu of heat loss.
- F = quality of fuel required per heating season.
- C = calorific value of one unit of fuel, the unit being the same as that on which F is based.
- E = efficiency of utilization of the fuel, percent.

Example 1. A small factory building located in Philadelphia is to be heated to 60 F between the hours of 7 A.M. and 7 P.M., and to 50 F during the remaining hours. The calculated hourly heat loss based on a design temperature of -6 F is 500,000 Btu. If coal having a calorific value of 12,500 Btu is fired and the overall heating efficiency is assumed to be 60 per cent, how many pounds of steam would be required for a normal heating season?

¹This is the period for which t_a (Table 2, Chapter 7) is calculated. If the heating season is different than this period, the corrected values may be substituted for N and t_a .

Solution. Since there are no partitions in the building, the entire heat loss is considered. From Table 2, Chapter 7, the average outside temperature (t_a) during the heating season is 41.9 F; N for the period for which t_a is taken (October 1 to May 1) is 5088; H = 500,000; t_o = -6 F; t = 50 F and 60 F; t_d = 60 F; E = 60 per cent average for heating season; C = 12,500.

The average daily temperature for the 24 hours is:

$$\frac{50 \times 12 + 60 \times 12}{24} = 55 \text{ F}$$

Substituting in Equation 1:

$$M = \frac{500,000 \times (55 - 41.9) \times 5088}{60 - (-6)} = 504,942,000$$

S = 504,942 lb of steam.

$$F = \frac{504,942,000}{0.60 \times 12,500} = 67,325 \text{ lb of coal} = 33.7 \text{ tons.}$$

Method 2—Degree-Day Method

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. Some years ago the *American Gas Association*² determined from experiment in the heating of residences that the gas consumption varied directly as the difference between 65 F and the 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. The degree day is defined in Chapter 44. Degree-days for various cities in the United States and Canada are given in Table 1.

Establishing the Base Inside Temperature. Recently the *National District Heating Association* has studied the metered steam consumption of 163 buildings³ in 22 different cities and has published data substantiating the fact that the 65 F base originally chosen by the gas industry is approximately correct.

The steam consumption of each building by months was divided by the number of days in each month, thus giving the average daily steam consumption by months. The average steam consumption was then plotted against the average monthly temperature, as shown in Fig. 1, and the temperature at which a line drawn through the points crossed the base line indicated the temperature corresponding to zero steam consumption, or the base temperature. The composite results from 163 buildings calculated in this manner are shown in Table 2.

The resultant average of 66.0 F is close to the A.G.A. figure of 65 F. It will be noted that the base temperature calculated for hotels, apartments and residences is consistently higher than those for such buildings as garages, auto sales buildings, and manufacturing buildings. This, of course, would be expected in view of the higher inside temperatures carried in the former group; in fact, an even greater difference would be

²See *Industrial Gas Series, House Heating* (third edition) published by the American Gas Association.

³These buildings are all served with steam from a district heating company.

TABLE 1. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA^a

STATE	CITY	JAN.	FEB.	MAR.	APR.	MAY	SEPT.	OCT.	NOV.	DEC.	TOTAL
Ala.	Birmingham	589	577	260	69				318	595	2408
	Mobile	428	311	152					186	394	1471
Ariz.	Flagstaff	1153	969	896	654	636 ^b	292 ^d	577	840	1128	7145
	Tucson	459	325	257	87				252	465	1845
Ark.	Hot Springs										2665
	Little Rock	719	582	353	78			47	381	651	2811
Calif.	Los Angeles	326	266	239	159	90			123	301	1504
	San Francisco	465	356	354	294	458 ^b	146	261	428	3264	
Col.	Colorado Springs	1085	993	884	612	459 ^b	162	502	789	1067	6553
	Denver	1079	918	799	534	267	72	428	759	1017	5873
Conn.	New Haven	1110	1011	899	543	223	39	360	693	1017	5895
D. C.	Washington										4626
Fla.	Jacksonville	285	207	56					75	267	890
Ga.	Atlanta	682	558	388	132			96	396	639	2891
	Savannah	409	316	167					201	397	1490
Idaho.	Boise	1098	848	651	435	235	108	434	738	1011	4558
	Lewiston										4924
Ill.	Chicago	1262	1095	909	549	248	30	353	756	1113	6315
	Springfield	1180	1008	760	365	56		282	681	1038	5370
Ind.	Evansville	949	854	640	276			155	528	862	4164
	Indianapolis	1128	969	756	384	58		298	687	1017	5297
Iowa	Des Moines	1392	1173	890	429	118		357	798	1216	6373
	Sioux City	1434	1386	967	489	164	33	415	870	1265	7023
Kans.	Dodge City	1116	890	688	342	46		276	672	1004	5034
	Topeka	1221	980	741	339			270	699	1051	5301
Ky.	Lexington	974	867	648	342	25		245	612	903	4616
	Louisville	939	801	589	264			186	552	849	4180
La.	New Orleans	332	230	58					102	301	1023
Me.	Eastport	1380	1232	1110	786	843 ^b	566 ^c	543	843	1228	8531
	Portland	1321	1168	1017	642	368	120	443	780	1153	7012
Md.	Baltimore	955	843	700	348	22		223	567	875	4533
Mass.	Boston	1150	1042	908	570	245	48	363	693	1026	6045
	Springfield										6464
Mich.	Detroit	1253	1134	976	573	226	42	400	777	1113	6494
	Marquette	1501	1360	1249	804	682 ^b	268 ^d	567	960	1301	8692
Minn.	Duluth	1727	1473	1277	810	722 ^b	298 ^d	620	1062	1491	9480
	Minneapolis	1609	1400	1095	570	235	93	481	963	1405	7851
Miss.	Vicksburg	520	384	195					252	471	1822
Mo.	Kansas City	1201	987	750	321	15		285	605	1038	5202
	St. Louis	1060	654	657	276			205	597	936	4585
Mont.	Billings	1316	1120	955	534	376 ^b	189	524	909	1192	7115
	Havre	1624	1450	1168	630	513	270	620	1041	1383	8699
Nebr.	Lincoln										6231
	Omaha	1355	1125	868	414	84		328	780	1174	6128
Nev.	Reno	1041	823	753	534	456 ^b	144	452	714	974	5891
N. H.	Concord	1349	1240	1011	669	351 ^b	168	484	846	1234	6852
N. J.	Atlantic City	992	903	806	519	220		254	588	893	5175
	Trenton	1014	942	735	402	81		242	588	930	4934
N. M.	Santa Fe	1110	902	775	543	301 ^b	120	459	780	1073	6063
N. Y.	Albany	1286	1142	980	549	493	72	446	774	1147	6889
	Buffalo	1240	1156	1032	675	347	75	418	774	1104	6821
	New York	1061	960	837	486	155		276	618	955	5348
	Utica	1242	1181	991	587	244	187	426	787	1140	6785
N. C.	Raleigh	722	630	446	183			130	429	694	3234
	Wilmington	555	468	322	108			19	303	527	2302
N. Dak.	Bismarck										8498

^aHeating and Ventilating Degree-Day Handbook.^bIncluding June.^cIncluding July and August.^dIncluding August.TABLE 1. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA^a (Continued)

STATE	CITY	JAN.	FEB.	MAR.	APR.	MAY	SEPT.	OCT.	NOV.	DEC.	TOTAL
Ohio.	Cincinnati	1011	871	678	339	19		248	615	921	4702
	Cleveland	1180	1075	950	564	220	27	366	732	1060	6154
	Columbus	1113	980	703	420	87		313	690	1017	5323
Okla.	Oklahoma City	865	742	465	162			105	459	815	3613
Oreg.	Portland	806	644	558	402	335 ^b	105	332	558	728	4468
	Salem										4629
Pa.	Philadelphia	1001	895	756	402	68		242	588	903	4855
	Pittsburgh	1054	944	787	423	78		313	669	967	5235
R. I.	Providence	1116	1069	890	558	251	63	348	693	1026	6014
S. C.	Charleston	487	372	242	36			207	425	1769	
	Spartanburg	725	688	431	147			121	429	716	3257
S. Dak.	Sioux Falls										7683
Tenn.	Memphis	744	599	384	96			62	402	663	2950
	Nashville	812	747	476	180			136	483	744	3578
Texas.	Austin										1578
	Dallas										2455
	Houston	366	277	65					114	335	1157
	San Antonio	381	274	74					126	347	1202
Utah.	Logan	1260	1072	893	525	376	114	468	819	1218	6735
	Salt Lake City	1110	885	722	453	234	18	388	723	1020	5553
Vt.	Burlington	1535	1294	1089	654	276 ^b	144	481	861	1286	7620
Va.	Fredericksburg	887	820	583	303			223	549	878	4243
	Norfolk	738	650	520	246			99	411	685	3349
	Richmond	825	702	552	240			158	483	765	3725
Wash.	Seattle	775	653	623	465	487 ^b	276 ^c	403	570	716	4968
	Spokane	1171	952	778	504	366	192	514	819	1057	6353
W. Va.	Morgantown	1026	944	713	414			294	648	977	5016
	Parkersburg										4884
Wis.	Fond du Lac	1507	1321	1046	603	276	117	493	921	1328	7612
	Green Bay	1538	1358	1125	600	322	132	505	921	1322	7823
	LaCrosse	1535	1265	1032	528	183	96	462	909	1280	7290
	Milwaukee	1383	1328	1023	648	389 ^b	84	449	846	1222	7372
Wyo.	Cheyenne	1215	1075	995	720	569	240	605	900	1143	7462

PROVINCE	CITY	JAN.	FEB.	MAR.	APR.	MAY	SEPT.	OCT.	NOV.	DEC.	TOTAL
B. C.	Victoria										5777
	Vancouver										5976
	Kamloops										6724
Alb.	Medicine Hat										8152
Sask.	Qu'Appelle										11,261
Man.	Winnipeg										11,166
Ont.	Port Arthur										10,803
	Toronto										7732
Que.	Montreal	1615	1409	1219	720	309	190	372	961	1422	8417
	Quebec										8628
N. B.	Fredericton										9099
N. S.	Yarmouth										7694
P. E. I.	Charlottetown										8485

^aHeating and Ventilating Degree-Day Handbook.^bIncluding June.^cIncluding July and August.

TABLE 2. BASE TEMPERATURE FOR THE DEGREE-DAY^a

TYPE OF BUILDING	NO. OF BUILDINGS ANALYZED	TEMPERATURE F CORRESPONDS TO ZERO STEAM CONSUMPTION
Office.....	60	66.2
Office and Bank.....	4	65.8
Bank.....	3	66.2
Office and Telephone Exchange.....	2	65.5
Office and Stores.....	6	67.4
Stores.....	11	64.0
Department Stores.....	12	64.3
Hotels.....	7	66.5
Apartments.....	14	68.8
Residences.....	8	66.9
Clubs.....	4	65.5
Lodges.....	5	64.9
Theatres.....	3	67.6
Churches.....	2	65.8
Garage.....	2	64.8
Auto Sales and Service.....	4	61.2
Newspaper and Printing.....	3	67.7
Warehouse and Loft.....	3	67.7
Office and Loft.....	2	65.2
Manufacturing.....	8	65.4
Average for 163 Buildings.....		66.0 F

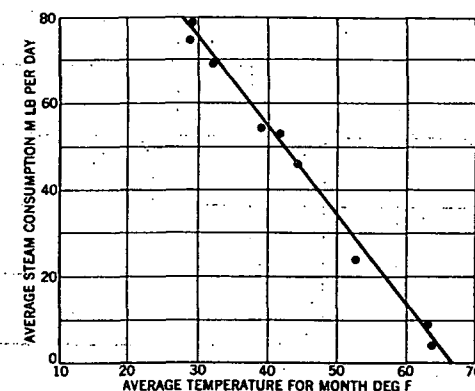
^aReport of Commercial Relations Committee, 1938 Proceedings, National District Heating Association.TABLE 3. STEAM CONSUMPTION FOR VARIOUS CLASSES OF BUILDINGS^a
(Heating Season Only)

BUILDING CLASSIFICATION	No. OF BUILDINGS LISTED	STEAM CONSUMPTION POUNDS PER DEGREE-DAY—65 F BASED		
		Per M Cu Ft of Heated Space	Per M Sq Ft of Radiator Surface	Per M Btu per Hr of Heat Loss ^b
Apartments.....	16	1.78	97.5	0.359
Hotels.....	10	1.46	80.6	0.371
Residences.....	12	1.32	64.2	-----
Printing.....	7	1.25	105.5	-----
Clubs and Lodges.....	10	0.96	77.0	-----
Retail Stores.....	18	0.90	80.6	0.268
Theatres.....	6	0.90	75.0	0.498
Loft and Mfg.....	16	0.89	72.3	0.283
Banks.....	7	0.88	45.2	-----
Auto Sales and Service.....	8	0.83	62.2	-----
Churches.....	6	0.58	49.4	-----
Department Stores.....	14	0.57	60.7	0.238
Garages (Storage) ^c	6	0.42	72.3	-----
Offices (Total).....	35	1.09	70.0	0.283
Offices (Heating only).....	35	0.975	65.4	0.256

^aIncludes steam for heating domestic water for heating season only.^bHeat loss calculated for maximum design condition (in most cases 70 F inside, zero outside).^cEquivalent steam radiator surface.^dThe figures are a numerical—not a weighted—average for the several buildings in each class.^eBased on zero consumption at 55 F.

expected. For an average figure, the A.G.A. base of 65 F may therefore be safely used, and if greater refinement is desired, the figure for the type of building under consideration can be taken from Table 1.

Table 3⁵ gives the steam consumption per degree-day, expressed in three different ways, for 196 buildings in 14 different classifications. These buildings are divided among 21 different cities in the United States. The steam used for heating the domestic water is included in these figures, but in the case of office buildings, the steam for heating only is also shown. The data are placed on a comparable basis by expressing the steam consumption in terms of pounds per degree-day per thousand square feet of equivalent installed radiator surface, per thousand cubic feet of heated space, and per thousand Btu of calculated heat loss.

FIG. 1. METHOD OF DETERMINING BASE TEMPERATURE FOR DEGREE-DAY CALCULATIONS⁴

The choice of these units of comparison require some explanation. The use of *heated* space in preference to the gross cubage used by architects is obviously more accurate for this purpose. The architect's cubage includes the outer walls and certain percentages of attic and basement space which are usually unheated. The net heated space is usually about 80 per cent of the gross cubage and can be calculated from the latter if it cannot be measured. The cubical content is somewhat inaccurate as a basis of comparison due to differences in types of construction, exposure, and ratio of exposed area to cubical contents.

The use of radiator surface as the basis of comparison has two objections. One is that the amount of radiator surface in a building is often either excessive or deficient, and figures for steam consumption based on it are therefore likely to be in error. Another reason is that it is difficult to convert fan coil surface into equivalent direct radiator surface with accuracy. On the whole, the use of radiator surface as the basis of comparison is the least satisfactory of the three methods.

⁴Report of Commercial Relations Committee, 1938 Proceedings, National District Heating Association.⁵The Heat Requirements of Buildings, by J. H. Walker and G. H. Tuttle (A.S.H.V.E. JOURNAL SECTION, Heating, Piping and Air Conditioning, December, 1934).

It should be noted that the figures in Table 2 are for the heating season only and include steam for heating domestic water.

Example 1 solved by the degree-day method and using values taken from Table 3 would show a higher steam consumption.

Example 2. Factor for steam consumption for a manufacturing building per M Btu per hour heat loss per degree-day = 0.283; total number of degree-days per year (Table 1) = 4855; heat loss = 500,000 Btu per hour.

Solution. $0.283 \times 4855 \times 500 = 686,982$ lb of steam per year. This calculation results in an estimate 36 per cent higher than the previous calculation and one which would be more nearly correct for actual practice.

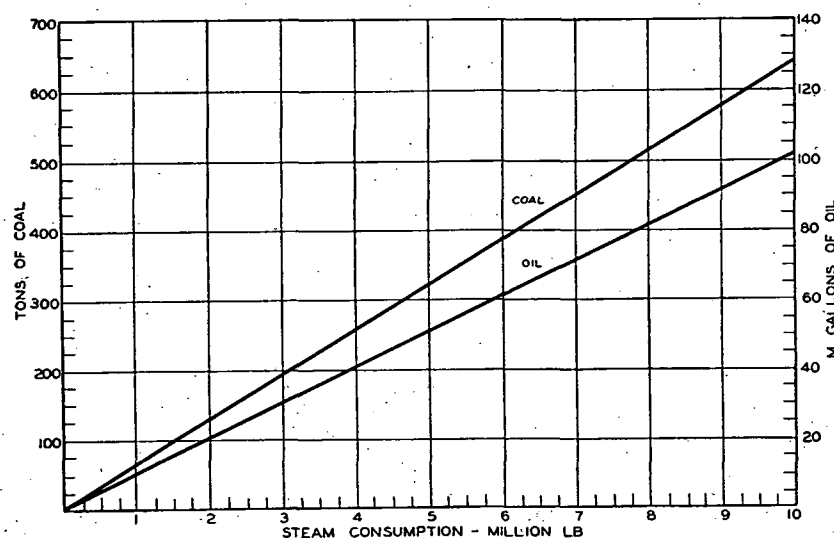


FIG. 2. CURVE FOR ESTIMATING FUEL CONSUMPTION FOR VARIOUS KNOWN STEAM CONSUMPTION^a

^aThis curve is based on heating efficiencies of 60 to 70 per cent for coal and oil, respectively, a calorific value of coal of 13,000 Btu per pound, a calorific value of oil of 140,000 Btu per gallon.

In case the heat loss figure is not known this method of estimating heat or steam consumption can also be applied if the net heated space figure is available.

CALCULATION OF FUEL CONSUMPTION

After the heat and steam consumption of the building have been calculated, the corresponding fuel requirements may also be estimated by assuming the correct boiler and furnace efficiencies. If the building is to be supplied with steam from a district heating company, the steam consumptions as calculated by Methods 1 and 2 are generally assumed to be correct. However, if the steam is to be supplied from an individual boiler

unit, the consumption should be assumed as from 10 to 20 per cent greater. One reason for this difference in steam consumption is that district steam is a metered service, and building managers are therefore more conscious of their heating costs, which generally results in better maintained heating systems. Also, the district steam service is usually installed with thermostatic control which reduces overheating to a minimum.

Fig. 2 shows the amount of coal or oil that may be estimated when the steam consumption is known. Assuming the steam consumption that

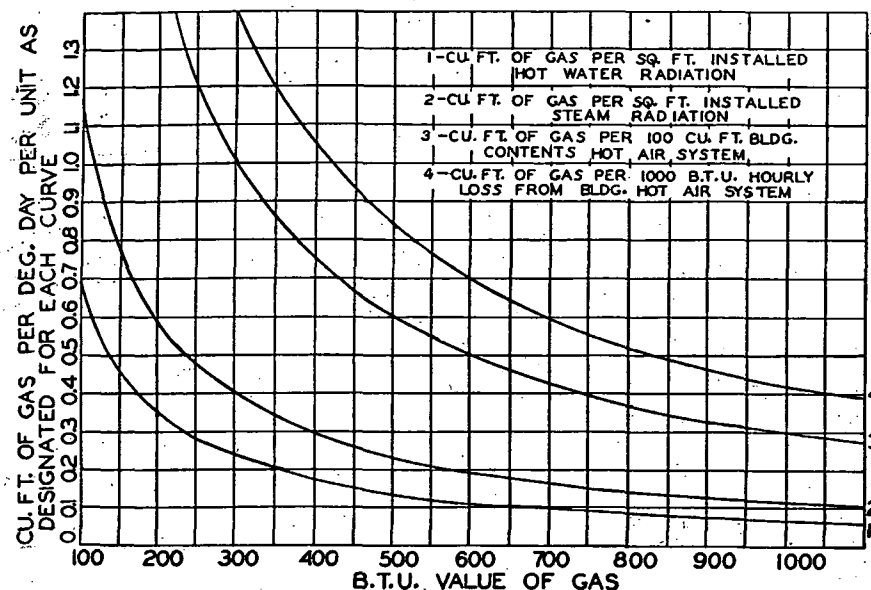


FIG. 3. CHART GIVING GAS REQUIREMENTS PER DEGREE-DAY FOR VARIOUS CALORIFIC VALUES OF GAS AND FOR DIFFERENT HEATING SYSTEMS^a

^aThis chart is based on an inside temperature of 70 F and an outside temperature of zero. If the radiation is installed on the basis of any other temperature difference, multiply the result obtained from this chart by 70, and divide by the actual temperature difference. From *Industrial Gas Series House Heating* (third edition) published by the American Gas Association.

was calculated in Example 2, 686,982 lb, the corresponding coal consumption, from the curve, is 44 tons and the oil consumption 6000 gal.

Fig. 3 indicates the average gas consumption per degree-day for various heat contents. While the fuel consumption in individual cases may vary somewhat from the curve values, these average values are sufficiently accurate for estimating purposes and give satisfactory results.

The value generally used in the manufactured gas industry for residences is 0.21 cu ft per degree-day per square foot of equivalent steam radiation (240 Btu) based on the theoretical requirements. A correction for warmer climates is necessary and it is customary to gradually increase the relative fuel consumption below 3000 degree-days to about 20 per cent more at 1000 degree-days.

For hot water or warm air heat the fuel consumption is about 0.19 cu ft per degree-day per square foot of equivalent steam *radiation*, that is, per 240 Btu per hour. The actual requirements likewise relatively increase with hot water or warm air systems as the number of degree-days decreases below 3000. For larger installations, that is 1000 sq ft of theoretical *radiation* and above, there is an increase in efficiency, and a consequent decrease in the fuel consumption per degree-day per square foot of heating surface.

The approximate quantities of steam required in New York City per square foot of heating surface for various classes of buildings are given in Chapter 37.

The preceding discussion on fuel consumption has dealt with the heating requirements of the building irrespective of any air that may be introduced for ventilation purposes other than the normal infiltration of outside air. The heat required for warming air brought into the building for ventilation may be estimated from data given in Chapters 3 and 22.

MAXIMUM DEMANDS AND LOAD FACTORS

In one form of district heating rates, a portion of the charge is based upon the maximum demand of the building. The maximum demand may be measured in several different ways. It may be taken as the instantaneous peak or as the rate of use during any specified interval. One method is to take the average of the three highest hours during the winter. These figures are available for a number of buildings in Detroit, as shown in Table 4.

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.

TABLE 4. BUILDING LOAD FACTORS AND DEMANDS OF SOME DETROIT BUILDINGS*

BUILDING CLASSIFICATION	LOAD FACTOR	LB OF DEMAND PER HR 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
Theatres.....	0.126	0.151

*See Footnote 5 p. 515.

The load factor of a building is the ratio of the average load to the maximum load and is an index of the utilization habits. Thus, in Table 4,

the theatres, operating for short hours, have a load factor of 0.126 as compared with the figure of 0.318 for clubs and lodges.

PROBLEMS IN PRACTICE

1 • Is it correct to use the total calculated heat loss of a building for estimating fuel consumption?

No. The heating load does not generally involve the sum of the separate infiltration losses of the several heated spaces. When a building has interior walls and partitions it is sufficiently accurate to consider only half of the total calculated infiltration losses.

2 • What will be the 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 heating 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.

The average hourly temperature is

$$\frac{(72 \times 16) + (55 \times 8)}{24} = 66.3 \text{ F.}$$

The maximum hourly heat loss will be

$$92,000 - \frac{26,000}{2} = 79,000 \text{ Btu} = H.$$

$$M = \frac{79,000 (66.3 - 36.4) \times 24 \times 210}{100,000 \times 0.75 \times (72 - 0)} = 2204.6 \text{ hundred thousand Btu.}$$

$$2204.6 \times 0.07 = \$154.34 = \text{cost per year of heating the building.}$$

3 • What factors should be taken into consideration when determining the efficiency at which a fuel will be burned?

Manufacturers' catalogs usually give equipment efficiencies obtained under test conditions. These values do not allow for poor attendance, defects in installation, or poor draft. Such efficiencies do not consider heat radiated from the outside of the equipment, but in many cases this heat is utilized.

4 • If 20 tons of coal having a calorific value of 13,000 Btu per pound are burned in a warm air furnace and produce 286,000,000 Btu at the bonnet, what is the efficiency of the furnace?

$$\frac{\text{Number of Btu at bonnet}}{\text{Number of tons} \times \text{calorific value} \times \text{number of lb in one ton}} = \text{efficiency.}$$

$$\frac{286,000,000 \times 100}{20 \times 13,000 \times 2000} = 55 \text{ per cent.}$$

5 • In making degree-day calculations, why is the base of 65 F used for an inside temperature of 70 F?

This base was chosen because data collected from numerous installations show that heat is seldom supplied to a residence when the outdoor temperature is greater than 65 F. It was also found that the amount of fuel consumed varied in almost direct proportion with the difference between 65 F and the outside temperature.

6 ● Use the degree-day method of computing the amount of coal required to heat an office building located in Cleveland, Ohio, assuming that the net heated space is 30,000 cu ft.

The steam consumption factor for office buildings is 0.975 pounds per M cu ft per degree-day (Table 3). Cleveland has 6154 degree-days per year (Table 1).

$$0.975 \times 6154 \times 30 = 180,000 \text{ lb of steam.}$$

From Fig. 2, this is equivalent to 13 tons of coal.

7 ● Make a rough approximation of the gas required to heat a building located in Chicago, Ill., assuming that the calculated heating surface requirements are 1000 sq ft of hot water radiation based on design temperatures of 0 F and 70 F. Chicago has 800-Btu mixed gas, and 6315 degree-days.

Using Fig. 2, the fuel consumption for a design temperature of 0 F with 800-Btu gas is found to be 0.08 cu ft of gas per degree-day per square foot of hot water radiation.

$$0.08 \times 6315 \times 1000 = 505,200 \text{ cu ft.}$$

8 ● A certain building has a maximum heat loss of 250,000 Btu per hour in -15 F weather. How many tons of fuel will be required to maintain a temperature of 70 F during a 260-day heating season in which the average temperature is 39 F? The heating value of the fuel is 13,200 Btu per pound and the efficiency of combustion is 60 per cent.

$$\frac{250,000 (70 - 39) 260 \times 24}{(70 + 15) 13,200 \times 0.60 \times 2000} = 35.9 \text{ tons.}$$

9 ● Which item may be determined more closely, the heating value of a fuel or the efficiency of its combustion?

The heating values of oil, gas, and solid fuels are closely determinable, whereas the efficiency of burning depends on the particular equipment chosen and the skill used in handling it.

10 ● In an office building, the thermostats are set to maintain 70 F from 7 a.m. to 5 p.m. and 50 F during the rest of the time. When the outside temperature is 30 F, how much saving might be expected because the temperatures are lowered? Under the above conditions the building becomes 50 F by 11 p.m. and warms up to 70 F by 8 a.m.

A temperature of 70 F is maintained during 9 hours, and one of 50 F during 8 hours; the temperature would average about 60 F during the 7 hours required for cooling down and warming up. The average is 60.4 for the 24 hours. (The average temperature calculated would have been 58.3 F, had the warming and cooling periods been neglected.)

$$\text{The saving is } \left(\frac{70 - 60.4}{70 - 30} \right) \times 100 = \frac{9.6}{4} \times 100 = 24 \text{ per cent.}$$

11 ● How does the heat capacity of a structure influence the saving made by carrying lower temperatures during the night?

The heat storage capacity of the walls prevents rapid dropping of temperatures at night-time and delays the warming up process in the morning. In an extreme case, the building would not reach the lowered temperature by the time the higher temperature is called for in the morning. But under any conditions, the saving made by lowering the temperature can be correctly estimated by using the average temperature observed over the 24-hour period as a factor, as in Question 10.

12 ● What are some of the miscellaneous factors that may cause actual fuel consumption to vary from the theoretical fuel requirements as calculated by the use of heat losses, temperature difference, and fuel burning efficiency?

The opening of windows; abnormally high or low inside temperatures; other sources of heat, such as machinery or lights; sun effect; and unusual winds.

Chapter 30

RADIATORS AND GRAVITY CONVECTORS

Heat Emission of Radiators and Convectors, Types of Radiators, Output of Radiators, Heating Effect, Heating Up the Radiator, Enclosed Radiators, Convectors, Selection, Code Tests, Gravity-Indirect Heating Systems

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

All 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 less than half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, 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.

TYPES OF RADIATORS

Present day radiators may be classified as tubular, wall, or window types, and are generally made of cast iron. Catalogs showing the many designs and patterns available now include a junior size which is more compact than the standard unit.

Pipe Coil Radiators

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.

OUTPUT OF RADIATORS

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 now the *Mb* or 1000 Btu.) 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 per square foot.

TABLE 1. VARIATION IN DIMENSIONS AND CATALOG RATINGS OF 10-SECTION TUBULAR RADIATORS

No. of Tubes	3	4	5	6	7
Width of Radiator—Inches	4.6-5.1	6.0-7.0	8.0-8.9	9.1-10.4	11.4-12.8
Length per Section—Inches	2.5	2.5	2.5	2.5	2.5-3.0
HEIGHT WITH LEGS—INCHES	HEAT EMISSION—EQUIVALENT SQUARE FEET				
13-14				20	25.0-32.5
16-18			28.5		30.0-38.3
20-21	15.0-17.5	20.0-22.5	25.0-31.2	30	36.7-45.0
22-23	20.0-21.3	25	30.0-33.9	35	40.0-45.2
25-26	20.0-26.7	25.0-27.5	32.5-39.8	37.5-40.0	50.0-53.5
30-32	25.0-30.9	33.3-35.0	40.0-48.6	50	63.3-62.5
36-38	30.0-36.7	40.0-42.5	50.0-56.5	60	70.0-75.4

Output of Tubular Radiators

Table 1 illustrates the difficulty in tabulating tubular radiator outputs since there is so much variation in design between the products of the different manufacturers. Only on the four-tube and six-tube sizes is there any practical agreement in output value. The heat emission values appear as square feet but are entirely empirical, being based on the heat emission of the radiator and not on the measured surface.

Output of Wall Radiators

An average value of 300 Btu per actual square foot of surface area per hour has been found for wall radiators one section high placed with their bars vertical. Several recent tests show that this value will 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 so noticeable a difference in temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of a room satisfactorily.

*University of Illinois, Engineering Experiment Station Bulletin No. 223, p. 30.

Output of Pipe Coils

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 available 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.

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

Effect of Paint

The prime coat of paint on a radiator has little effect on the heat output, but the finishing coat of paint does influence the radiation emission. Since this is a surface effect, there is no noticeable change in the convection loss. Thus, the larger the proportion of direct radiating surface, the greater will be the effect of painting on the radiation. Available tests are on old-style column type radiators which gave results shown in Table 3.

TABLE 3. EFFECT OF PAINTING 32-IN. THREE COLUMN, SIX-SECTION CAST-IRON RADIATOR^a

RADIATOR No.	FINISH	AREA Sq Ft	COEFFICIENT OF HEAT TRANS. BTU	RELATIVE HEATING VALUE PER CENT
1	Bare iron, foundry finish.....	27	1.77	100.5
2	One coat of aluminum bronze.....	27	1.60	90.8
3	Gray paint dipped.....	27	1.78	101.1
4	One coat dull black Pecora paint....	27	1.76	100.0

^aComparative Tests of Radiator Finishes, by W. H. Severns (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927).

Effect of Superheated Steam

Available research data indicates that there is probably a decrease in heat transfer rate for a radiator or gravity convector with superheated steam in comparison with saturated steam at the same temperature. The decrease is probably small for low temperatures of superheats and additional tests are necessary with varying degrees of superheat to establish accurate comparisons for all types of radiators and convectors².

²Tests of Radiators with Superheated Steam, by R. C. Carpenter (A.S.H.V.E. TRANSACTIONS, Vol. 7, 1901, p. 206).

HEATING EFFECT

For several years the *heating effect* of radiators has been considered by engineers in order to use it for the rating of radiators and in the design of heating systems. Heating effect is the *useful output* of a radiator, in the comfort zone of a room, as related to the total input of the radiator³.

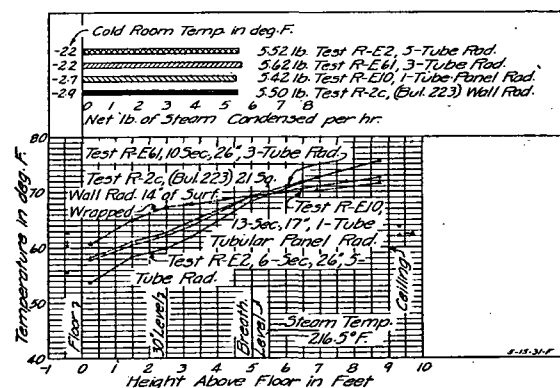


FIG. 1. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 60-IN. LEVEL

Note that the steam condensations are practically the same for all four radiators when the same air temperature of 69 F is maintained at the 60-in. level.

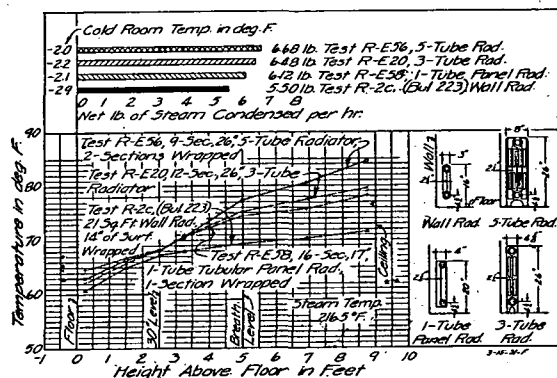


FIG. 2. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR FOUR TYPES OF CAST-IRON RADIATORS WITH A COMMON TEMPERATURE AT THE 30-IN. LEVEL

Note that the steam condensations are different for all four radiators when the same air temperature of 68 F is maintained at the 30-in. level.

The results of tests conducted at the University of Illinois are shown in Figs. 1 and 2⁴. For the four types of radiators shown, the following conclusions are given:

³The Heating Effect of Radiators, by Dr. Charles Brabbée (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 33).

⁴Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.

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 window openings⁵.

HEATING UP THE RADIATOR

The maximum condensation occurs in a heating unit when the steam is first turned on. Fig. 3 shows a typical curve for the condensation rate in pounds per hour for the time elapsing after steam is turned into a cast-iron radiator. The data are from tests on old style column type radiators.

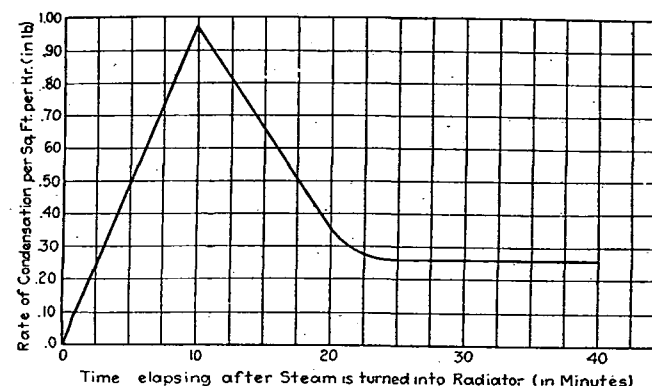


FIG. 3. CHART SHOWING THE STEAM DEMAND RATE FOR HEATING UP A CAST-IRON RADIATOR WITH FREE AIR VENTING AND AMPLE STEAM SUPPLY

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.

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. Recent investigations⁶ indicate that in the design of the enclosure three things should be considered:

⁵Effect of Two Types of Cast Iron Steam Radiators in Room Heating, by A. C. Willard and M. K. Fahnestock (Heating, Piping and Air Conditioning, March, 1930).

⁶University of Illinois Engineering Experiment Station Bulletins No. 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).

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.
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

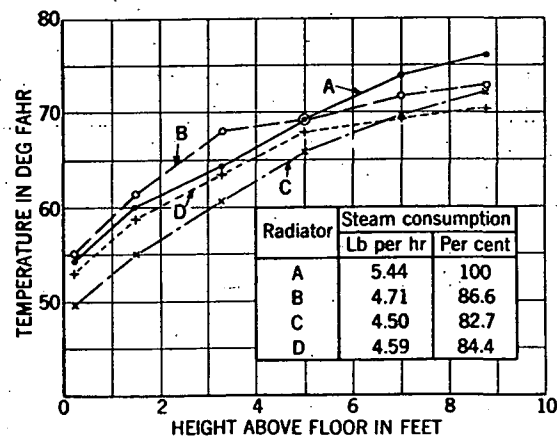


FIG. 4. STEAM CONSUMPTION OF EXPOSED AND CONCEALED RADIATORS

effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

Practically all 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. Tests⁷ 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.

CONVECTORS OR CONCEALED HEATERS

Although any standard radiator may be concealed in a cabinet or other enclosure so that the greater percentage of heat is conveyed to the

⁷University of Illinois Engineering Experiment Station Bulletin No. 230, p. 20.

room by convection thereby resulting in a form of gravity convector, generally better results are obtained with specially designed units which permit a free circulation of a larger volume of air at moderate temperatures. Since air stratifies according to temperature, moderate delivery temperatures at the outlet of the enclosure reduce the temperature differential between the floor and ceiling and accordingly accomplish the desired heating effect in the living zone.

Fig. 5 shows a typical built-in convector. The heating element consisting of a large percentage of fin surface is usually shallow in depth and

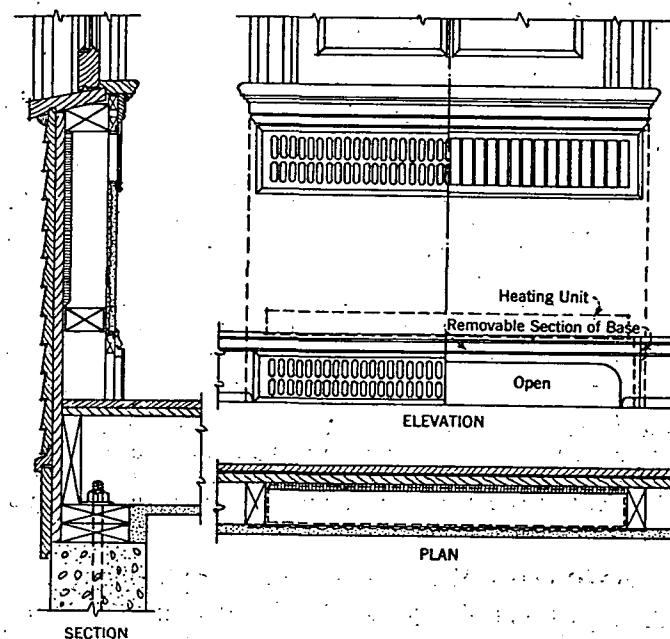


FIG. 5. TYPICAL CONCEALED CONVECTOR USING SPECIALLY DESIGNED HEATING UNIT

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 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.

The *Convactor Manufacturers Association* has adopted the A.S.H.V.E. Standard⁸ in the formulation of its ratings and has compiled a tentative

⁸A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam), (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931), (Hot Water), (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).

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 (Convactor 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 *Convactor Manufacturers Association*.

Concealed heaters or convectors are generally sold 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 catalogued 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 important 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 variable 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's. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation.

RADIATOR AND CONVECTOR SELECTION

Since the capacity of a radiator⁸ varies as the 1.3 power and a convector⁹ as the 1.5 power of the temperature difference between the inside of radiator and surrounding air it is obvious that for other than 70 F room temperatures the heat emission will be other than 240 Btu per square foot of rating. Therefore in selecting the size of radiator or convector to be used it is necessary to correct for this difference. Table 4 shows factors by which radiation requirements, as determined by dividing heat load by 240, shall be multiplied to obtain proper radiator or convector sizes from published rating tables for room temperatures ranging between 66 and 80 F as well as for 1 and 2 lb per square inch gage steam pressures. For other room and heating medium temperatures the factor is determined by the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{t_s - t_r} \right)^{1.3}$$

⁸Factors 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).

For convectors:

$$C_s = \left(\frac{215 - 65}{t_s - t_r} \right)^{1.5}$$

where

C_s = correction factor.

t_s = temperature of heating medium.

t_r = room temperature.

TABLE 4. CORRECTION RATING FACTORS

ROOM TEMPERATURE DEG F	CORRECTION FACTOR	
Steam Pressure Lb per Sq In. Gage	1	2
66	0.96	0.92
68	0.98	0.94
70	1.00	0.96
72	1.02	0.98
74	1.04	1.00
76	1.06	1.02
78	1.08	1.04
80	1.10	1.06

CODE TESTS FOR RADIATORS AND CONVECTORS

As previously indicated, the output of radiators and convectors is still designated by the terms of older practice, but this is gradually giving place to an engineering method of designating heat emission. The A.S.H.V.E. has adopted the following standards: Code for Testing Radiators (1927); Codes for Testing and Rating Concealed Gravity Type Radiation (Steam, 1932, and Hot Water, 1933).

For steam services the actual condensation weight is taken without any allowance for heating effect; for hot water services the weight of circulated water is used without allowance for heating effect. In all cases the total heat transmission varies as the 1.3 power for radiators⁸ and the 1.5 power for convectors⁹ of the temperature difference between that inside the radiator and the air in the room, and is expressed in Btu or Mb per hour.

Standard test conditions specify either a steam pressure of 1 lb gage (215 F), or hot water at 170 F and a room temperature of 70 F for radiators, or an inlet air temperature of 65 F for convectors. The heating capacity of a *steam radiator* or *steam convector* is determined as follows:

$$H_t = W_s h_{tg} \quad (1)$$

where

H_t = Btu per hour under test conditions.

W_s = condensation in lb per hour.

h_{tg} = latent heat in Btu per lb.

H_t may be converted to standard conditions of code ratings by using the proper correction factor from the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{T_s - T_r} \right)^{1.3} = \left(\frac{145}{T_s - T_r} \right)^{1.3} \quad (2)$$

For convectors:

$$C_s = \left(\frac{215 - 65}{T_s - T_i} \right)^{1.5} = \left(\frac{150}{T_s - T_i} \right)^{1.5} \quad (3)$$

The output under standard conditions will be:

$$H_s = C_s H_t \quad (4)$$

where

C_s = correction factor.

T_s = steam temperature during test, degrees Fahrenheit.

T_r = room temperature during test, degrees Fahrenheit.

T_i = inlet air temperature during test, degrees Fahrenheit.

H_s = heat emission rating under standard conditions, Btu per hour.

Similarly, for *hot water convectors*, the output under test conditions may be determined as follows:

$$H = W (\theta_1 - \theta_2) \frac{3600}{t} \quad (5)$$

where

H = Btu per hour under test conditions.

W = pounds of water handled during test.

θ_1 = average temperature of inlet water, degrees Fahrenheit.

θ_2 = average temperature of outlet water, degrees Fahrenheit.

t = duration of test, seconds.

To convert test results to standard conditions, the following correction factor is used:

$$C = \left(\frac{170 - 65}{\frac{\theta_1 + \theta_2}{2} - T_i} \right)^{1.5} = \left(\frac{105}{\frac{\theta_1 + \theta_2}{2} - T_i} \right)^{1.5} \quad (6)$$

It has been shown that when the exponent 1.5 is used the range of error is less than 3 per cent⁹ for convectors.

GRAVITY-INDIRECT HEATING SYSTEMS¹⁰

The heating units for this system are usually of the extended surface type for steam or hot water, and are installed about as shown in Fig. 6. The temperature and volume of the air leaving the register must be great enough so that in cooling to room temperature the heat available will just equal the heat loss during the same time. In cases where ventilation is a

¹⁰For further information on this subject see A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (edition of 1929) and *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. I, second edition, 1929.

requirement, the air volume needed may become so large that the entering air temperature will be but slightly above the room temperature. To establish and maintain a constant heat flow, provision must be made for removing the air in the room, after it has cooled to the desired room temperature, by a system of vent flues or ducts. As the air flow is maintained

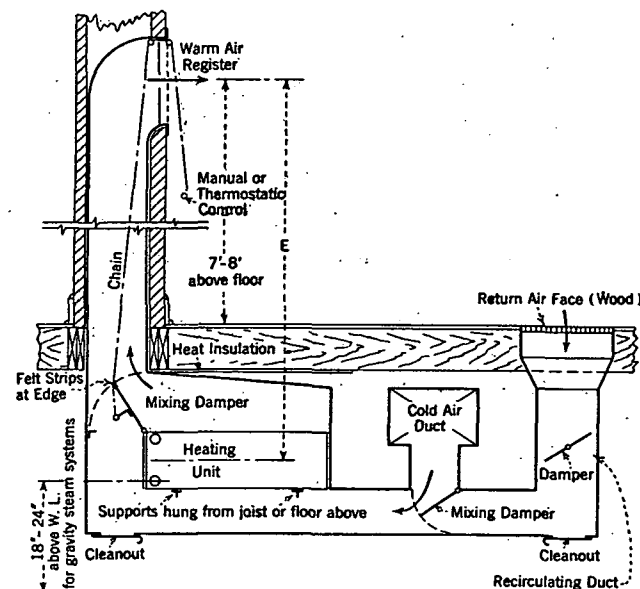


FIG. 6. GRAVITY-INDIRECT HEATING SYSTEM^a

^aSee *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. I, second edition, 1929.

by natural draft and this gravity head is very slight, it is necessary to make all ducts as short as possible, especially the runs from the heating units to the base of the vertical warm air flues. Gravity-indirect arrangements, such as illustrated in Fig. 6, are not to be generally recommended for hot water systems unless the water temperature can be maintained at a reasonably high temperature and rapid circulation of the water can be had.

PROBLEMS IN PRACTICE

1 • What is the effect on the heat output of a wall radiator when installed on the ceiling of a room?

Because the temperature differential is increased between the floor level and the ceiling when a wall radiator is placed near the ceiling, the heat output may be decreased from 5 to 10 per cent. Under such circumstances it becomes difficult to heat the living zone of a room satisfactorily.

2 • What are the principal differences between a radiator and a convector?

A radiator is commonly thought of as a commercial heating unit having a maximum amount of direct heating surface, whereas a convector is a heating device in which the extended or secondary surface may be several times that of the prime surface and which is specially designed to utilize to the fullest extent the convection principal of heating. The radiator ordinarily has vertical tubular chambers for the heating medium but most convectors have horizontal tubular chambers to which fins are attached so as to form vertical flues for the passage of air. While radiators are either exposed, enclosed, or shielded, convectors are concealed by means of a tight-fitting enclosure. Radiators are commonly made of cast-iron but convectors may be made of a combination of metals, such as copper and brass, or copper and aluminum, as well as entirely of cast iron.

3 • How did the term heating effect come into use?

It has been found that a room requiring a radiator of a certain determined capacity could under certain conditions be properly heated, with less temperature gradient between floor and ceiling and with less steam condensation, by the same radiator or by one of a different design having the same commercially rated capacity. This resulted in the use of the term *heating effect* to apply to the useful heat output of a radiator, in the comfort zone of a room, as related to the total input to the radiator.

4 • Is it necessary to make any allowance for the performance of a convector because it is enclosed?

No. The commercial ratings of convectors have been determined by testing the convectors in proper enclosures with grilles in place just as they should be installed for ordinary service.

5 • On what basis are the capacities of convectors published?

Published ratings of convectors are on the basis of equivalent square feet of direct exposed cast-iron. If any allowance is made for heating effect, the amount of such allowance is generally stated in the manufacturers' catalogs.

6 • How are fins of convectors attached to the tubes or prime surface?

Tubes or a solid core may be forced through piercings in the fins under pressure, or the tubes may be expanded into the holes through the fins. In addition a metallic bonding agent is sometimes used to insure permanent contact.

7 • What is the procedure in selecting a convector when the required amount of radiation is known?

First the limiting factor or factors of the enclosure must be determined so the available size of the wall recess can be found. Manufacturers' catalogs show capacities of convectors of each standard length and depth with varying enclosure heights. From these capacity tables, the proper convector of the required capacity can be selected for the available wall recess. If all three dimensions of the wall recess are insufficient to accommodate a convector of the required capacity, the available height and length can be maintained, but greater depth can be obtained by using a partially recessed enclosure.

8 • Given a room to be heated to 80 F with outside temperature at zero F. Assume the heat loss under these conditions to be 10,000 Btu per hour. Determine the size of the steam radiator to be installed.

A square foot of radiation is equivalent to a heat emission of 240 Btu per hour under standard conditions of steam at one pound gage pressure (215 F) and surrounding air at 70 F. With surrounding air at 80 F, the heat emission from a radiator will be less. Under these conditions, the heat emission will not be 240 Btu per square foot of catalog rating per hour, but 240 C_s .

$$C_s = \frac{(t_s - t)^{1.25}}{(215 - 70)^{1.25}} = \frac{(215 - 80)^{1.25}}{(215 - 70)^{1.25}} = 0.912,$$

and $240 C_s = 240 \times 0.912 = 218.5$ Btu. Therefore, the size of the radiator to be selected shall have a catalog rating of 10,000 divided by 218.5 or 45.8 sq ft.

Chapter 31

STEAM HEATING SYSTEMS

Gravity and Mechanical Return, Gravity One-Pipe Air-Vent System, Gravity Two-Pipe Air-Vent System, One-Pipe Vapor System, Two-Pipe Vapor System, Atmospheric System, Vacuum System, Sub-Atmospheric System, Orifice System, Zone Control, Condensation Return Pumps, Vacuum Pumps, Traps

THE essential features of the common type of steam heating systems are described in this chapter. They may be classified according to the piping arrangement, the accessories used, the method of returning the condensate to the boiler, the method of expelling air from the system, or the type of control employed. Information concerning the design and layout of steam heating systems will be found in Chapter 32.

GRAVITY AND MECHANICAL RETURN

In *gravity systems* the condensate is returned to the boiler by gravity due to the static head of water in the return mains. The elevation of the boiler water line must consequently be sufficiently below the lowest heating units and steam main and dry return mains to permit the return of condensate by gravity. The *water line difference*¹ must be sufficient to overcome the maximum pressure drop in the system and, when radiator and drip traps are used as in two-pipe vapor systems, the operating pressure of the boiler. This applies only to closed circuit systems, where the condensation is returned to the boiler. If the condensation is wasted, no water line difference is required.

In *mechanical systems* the condensate flows to a receiver and is then forced into the boiler against the boiler pressure. 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, but the relative elevation of the boiler water line is unimportant in such cases except that the head on the pump or trap discharge becomes greater as the height of the boiler water line above the trap or pump increases.

There are three general types of mechanical returns in common use, namely, (1) the *mechanical return trap*, (2) the *condensation return pump*, and (3) the *vacuum return pump*. Further information on pumps and traps will be presented later in this chapter.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

In the gravity one-pipe air-vent system each radiator has but a single connection through which steam must enter and condensation must

¹The *water line difference* is the distance between the water line of the boiler and the level of the water in the dry or wet return main. (See Fig. 4.)

return in the opposite direction. Each radiator has an individual air valve.

Up-Feed 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 of installation and its simplicity. Where the size of the system is moderate or large, it cannot be assumed that these systems will be lower in cost than two-pipe systems using steam traps. In some instances it has been found that the cost of one-pipe systems under these conditions is greater owing to the higher cost of labor and materials due to the larger pipe sizes. As will be seen from Fig. 1, the steam piping rises to a point as high as possible at the boiler and pitches downward from this location until the far end of

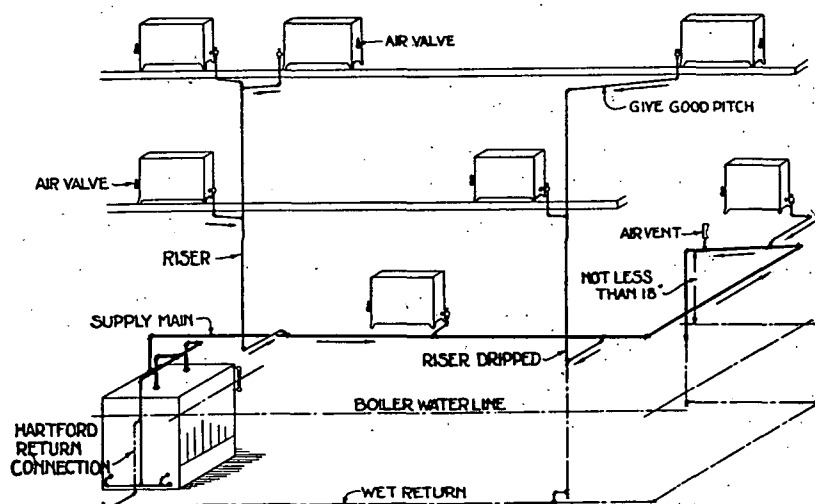


FIG. 1. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

the main or mains is reached. At the far ends drips are taken off at the low points of the steam mains, are water-sealed below the boiler water line, and then brought back to the boiler in a wet return. Single pipe risers are branched off the main or mains to feed the radiators, the steam passing up the riser and the condensation flowing down it. The steam and condensation flow in opposite directions in the riser but after the condensation enters the steam main it flows in the same direction as the steam and is disposed of through the drip connection at the end of the main. In buildings of several stories, it is customary to drip the heel of each riser separately, whereas in one- or two-story buildings this is not necessary. Both types of branches and risers are shown in Fig. 1.

Rapid elimination of air and condensation from the steam piping is essential to the successful operation of this system. It is therefore desirable that the venting and dripping of the steam main in long runs be

made at several intermediate points where the steam main may again be brought to a higher elevation.

It is desirable to install the air vent valves on the steam main about a foot ahead of the drips, as is indicated in Fig. 1 to prevent possible damage to the mechanism of the air vent valve by water, in case the valves are installed directly above the drips.

Horizontal branches to radiators and risers should be pitched at least $\frac{1}{2}$ in. in 10 ft downward toward the riser or vertical pipe, and the horizontal branches from the steam main should be graded at least this amount toward the main, except where the heel of the riser is dripped, in which case the branch should pitch down toward the riser drip (Figs. 2 and 3). The return line, if wet, may be run without pitch or may be pitched in either direction, but if it is necessary to carry the return main overhead for any distance before dropping, the return should slope downward with the flow. It is desirable to install the wet return pipe with a

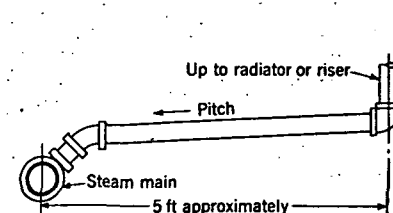


FIG. 2. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

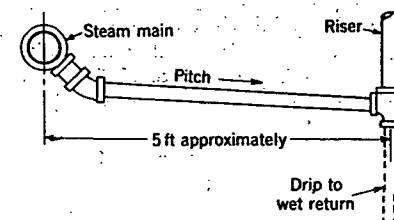


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

pitch so that the system may be drained to prevent freezing in case the building remains unoccupied for a considerable length of time.

The radiator valves may be of the angle-globe or gate type. They should not be of the straight-globe type because the damming effect of the raised valve seat interferes with the flow of condensation through the valve. Graduated valves cannot be used, as the steam valves on this system must be fully open or closed to prevent the radiators filling with water. Air valves may be manual or automatic, with or without a check to prevent the re-entrance of expelled air. Usually the automatic type is installed. An objection to one-pipe steam systems is that the heat is all on or all off, with no intermediate position possible. However, intelligent use of the on-and-off method of manual control gives reasonably satisfactory results. Improved systems and devices are now available which make it possible to obtain a modulating effect from one-pipe gravity heating systems.

It is important that the lowest points of the steam mains and heating units be kept sufficiently above the water line of the boiler to prevent flooding. Usually 18 in. is sufficient but construction limitations frequently make shorter distances necessary. The distance may be checked in the following manner:

Referring to Fig. 4 it will be seen that the water in the wet return is really in an inverted siphon, or 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 of the steam in passing from the boiler to the far end of the main. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise enough farther to produce a flow through the return into the boiler.

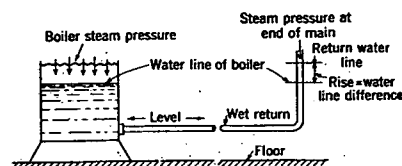


FIG. 4. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

(usually about 3 in. unless the pipes are small or full of sediment), and it will rise still farther if a check valve is installed in the return so as to obtain sufficient head to lift the tongue of the check (usually 4 in. will be necessary).

If a one-pipe steam system is designed, for example, for a total pressure drop of $\frac{1}{8}$ lb, and utilizes an Underwriters Loop² instead of a check valve on the return, the rise in the

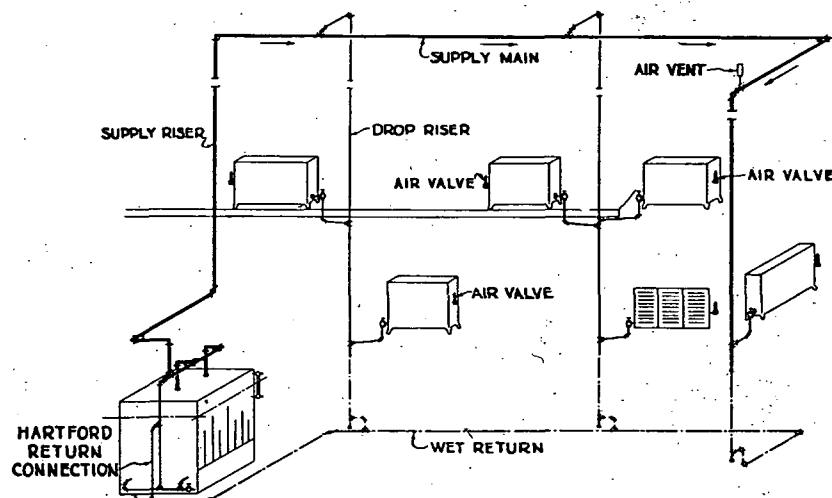


FIG. 5. TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

water level at the far end of the return due to the difference in steam pressure would be $\frac{1}{8}$ of 28 in., or $3\frac{1}{2}$ in. Adding 3 in. to this for the flow through the return main and 6 in. as a factor of safety gives $12\frac{1}{2}$ 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 $\frac{1}{2}$ lb, and with a check in the return, would require $\frac{1}{2}$ of 28 in., or 14 in., for the difference in steam pressure, 3 in. for

²See discussion of piping details in Chapter 32.

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.

Down-Feed Gravity One-Pipe Air-Vent System

In the overhead down-feed gravity one-pipe air-vent system there is no change over the *up-feed system* in the radiators, the radiator valves, the air valves, or the radiator runouts as far back as the risers. Beyond this point there are basic differences. The steam is taken from the boiler and carried to the top of the building as near the boiler as possible (Fig. 5). If the run to the main riser is long, or if the riser extends several stories in order to reach the top, the bottom of the riser should be dripped into the wet return. The horizontal main is taken off the top of the riser and grades down from the riser toward all of the drops, each drop taking its share of the main condensation (Fig. 6), or all of the drops except the last may be taken from the top of the main (Fig. 7), the last drop being from the bottom and serving as a drain for the entire main. As the overhead

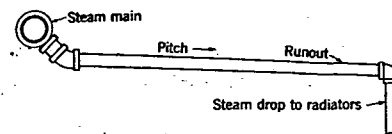


FIG. 6. STEAM RUNOUTS DRIPPING MAIN

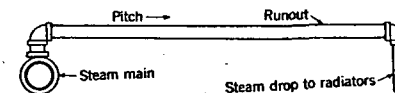


FIG. 7. STEAM RUNOUTS WITH MAIN DRIPPED AT END ONLY

main does not carry any condensation from the radiators it is immaterial which method is used. The air vent shown on the main just before the last drop (Fig. 5) may be placed at this point or it may be located at the bottom of the drop under the last radiator connection and sufficiently above the water line of the boiler to prevent flooding.

GRAVITY TWO-PIPE AIR-VENT SYSTEM

The gravity two-pipe system is now considered obsolete although many of these systems are still in use in older buildings. Separate supply and return mains and connections are required for each heating unit; air valves are installed on the heating units and mains; hand valves are installed on the returns.

Up-Feed Gravity Two-Pipe System

This system (Fig. 8) has a steam and a return connection to each radiator. The radiator valves for steam, return, and air are the same as those described for the gravity one-pipe air-vent system. The steam main is run and pitched in the same manner as in the one-pipe system, but the returns from each radiator are connected into a separate return line system which has its risers carried down and joined to a wet return line under the boiler water line level. Where the return has to be kept high to function as a dry return, it is advisable to connect the return

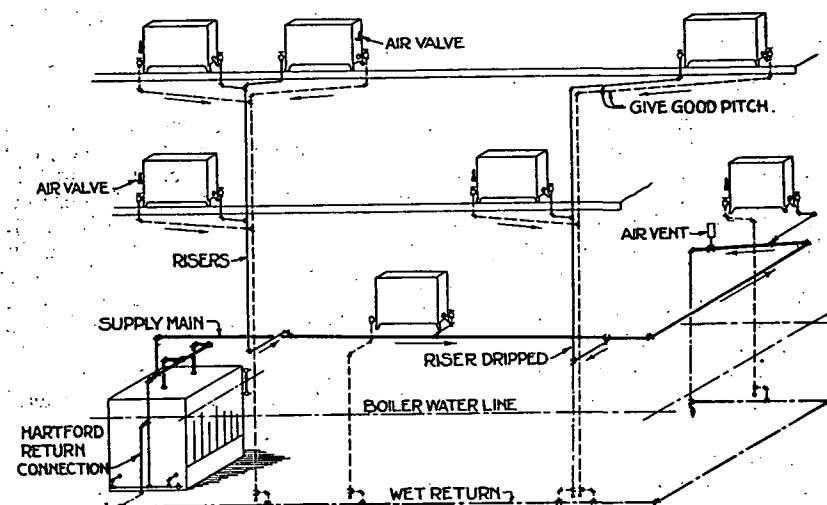


FIG. 8. TYPICAL UP-FEED GRAVITY TWO-PIPE AIR-VENT SYSTEM

risers to the dry return main through water seals about 36 in. deep, as shown in Fig. 9, to prevent steam from one riser entering another and closing the air valves on the nearest radiators.

Down-Feed Gravity Two-Pipe System

The steam main in the down-feed system is carried to the top of the building, and the piping of the steam side is arranged practically as in the down-feed one-pipe gravity system. The drips at the bottoms of the steam drops and the runouts to the radiators are similar to those shown in Fig. 8 for the up-feed gravity two-pipe system. On the return side of the system, the piping is arranged in exactly the same manner as the up-feed gravity two-pipe system.

ONE-PIPE VAPOR SYSTEM

A vapor system is one which operates under pressures at or near atmospheric and which returns the condensation to the boiler by gravity. The piping arrangement of a one-pipe vapor system is similar to that of the gravity one-pipe steam system; in fact, one-pipe gravity installations may readily be changed to one-pipe vapor systems by making a few simple alterations. The steam radiator valve is a plug cock which when opened gives a free and unobstructed passageway for water. The automatic air valve is of special design to permit the ready release of air from the radiator and to prevent the return of the air after it is expelled. The air valves on the main are a quick relief type, and the whole system is designed to operate on a few ounces of pressure.

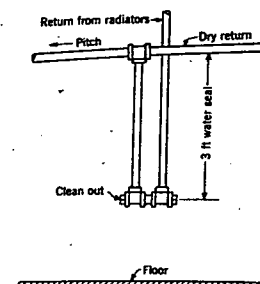
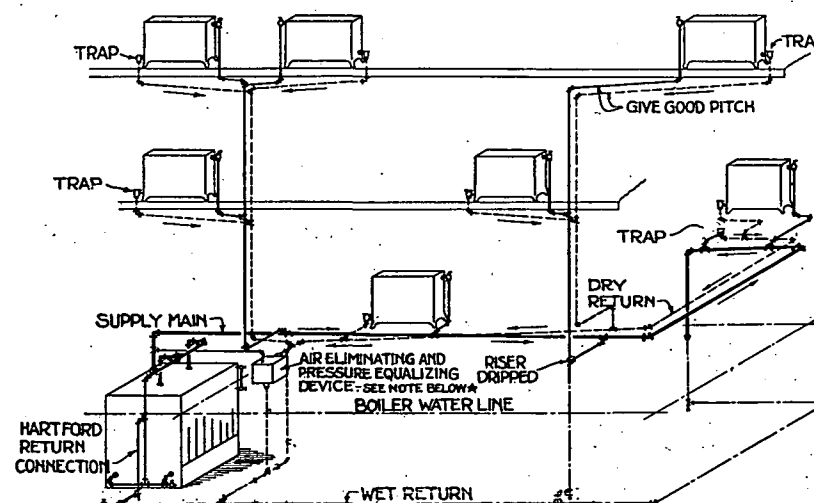


FIG. 9. METHOD OF CONNECTING TWO-PIPE GRAVITY RETURNS TO DRY RETURN MAIN

TWO-PIPE VAPOR SYSTEM

Two-pipe vapor systems may be classified as (1) *closed systems* consisting of those which have a device to prevent the return of air after it is once expelled from the system, and which can operate at sub-atmospheric pressures for a period of four to eight hours depending upon the tightness of the system and rate of firing, and (2) *open systems* consisting of those which have the return line constantly open to the atmosphere without a check or other device to prevent the return of air, and which operate at a few ounces above atmospheric pressure. The open systems have the disadvantage of not holding heat when the rate of steam generation is diminishing.

Under the first classification the essentials are packless graduated

FIG. 10. TYPICAL UP-FEED VAPOR SYSTEM WITH AUTOMATIC RETURN TRAP^a

^aProper piping connections are essential with special appliances for pressure equalizing and air elimination.

valves on the radiators, thermostatic return traps on the returns, and traps on all drips unless they are water sealed. Such a system, illustrated in Fig. 10, should be equipped with an automatic return trap to prevent the water from backing out of the boiler. In this up-feed arrangement the supply piping is carried to a high point directly at the boiler and is graded down toward the end or ends of the supply main, each supply main being dripped at the end into the wet return or carried back to a point near the boiler where it drops down below the boiler water line and becomes a wet return. From this main, runouts are branched off to feed risers or radiators above, these being graded back toward the steam main

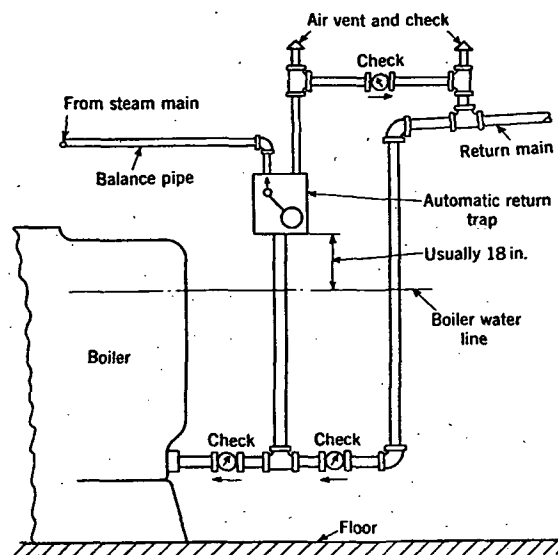


FIG. 11. TYPICAL CONNECTIONS FOR AUTOMATIC RETURN TRAP

if they are not dripped at the bottom of the riser, or toward the riser if the riser heel is dripped. Both conditions are illustrated in Figs. 2 and 3.

Return risers are connected to each radiator on its return end through thermostatic traps. Their bottoms are connected to the return main through runouts which slope toward the main. The return main itself is sloped back toward the boiler if it is carried overhead; if run wet, the slope may be neglected, although it is desirable to slope the pipe so that the system may be drained. An air vent is installed at the point at which the return main drops below the water line. In the simplest cases this vent consists of a $\frac{3}{4}$ -in. pipe with a check valve opening outward, but certain systems employ special patented forms of vent valves, designed to allow the air readily to pass out of the system and to prevent its return. 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 usually

is 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. 11).

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 branches are taken off the main from the bottom or at a 45-deg angle downward, with the runouts

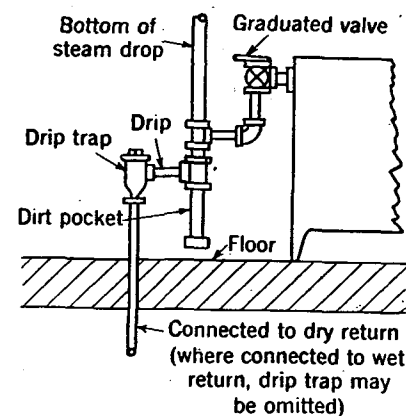


FIG. 12. DETAIL OF DRIP CONNECTIONS AT BOTTOM OF DOWN-FEED STEAM DROP

sloped toward the drops (Fig. 6). Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop. Another method of running the steam main, which is not considered as satisfactory but which is practical, is to take the branches off the top of the main (Fig. 7) and to drip the end of the main through the last riser, as illustrated in the down-feed one-pipe system detail shown in Fig. 6. If this is done, the pipe drop at the end or ends of the mains should be enlarged one pipe size to provide capacity for this concentration of the main drip.

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 the steam drops should terminate with a dirt pocket above which a drip trap

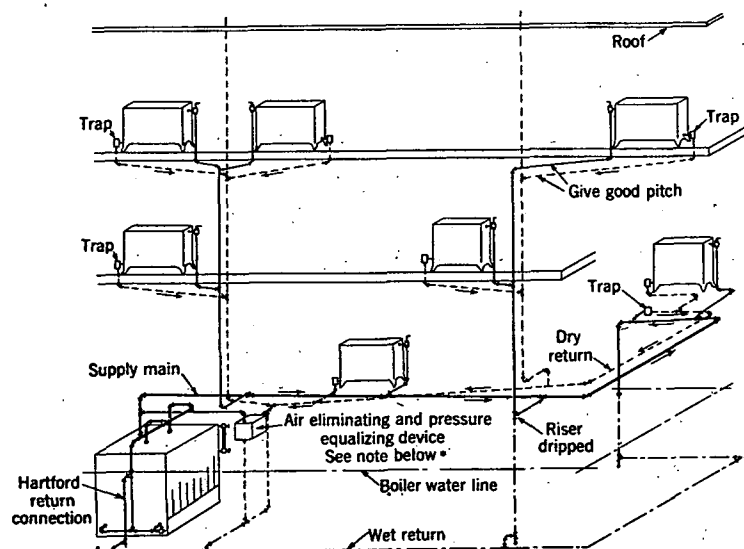


FIG. 13. TYPICAL ATMOSPHERIC SYSTEM WITH AUTOMATIC RETURN TRAPS

*Proper piping connections are essential with special appliances for pressure equalizing and air elimination

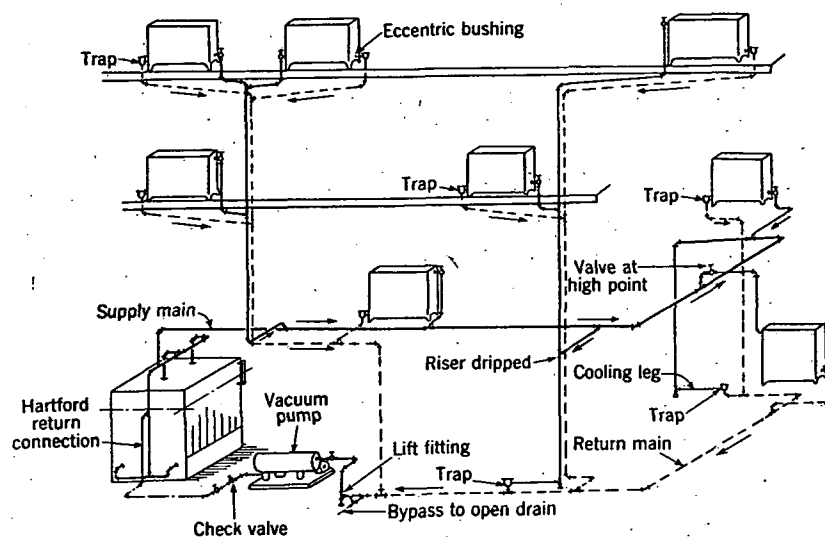


FIG. 14. TYPICAL UP-FEED VACUUM PUMP SYSTEM

connection is located, as shown in Fig. 12. The returns on a down-feed vapor system are the same as on an up-feed system except that every steam drop must have a drip at the bottom connected either into the return through a trap or into a separate water-sealed drip line below the boiler water line, as illustrated in Fig. 10, in which case the thermostatic traps may be omitted. The runouts to the radiators and the radiator connections of the down-feed system are the same as those of the up-feed system already described.

ATMOSPHERIC SYSTEM

The distinguishing features of the atmospheric system are gravity return to the boiler or to waste, graduated or ordinary radiator valves, no automatic air valves on the radiators, thermostatic traps on the radiator returns, and the venting of all air from the system by means of pipes open to the atmosphere. The returns are open to the atmosphere at all times, usually by extending the return risers to the top of the building where they are either connected together in groups and carried through the roof or extended through the roof individually. Atmospheric systems, either up-feed or down-feed, are often used where the condensation is not returned to the boiler, as in heating systems supplied by high pressure steam through pressure-reducing valves at locations far from the boilers. The returns may be delivered back to the boiler, if desired, by condensation return pumps which are vented to the atmosphere. The return lines in such systems are simply gravity waste lines in which the condensation flows entirely by gravity and is not aided by any pressure difference.

Atmospheric systems contemplate maintaining a practically constant pressure in the steam pipe and atmospheric pressure in the return pipe. When graduated steam valves are provided, they enable the occupant of a room to vary the flow area to the radiator so as to obtain a greater or lesser heating effect.

The steam side may be run as that for either up-feed or down-feed two-pipe vapor systems, as the conditions require, and the radiator connections are the same as for vapor systems in that they have graduated valves on the radiator supply ends and thermostatic traps on the radiator return ends. All drips from the supply main and the steam side of the system must pass through thermostatic drip traps before entering the return system where only atmospheric pressure exists. Fig. 13 illustrates a typical scheme of piping used on atmospheric systems.

VACUUM SYSTEM

In the vacuum system, a vacuum is maintained in the return line practically at all times but no vacuum is carried on the steam side, and the usual accessories include graduated valves on the radiator supply and thermostatic traps on the radiator return. The air is expelled from the system by a vacuum pump and all drips must pass through thermostatic traps before connecting to the return side of the system.

These systems are often fed from high pressure steam mains through pressure-reducing valves but they may be fed direct from a low-pressure steam heating boiler as shown in Fig. 14, in which a typical up-feed

vacuum system is illustrated. The supply main slopes down in the direction of flow; the runouts pitch down toward the riser if the riser is dripped (Fig. 3) or up toward the riser if the riser is not dripped (Fig. 2); both conditions are indicated in Fig. 14. The matter of dripping the risers depends largely on the height of the riser and the judgment of the designer. Ordinarily risers less than three stories high are not dripped and those more than four stories high are dripped, but there is no set rule for this. When risers are dripped the runouts from the steam main may be taken from the bottom if desired and each runout then serves as a drip for the main.

The risers are carried up to the highest radiator connection and are connected to the radiator through runouts sloping back toward the riser. The radiators usually have graduated valves on the supply end, although this is not absolutely necessary. Angle-globe valves and gate valves may be used where graduated manual control is not desirable. The return valves must be of the thermostatic type which will pass air and water but which will close against the passage of steam.

The return risers are connected in the basement into a common return line, which slopes downward toward the vacuum pump. The vacuum pump discharges the air from the system and pumps the water back to the boiler, or other receiver, which may be a feed-water tank or a hot well. It is essential on these systems that no connection from the supply side to the return side be made at any point except through a trap.

While the best practice demands a return flowing to the vacuum pump in an uninterrupted downward slope, in some cases limitations make it necessary to drop the return below the level of the vacuum pump inlet before the pump can be reached. In such event one of the advantages of the vacuum system is that the return can be raised by the suction of the vacuum pump to a considerable height, depending on the amount of vacuum maintained, by means of a lift fitting inserted in the return. Best practice dictates that the lift should be limited to a single lift connection at the entrance to the vacuum pump and that lifts scattered throughout the system be avoided. When the lift is considerable, several lift fittings are used in steps (Fig. 15), more successful operation being obtained by this method than when the lift is made in one step. If the lift occurs close to the vacuum pump, a special arrangement is used as shown in Fig. 16. It is desirable that means be provided for draining manually the low points of the lift fittings to eliminate from the return piping all water in danger of freezing in case the system is shut down for a considerable length of time.

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 (Fig. 6) 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. 17). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 18.

SUB-ATMOSPHERIC SYSTEMS

Sub-atmospheric systems are similar to vacuum systems, but in contrast provide temperature control by variation of the heat output from the radiators both by varying the pressure at which steam is circulated in the radiation and the amount of steam. The steam supply is continuous

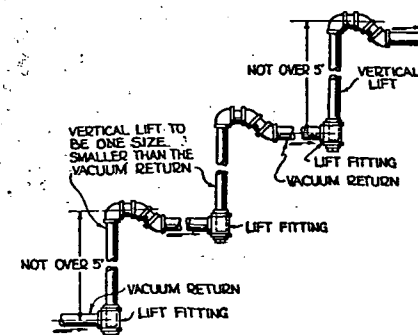


FIG. 15. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

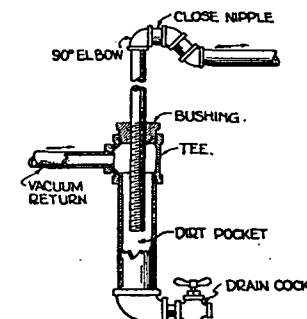


FIG. 16. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP



FIG. 17. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

at varying rates. A vacuum pump capable of operating at high partial vacua is preferable since the higher the vacuum the greater is the accuracy in the distribution of steam through the system, particularly in mild weather. A pump capable of producing up to 25 in. of vacuum on the system is used in such cases. A controller is placed on the pump so that the vacuum or absolute pressure carried in the returns can be maintained at a certain amount below that existing in the line to insure circulation.

The traps are designed to operate in high vacuum. It is apparent that this system differs from the ordinary vacuum system by having a vacuum on both sides of the system, instead of only on the return side, in order to secure control of the heat emission from the radiators and thus to control the temperature in the building. These systems permit the heat output from the steam mains and risers to be diminished as the weather becomes milder, thus giving control to this portion of a heating system. The system can be operated in the same manner as the ordinary vacuum system when desired.

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, steam pressure exists in the steam main and radiators only during the most severe weather, while under average winter temperatures the steam is under a partial vacuum which in mild weather may reach as high as 25 in. after which further reduction in heat output is obtained by partially filling the radiation with steam.

This vacuum is partially self-induced by the condensation of the steam in the system due to the supply of steam being furnished through the control which admits it, and it being proportioned to balance the existing heat loss. In the sub-atmospheric system, a control valve is inserted on

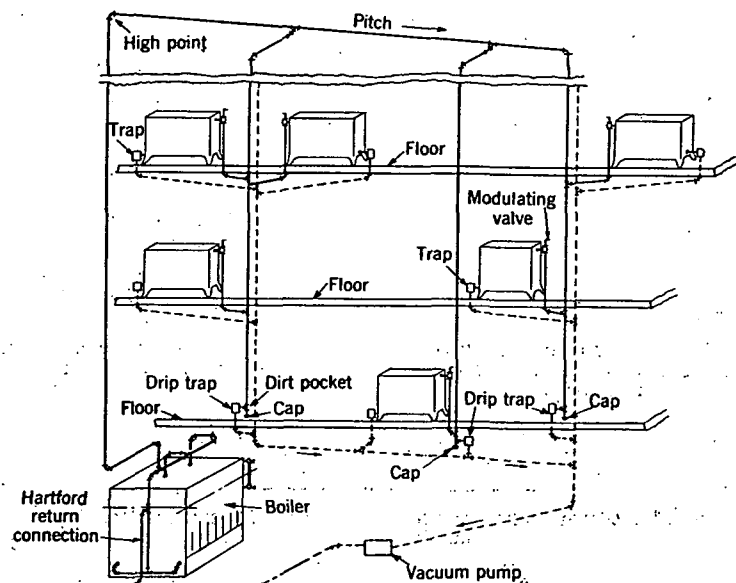


FIG. 18. TYPICAL DOWN-FEED VACUUM SYSTEM

the steam main of an ordinary vacuum system near the boiler or the boiler is automatically controlled, a high-vacuum pump is substituted for the ordinary type and is supplied with a pressure-difference control, and traps are placed on the radiators and drips which will operate satisfactorily at any pressure from 5 lb gage to 26 in. of vacuum.

The control valve is either a special pressure-reducing valve which may be controlled manually, or a control valve or combustion equipment which may be operated thermostatically from points selected in the building. The vacuum pump regulator is simply a diaphragm so arranged that, when the vacuum in the return line is insufficient to hold the desired difference in pressure between the steam and return sides of the system, the vacuum pump is automatically started and the vacuum increased to the necessary amount. The actual pressure difference main-

tained between the two sides of the system is only enough to secure adequate circulation and is often about 2 in. of mercury. This fixed pressure difference between the supply and return sides of the system results in practically constant circulation under all pressure conditions.

In order to distribute the steam equally when the system is being warmed up and also to reduce the amount of steam delivered to the radiators on mild days, orifice plates are used in the graduated radiator control valves. The heat emitted from the radiators in mild weather and under conditions of high vacuum is not only reduced in proportion to the difference in the steam temperature between that for 2 lb gage and for 25 in. of vacuum but it is reduced still further by a reduction in the amount of steam which can pass through the orifice when the steam is expanded due to the vacuum. This renders possible the control of heat emission from the radiators to a point not indicated entirely by the difference in steam temperatures, but far beyond it.

Sub-atmospheric operation has advantages even where individual thermostatic radiator control is installed. By operating the system with steam temperatures in parallel with the outside temperature requirements, a large part of the load is removed from the temperature control system, it makes fewer operations and the radiator follows an even temperature without fluctuating from extreme hot to extreme cold.

The high-vacuum pumps on this system are equipped with receivers having float control so that the pump can be placed on a receiver-return-pump basis at night if desired so no high vacuum will be carried. One radical difference between this system and the ordinary vacuum system is that no lifts can be made in the return line, except at the vacuum pump. The returns must grade downward constantly and uninterruptedly from the radiator return outlet to the inlet on the high-vacuum pump receiver.

No attempt should be made to heat service water on this system unless the steam line for water heating is taken off the boiler header back of the heating system control valve, and then only when 2 lb or more will be carried on the boiler at all times.

ORIFICE SYSTEM

Orifice systems of steam heating may have piping arrangements identical with vacuum systems but some of these systems omit both the radiator thermostatic traps and the vacuum pump in cases where the returns are wasted to a sewer or delivered to some type of receiver in which no back pressure exists. The principle on which they operate is embodied in the well-known fact that an orifice will deliver varying velocities 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 velocity will be obtained.

As a result, if an orifice is so designed in size as to exactly fill a radiator with steam at 2-lb gage on one side and $\frac{1}{4}$ -lb gage on the other, the absolute pressure relation is

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 90 \text{ per cent}$$

Should the steam pressure be dropped to $\frac{1}{4}$ -lb gage, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be seen that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, it is simply a question of dropping this main pressure provided the supply pipe pressures be controlled sufficiently closely, so as to fill any desired portion of the radiator down to the point where the main pressure equals the back pressure in the radiator, at which time no steam will be supplied at all. If orifices throughout a system are designed on a similar basis, all radiators will heat proportionately to the steam pressure within the limits for which the orifices are designed.

Some systems use orifices not only in radiator inlets but also at different points on the main, thus balancing the system to a greater extent. For example, the system may be designed for a particularly long run involving an initial pressure of 3-lb gage on the main and 2 lb at the end of the main, but each branch from the main may have an orifice for reducing the pressure at it to 2 lb-gage. This is particularly useful for branches near the boiler where the drop in the main has not yet been produced.

Orifice systems using a vacuum pump operate successfully with the ordinary low vacuum type of pump producing 8 to 10 in. of vacuum. They are controlled by various means to regulate the steam pressure. One method is by a thermostat located on the roof to govern the steam pressure by a combination of outside and inside temperatures; another, useful on systems without traps and vacuum pumps, controls the steam pressure manually from temperature indication stations in the building, or automatically by a thermostatically-controlled pressure reduction valve or draft regulator on the boiler; with oil or gas firing, the on-and-off control or a boiler pressure control may be used.

ZONE CONTROL

Certain portions of a building may require more heat at times than others but if the whole building is on one general control, such as would occur with a single piping system with an on-and-off control or with the sub-atmospheric or the orifice systems, it would be necessary to supply sufficient heat to accommodate the coldest portion of the building even though some sections would be overheated. By separation of a building into zones each with its own piping system, each zone of the building may be controlled separately.

The sides of the building with different exposures should be considered first, because of the varying effects of the wind and sun. With the prevailing winter winds from the northwest, a simple zoning would place the north and west sides of the building on one system and the south and east sides on another. If the building is large enough to justify the expenditure, a better arrangement would be to place all north walls on one zone, all west walls on a second, all east walls on a third, and all south walls on a fourth.

In case of high buildings, the lowest 8 or 10 stories may be well protected from wind by surrounding buildings, the next 10 stories may have moderate exposure, and above this there may be an unobstructed exposure to gales. On still days the heat demands vertically will vary little, but on

windy days there will be a marked difference in the heat requirements for the different horizontal sections. In addition, the *chimney effect* caused by the difference in density between the warm air on the inside of a building and the colder air on the outside will give an air movement which will require zoning to correct. Where such conditions are encountered, the building should be divided horizontally as well as vertically. An arrangement of this character would give 12 zones: namely, north, east, south, and west lower zones; similar middle zones; and similar top zones. Each zone should constitute an individual and separate system of piping with its own supply steam valve (controlled by thermostats in its respective zone) and with its own return or vacuum pump, if one is used. Certain interior areas, such as basements, light well walls and other locations where sun and wind do not affect the conditions, should be placed in still another zone if the most economical results are to be secured.

Zoning has advantages even where individual thermostatic radiator control is installed whether this be of pneumatic, electric, or the self-contained radiator valve type. By operating each zone to supply heat in parallel with its outside temperature and wind fluctuations, a large part of the load is taken off the thermostatic controls; they operate less frequently and the radiators follow a more even temperature instead of fluctuating from extreme hot to extreme cold.

Sub-atmospheric, orifice, and zone control systems, generally are proprietary.

CONDENSATION RETURN PUMPS

Condensation return pumps are generally required when the elevation of the boiler with respect to the heating units is such that the condensate will not return by gravity, or when the boiler pressure is greater than that supplied the heating units, as in a high-pressure boiler installation supplying steam through a reducing valve to the heating units. The condensate is commonly returned by gravity to a receiver, vented to the atmosphere, from which it flows to the pump.

Condensation return pumps are assembled with tank or receiver and arranged for either continuous operation or for automatic starting and stopping by float control. Any style of water pump may be employed for this service, the power available determining whether the mode of drive shall be steam or electric. The motor-driven, automatic, centrifugal pump and receiver has found wide acceptance for low pressure heating systems.

Fig. 19 shows a typical installation using an automatic condensation return pump and vented receiver. A float control operates the pump whenever sufficient water accumulates in the receiver. Condensation return pumps are suitable for use on systems in which the returns are under atmospheric pressure. These include atmospheric systems, orifice systems with open returns, and certain types of vapor systems which operate within a few ounces of atmospheric pressure, but ordinarily do not carry any sub-atmospheric pressure. They may also be used on one-pipe and two-pipe gravity steam systems with a proper arrangement for venting the receiver. In discharging to waste, there is no object in using a condensation pump unless the discharge must be elevated.

VACUUM PUMPS

A vacuum heating pump is employed to create a vacuum on the return side of a system to remove air and water from the return piping and to pump the condensate to the boiler or to a receiving tank. Pumps of this classification may be driven by steam or electricity; they may be continuous in operation, or automatic with float or vacuum control in one or more combinations.

For rating purposes³, vacuum pumps are classified as *low vacuum* and *high vacuum*. Low vacuum pumps are those rated under operation at $5\frac{1}{2}$ -in. mercury vacuum, and high vacuum pumps are those rated at vacuums above $5\frac{1}{2}$ in.

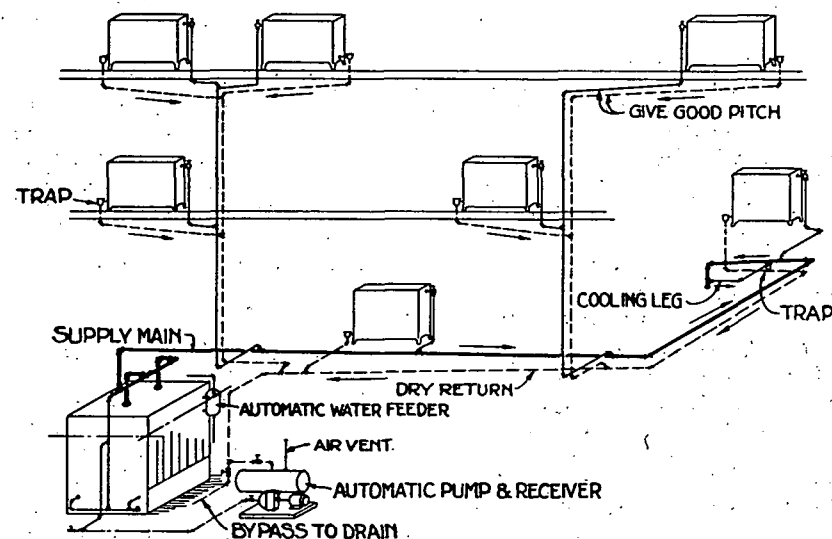


FIG. 19. TYPICAL INSTALLATION USING CONDENSATION PUMP

Return line vacuum pumps are classified in the method of their performance as follows:

- Those which perform the function of air separation under atmospheric pressure.
- Those which perform the function of air separation under a partial vacuum.

Pumps coming under the first classification will handle vacuum steam system condensation coming back by gravity at any temperature up to 205 F without either the sealing or the hurling water flashing into steam. These pumps, to operate under a combined water level and vacuum control, must be equipped with a float-control receiver between the vacuum pump and the system, but where they are intended for continuous operation, they do not require a receiver. Such pumps employ a single vacuum

producer which removes the condensate and air from the system and delivers it into a separating chamber under atmospheric pressure from which the condensate is delivered to the boiler or feed water heater. They are constructed on one of the following evacuating and discharge principles:

1. Hydraulic vacuum producer with one pump impeller.
2. Hydraulic vacuum producer with two pump impellers.
3. Water displacement vacuum producer with two pump impellers.
4. Piston displacement vacuum producer with one pump piston.

The second classification of pumps will handle vacuum steam system condensation coming back by gravity at any temperature not exceeding 190 F without the flashing into steam of either the sealing or the hurling water. In order to operate under a combined water-level and vacuum control, these pumps must be equipped with a float-control receiver between the vacuum pump and the system; where intended for continuous operation they do not require a receiver. Such pumps employ a vacuum producing impeller which removes air from the receiver or heating system under a partial vacuum and delivers it through an air separator against atmospheric pressure. The condensate is removed from the receiver under a partial vacuum by a separate impeller and is delivered to the boiler or feed water heater. For evacuating and discharge, a water displacement vacuum producer with two pump impellers is used.

Receiver Capacities for Vacuum Pumps

Where receivers are used in connection with vacuum pumps there is a definite relation between the capacity of the receiver and the capacity of the pump. The receiver should have a capacity of not less than $1\frac{1}{2}$ times the volumetric quantity of condensation per minute and should not have such a capacity that the pump will empty the receiver in less than half a minute. Receivers of larger capacities will result in less frequent periods of operation.

Piston Displacement Vacuum Pumps

Piston displacement return-line vacuum heating pumps may be either power or steam driven. They should be provided with mechanical lubricators and their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. While the volumetric displacement for such pumps was formerly figured at 8 to 10 times the volumetric flow of condensation to be handled, the more efficient thermostatic traps used today in connection with vacuum heating systems make it possible to change this proportion so that the volumetric displacement of these pumps may not be less than 6 times the volume of condensation.

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 which cuts out when the vacuum has been increased to the highest point. This is done largely to eliminate the constant starting and stopping of the vacuum pump which would occur if the vacuum were

³See A.S.H.V.E. Standard Code for Testing and Rating return line low vacuum heating pump.

maintained constant. In addition to this control, a float control is included which will automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum in the system. This arrangement makes the vacuum pump primarily a condensation pump and secondarily an air pump.

On the sub-atmospheric systems the high vacuum pump is controlled by a differential regulator which keeps the vacuum in the return line always a few inches higher than that in the steam line and in the radiators.

TRAPS

Traps are used for draining the condensate from radiators, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. The usual functions of a trap are to allow the passage of condensate and to prevent the passage of steam. In addition to these functions, traps are frequently required to allow the passage of air as well as condensate. Traps are also required to allow the passage of air and to prevent the passage of either water or steam, or both.

In addition, 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. If the static head is in favor of the trap discharge it is a minus quantity and may be deducted from the other factors of the discharge head.

Traps may be classified according to the principle of operation as (1) float, (2) bucket, (3) thermostatic, or (4) 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 indicates 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 this is 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*.

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.

Thermostatic traps are generally used for draining radiators and heaters, except for very large capacities where bucket, float or blast-type

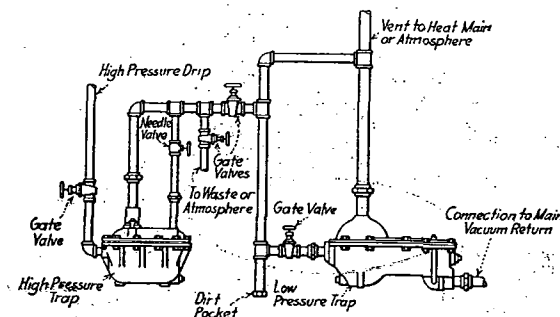


FIG. 20. METHOD OF DISCHARGING HIGH-PRESSURE APPARATUS INTO LOW-PRESSURE HEATING MAINS AND VACUUM RETURN MAINS THROUGH A LOW-PRESSURE TRAP

thermostatic traps are used. Thermostatic traps for this service usually pass both condensate and air and in the case of float and upright bucket traps the air is usually relieved through an auxiliary thermostatic trap in a by-pass around the main trap. Sometimes this auxiliary air trap is an integral part of the trap. Such traps are termed float and thermostatic traps.

Blast-type thermostatic traps are sometimes used on vacuum heating systems for connecting old one- or two-pipe gravity systems in parallel with vacuum return line systems, in which case the blast-type thermostatic traps should not be provided with auxiliary air by-pass, as the action of this will allow the vacuum to draw air into the old system through its air valves, especially when the steam is wholly or partially cut off. The air from the returns of such old systems should be relieved just ahead of the traps by means of quick-venting automatic air valves, preferably of the non-return type, especially if the other air valves on the old system are non-return valves.

Tilting traps used for discharging to a higher or a lower pressure are

provided with two or three valves operated by the action of the trap. In the case of the two-valve tilting traps, one valve closes a steam inlet and the other valve opens a vent outlet while the trap is filling, and as soon as the trap dumps, the first valve opens the steam inlet and the

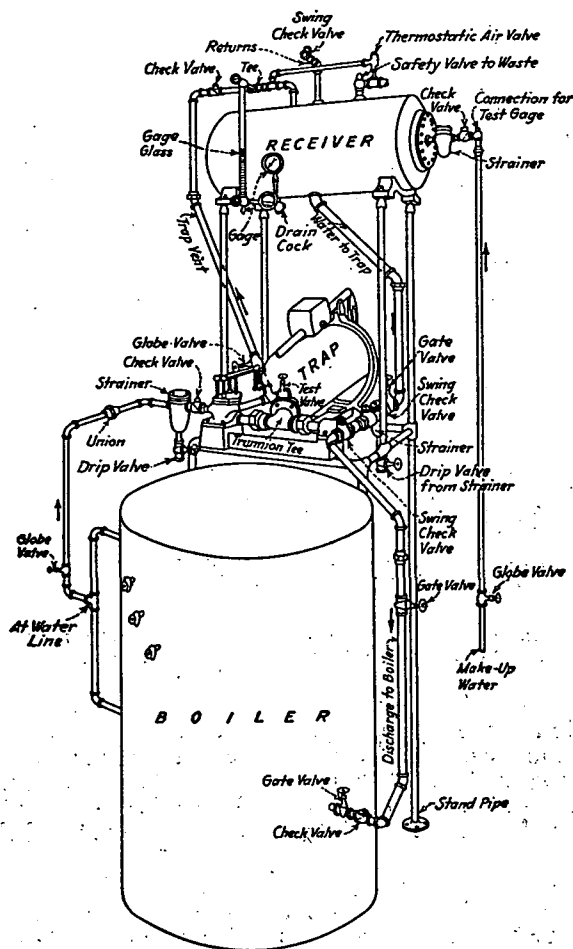


FIG. 21. RETURN TRAP AND RECEIVER FOR AUTOMATIC BOILER FEED

second valve closes the vent outlet, while the trap discharges. In this type of trap there must be a swinging check-valve on each side of the trap, in addition to the usual by-pass, to prevent the pressure in the trap, while discharging, from backing up through the inlet and the pressure in the discharge line from backing up into the trap while it is filling. This

type of trap will blow steam out through the vent while filling, if the pressure on the inlet side is sufficient, and should not be used, therefore, with such pressures unless the vent is properly piped back into the return to a feed water heater, a condenser or a perforated pipe in the bottom of the receiver to which the trap discharges in such a way as to prevent the escape of the steam that comes in with the condensate and passes through the vent. In the three-valve traps of this type there is an extra valve for closing the discharge while the trap is filling.

High pressure traps should not discharge directly into a vacuum return because of the vapor formed by the re-evaporation of a part of the hot condensation. Fig. 20 shows a method which may be used for disposing of the greater part of the vapor of re-evaporation. An expansion chamber often is installed between the high- and low-pressure traps.

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 counterbalanced, 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. Fig. 21 shows a direct return tilting trap and receiver properly connected for automatically feeding a boiler from a system of returns delivering the condensate to the receiver.

PROBLEMS IN PRACTICE

1 ● What is meant by water line difference in a gravity steam heating system?

The water line difference is the distance between the level of the water in the dry or wet return and the boiler water line. This difference is equivalent to the pressure required to overcome the maximum drop in the system and the operating pressure of the boiler.

2 ● How many types of common mechanical returns are there and what are they?

Three: (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum pump.

3 ● In the ordinary vacuum system of steam heating, where does the vacuum usually exist?

On the return side of the system only, between the radiator trap and the vacuum pump. If the radiator supply valve is closed off, the vacuum may extend back through the radiator as far as the supply valve; if an adequate supply of steam is furnished to the system, some vacuum may be developed in the steam main, but neither of these can be termed *normal operation*.

4 ● What is the distinction between the open and the closed vapor systems?

The open vapor system has the return line always open to the atmosphere, while the closed vapor system has an automatic device on the air vent so that air once expelled from the system through the vent cannot re-enter via this route.

5 ● On a vacuum system, what device must be placed on all drips before they enter the vacuum return line?

A thermostatic drip trap or occasionally, where large volumes of condensation are to be handled, a float trap.

6 ● How does the sub-atmospheric system differ in operation from the ordinary vacuum system?

The ordinary vacuum system has pressure in the steam line, and a vacuum produced by the vacuum pump in the return line, usually varying between 5 and 10 in. of water. The sub-atmospheric system may have either a vacuum or pressure on the steam and return lines, but a constant difference in pressure is maintained between the lines regardless of what pressure or vacuum may be carried. The vacuum, which is generally produced by condensation in the system under conditions of throttled steam supply, may run much higher than in the ordinary vacuum systems.

7 ● What is generally understood by zoning in building steam heating systems?

Zoning is a term applied to the placing of certain sections of a building on a single temperature control instead of having either individual room control or a single temperature control governing the whole building. Zones may be horizontal, such as a single story, a basement, or an attic, or vertical such as the north side, or the west side.

8 ● Why does the water line in the far end of a wet return in a gravity steam system rise higher than the water line in the boiler?

The friction of the steam flowing through the steam main from the boiler to the far end of the system causes a drop in steam pressure at the point where the wet return is connected; consequently, the steam pressure on top of the water in the wet return is less than the steam pressure on top of the water in the boiler, so the water in the end of the wet return rises until a balanced condition is set up.

9 ● On gravity one-pipe systems as indicated in Fig. 1 and Fig. 3, why is the drip on the steam runout connected to wet return?

Because if it were connected to dry return, the pressure drops to two different points would not necessarily be the same and the system would short circuit.

10 ● What is the function of the automatic return trap?

To insure the return of condensate to the boiler when the operating condition is such that the boiler pressure exceeds the static head on the returns.

11 ● What advantage is there to an air valve with a check to prevent the re-entrance of expelled air?

A system equipped with such valves builds up a vacuum and holds the heat longer. With proper controls on the boiler, lower radiator temperatures can be maintained in mild weather, giving better plant efficiency.

Chapter 32

PIPING FOR STEAM HEATING SYSTEMS

Flow of Steam in Pipes, Pipe Sizes, Tables for Pipe Sizing, Sizing One-Pipe Gravity Air Vent Systems, Two-Pipe Gravity Air Vent Systems, Two-Pipe Vapor Systems, Atmospheric Systems, Vacuum Systems, Sub-Atmospheric Systems, Orifice Systems, High Pressure Steam, Expansion in Steam and Return Lines, Piping Connections and Details, Boiler Connections, Hartford Return Connection

THE design of a steam heating system should be considered under four headings, namely, (1) the details of the heating units, (2) the arrangement of the general piping scheme, (3) the details of connections, and (4) the sizing of the lines. Items 1 and 2 are covered in Chapters 30 and 31, respectively, while this chapter considers the two latter items.

The functions of piping are to supply the heating units with steam and to remove the condensation. In some systems both the air and condensation are removed from the heating units by the return piping. To accomplish this effectively, the distribution of the steam should be efficient and equitable, without noise, and the returns should be as short as possible. When air is handled its escape should be facilitated to the utmost since an air-bound system will not heat properly. Condensation takes place in a steam system not only in the heating units, but throughout the piping system as well, and the returns also condense any steam or vapor that may be contained. At the same time part of the condensation may flash back into steam when the vacuum or pressure in the return is considerably below the steam pressure.

It is essential that steam piping systems not only distribute steam at full load but also at partial loads, as the average winter demand is less than half of the demand in most severe outside temperatures. Furthermore, in heating up rapidly the load on the steam main may exceed the maximum operating load even in extreme weather, due to the necessity of raising the temperature of the metal in the system to the steam temperature. This may require more heat than would be emitted from the system itself after it once is thoroughly heated.

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 following formula:

$$P = 0.0000000367 \left(1 + \frac{3.6}{d} \right) \frac{W^2 L}{D d^5} \quad (1)$$

or

$$W = 5220 \sqrt{\frac{P D d^5}{\left(1 + \frac{3.6}{d} \right) L}} \quad (2)$$

where

 P = loss in pressure, pounds per square inch. d = inside diameter of pipe, inches. L = length of pipe, feet. D = weight of 1 cu ft of steam. W = weight of steam flowing per hour, pounds.

Example 1. How much steam will flow per hour through 100 ft of 2-in. pipe if the initial pressure is 1.3 lb per square inch and the pressure drop is 1 oz?

Solution. $P = \frac{1}{16} = 0.0625$ lb; $d = 2.067$ in. (Table 1, Chapter 34); $L = 100$ ft; $D = 0.04038$ lb (Table 6, Chapter 1). Substituting these values in Formula 2:

$$W = 5220 \sqrt{\frac{0.0625 \times 0.04038 \times 2.067^5}{\left(1 + \frac{3.6}{2.067} \right) 100}} = 97.2 \text{ lb per hour.}$$

Formula 2 does not allow for entrained water in low-pressure steam, condensation in pipe, and roughness in commercial pipe as found in practice.

The latent heat of steam (h_{fg}) at atmospheric pressure (Table 6, Chapter 1) is 970.2 Btu per pound. Inasmuch as the heat emission of an equivalent square foot of heating surface (radiation) is 240 Btu, 1 lb of steam at this pressure will supply $\frac{970.2}{240}$ or 4.04 sq ft of equivalent heating surface. This figure is usually taken as 4 even. In Example 1, the weight of steam flowing per hour would therefore supply 4×97.2 or 388.8 sq ft of equivalent heating surface.

PIPE SIZES

The determination of pipe sizes for 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.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. Unusual conditions in the building to be heated.

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 pressure of the system; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems

CHAPTER 32—PIPING FOR STEAM HEATING SYSTEMS

TABLE 1. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR ONE-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	B	C	D	E	F
1	14.1	0.68	45	10,961	11.3
1¼	17.6	0.66	98	23,765	24.5
1½	20.0	0.66	152	36,860	38.0
2	23.0	0.57	288	69,840	72.0
2½	26.0	0.54	464	112,520	116.0
3	29.0	0.48	799	193,600	199.8
3½	31.0	0.44	1144	277,000	286.0
4	32.0	0.39	1520	368,000	380.0

INSTRUCTIONS FOR USING TABLE 1

1. Capacities given in Table 1 should never be exceeded on one-pipe risers.
2. Capacities are based on ¼-lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 4 and 5).

specially designed for varying initial pressures, such as the sub-atmospheric which normally operate under controlled partial vacua, the orifice, and the vapor systems which at times operate under such partial vacua as may be obtained due to the condition of the fire; (4) there is sufficient difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, and the dry return, when considered in relation to 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 pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. Laboratory experiments limit this to the capacities given in Tables 1 and 2 for vertical risers and in Table 3 for horizontal pipes at varying grades.

Maximum Velocity and Reaming

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

TABLE 2. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR TWO-PIPE LOW-PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT.	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A.	B.	C.	D.	E.	F.
¾	20.		40	9550	10.0
1	23	1.78	74	17,900	18.45
1¼	27	1.57	151	36,500	37.65
1½	30	1.48	228	55,200	57.0
2	35	1.33	438	106,100	109.5
2½	38	1.16	678	164,100	169.4
3	41	0.95	1129	273,500	282.2
3½	42	0.81	1548	375,500	387.0
4	43	0.71	2042	495,000	510.5

INSTRUCTIONS FOR USING TABLE 2

1. The capacities given in this table should never be exceeded on two-pipe risers.
2. Capacities are based on $\frac{1}{4}$ -lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 4 and 5.)

condensate is flowing, and the pressure drop in the pipe. Where the quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it runs horizontally, and (3) the quantity of condensate flowing against the steam.

Two factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided and which caused an actual difference of 20 per cent in the capacity of a 1-in. pipe in experiments carried on at the A.S.H.V.E. Research Laboratory (Table 4). The second is the reaming of the ends of the pipe after cutting, which, experiments indicate, might reduce the capacity of a 1-in. pipe as much as 28.7 per cent (Table 5). All of the capacity tables given in this chapter include a factor of safety. However, the pipe on which Table 4 is based showed no particular defects or constrictions on the inside, and the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

TABLE 3. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES^a

Pitch of Pipe in Inches per 10 Ft

Pitch of Pipe	¼ in.		½ in.		1 in.		1½ in.		2 in.		3 in.		4 in.		5 in.	
	Sq Ft. Based on 240 Btu	Max. Vel.	Sq Ft. Based on 240 Btu	Max. Vel.	Sq Ft. Based on 240 Btu	Max. Vel.	Sq Ft. Based on 240 Btu	Max. Vel.	Sq Ft. Based on 240 Btu	Max. Vel.	Sq Ft. Based on 240 Btu	Max. Vel.	Sq Ft. Based on 240 Btu	Max. Vel.	Sq Ft. Based on 240 Btu	Max. Vel.
¼	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
⅜	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
½	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	30
⅝	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
¾	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30

^aData from A.S.H.V.E. Research Laboratory.

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 6 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 is 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 4. PER CENT DIFFERENCE IN CAPACITY FOR CARRYING STEAM AND CONDENSATE DUE TO VARIATION OF PIPE SIZE AND SMOOTHNESS^a

	MAXIMUM CONDENSATION, LB PER HOUR			
Size of pipe.....	$\frac{3}{4}$ In.	1 In.	$1\frac{1}{4}$ In.	$1\frac{1}{2}$ In.
Minimum.....	14.00	24.89	45.42	70.50
Maximum.....	15.20	30.08	52.08	82.00
Per cent variation.....	8.6	20.8	14.7	16.3

^aData from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 5. EFFECT OF REAMING ENTRANCE TO ONE-INCH ONE-PIPE RISERS^a

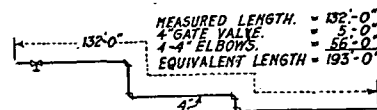
	MAXIMUM CAPACITY OF RISER	PER CENT DECREASE
Reamed entrances.....	24.7 lb per hour	0.0
Rounded entrances.....	23.9 lb per hour	3.2
Squared entrances.....	22.2 lb per hour	10.1
Three wheel cutter.....	19.2 lb per hour	22.2
Single wheel cutter.....	17.6 lb per hour	28.7

^aData from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 6. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

SIZE OF PIPE INCHES	ST'D. ELBOW	SIDE OUTLET TEE	GATE VALVE	GLOBE VALVE	ANGLE VALVE
	Length in Feet to be Added to Run				
2	5	16	2	18	9
2½	7	20	3	25	12
3	10	26	3	33	16
3½	12	31	4	39	19
4	14	35	5	45	22
5	18	44	7	57	28
6	22	50	9	70	32
7	26	55	10	82	37
8	31	63	12	94	42
9	35	69	13	105	47
10	39	76	15	118	52
12	47	90	18	140	63
14	53	105	20	160	72

Example of length in feet of pipe to be added to actual length of run.



TABLES FOR PIPE SIZING¹

Factors determining the size of a steam pipe and its allowable limit of capacity are as follows:

1. Pipe condensate flowing with steam.
2. Pipe condensate flowing against steam.
3. Pipe and radiator condensate flowing with steam.
4. Pipe and radiator condensate flowing against steam.

It is apparent that (3) and (4) are practically limited to one-pipe systems while (1) and (2) cover all other systems.

Tables 7 and 8, worked out for determining pipe sizes, have their columns lettered continuously, Columns A through L being in Table 7, and M through EE in Table 8. In the following text, reference made to columns will be by letter. The tables 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 radiation² (abbreviated EDR) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed without unusual or noticeable defects.

¹Pipe 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.

²As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of THE GUIDE to gradually eliminate the empirical expression square foot of equivalent direct radiation, EDR, and to substitute a logical unit based on the Btu. The new terms to express the equivalent of 1000 Btu (Mb), and 1000 Btu per hour (Mbh), have been approved by the A.S.H.V.E.

TABLE 7. STEAM PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Direct Radiation

(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.

CAPACITIES OF STEAM MAINS AND RISERS										SPECIAL CAPACITIES FOR ONE-PIPE SYSTEMS ONLY		
PIPE SIZE IN.	DIRECTION OF CONDENSATION FLOW IN PIPE LINE									Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
	With the Steam in One-Pipe and Two-Pipe Systems						Against the Steam Two-Pipe Only					
	1/32 lb or 1/4 Oz Drop	1/24 lb or 3/8 Oz Drop	1/16 lb or 1 Oz Drop	1/8 lb or 2 Oz Drop	1/4 lb or 4 Oz Drop	1/2 lb or 8 Oz Drop	Vertical	Horizontal				
	A	B	C	D	E	F						
3/4	-----	-----	-----	30	-----	-----	-----	30	-----	25	-----	-----
1	39	46	56	79	111	157	56	26	45	20	20	
1 1/4	87	100	122	173	245	346	122	58	98	55	55	
1 1/2	134	155	190	269	380	538	190	95	152	81	81	
2	273	315	386	546	771	1,091	386	195	288	165	165	
2 1/2	449	518	635	898	1,270	1,797	635	395	464	-----	260	
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	-----	475	
3 1/2	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	-----	745	
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	-----	1,110	
5	3,214	3,712	4,546	6,429	9,092	12,858	-----	3,150	-----	-----	2,180	
6	5,276	6,094	7,462	10,553	14,924	21,105	-----	-----	-----	-----	-----	
8	10,983	12,682	15,533	21,967	31,066	43,934	-----	-----	-----	-----	-----	
10	20,043	23,144	28,345	40,085	56,689	80,171	-----	-----	-----	-----	-----	
12	32,168	37,145	45,492	64,336	90,985	128,672	-----	-----	-----	-----	-----	
16	60,506	69,671	84,849	121,012	169,698	242,024	-----	-----	-----	-----	-----	
All Horizontal Mains and Down-Feed Risers							Up-Feed Risers	Mains and Un-dripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped	

Note.—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed.

aDo not use Column H for drops of 1/24 or 1/32 lb; substitute Column C or Column B as required.

bDo not use Column J for drop of 1/32 lb except on sizes 3 in. and over; below 3 in. substitute Column B.

cOn radiator runouts over 8 ft long increase one pipe size over that shown in Table 7.

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Table 7 may be used for sizing piping for steam heating systems by 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.

Sizing of return piping may be done with the aid of Table 8 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 7. It is customary to use the same pressure drop on both the steam and return sides of a system.

(Reference to this table will be by column letter *M* through *EE*)

Reference to this table will be by column letter *M* through *EE*.)

Reference to this table will be by column letter *M* through *EE*.)

CAPACITY OF RETURN MAINS AND RISERS

Pipe SIZE INCHES	Main																			
	1/32 Lb or 1/4 Oz Drop per 100 Ft				1/16 Lb or 1/2 Oz Drop per 100 Ft				3/16 Lb or 3/8 Oz Drop per 100 Ft				1/4 Lb or 1 Oz Drop per 100 Ft							
	Wet	Dry	Vac.	P	Wet	Dry	Vac.	S	T	U	V	W	X	Y	Z	AA	BB	CC	DD	EE
M	N	O	P	Q	R	S														
3/4	500	248		580	285	326				320	400	1,000	412	568	1,400	460	800			1,130
1	850	520		990	595	576	1,200	700		670	700	1,700	868	994	1,400	962	1,400			1,977
1 1/4	1,350	822		1,570	943	1,547	1,900	1,058		1,900	1,900	2,700	1,362	2,696	3,800	1,512	2,400			3,390
1 1/2	2,800	1,880		3,240	2,140	3,256	4,000	2,300	4,000	5,600	5,600	8,000	2,960	5,680	8,000	3,300	3,800			5,370
2	4,700	3,040		5,300	3,470	5,453	6,700	3,800	6,700	9,400	9,400	13,400	4,900	9,510	13,400	5,450	13,400			11,300
2 1/4	7,500	5,840		8,500	6,250	8,710	10,000	7,000	10,000	15,000	15,000	21,400	9,000	22,710	32,000	14,300	32,000			18,925
3	11,000	7,880		13,200	8,800	13,020	16,000	10,000	16,000	22,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000			30,230
3 1/2	15,500	11,700		18,300	13,400	17,910	22,000	15,000	22,000	31,000	31,000	44,000	21,500	44,000	77,400	21,500	77,400			45,200
4						31,500			38,700		62,000			88,000			124,000			62,180
5						50,450														109,300
6																				175,100

Riser									
Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet
190	570	700	190	994	1,400	190	994	1,400	1,977
450	976	1,200	450	1,704	2,400	450	1,704	2,400	3,390
990	1,547	1,900	990	2,696	3,800	990	2,696	3,800	5,370
1,500	3,256	4,000	1,500	5,680	8,000	1,500	5,680	8,000	11,300
3,000	5,453	6,700	3,000	9,510	13,400	3,000	9,510	13,400	18,925
	8,710	10,700		15,910	21,400		15,910	21,400	30,230
	13,020	16,000		22,710	32,000		22,710	32,000	45,200
	17,910	22,000		31,220	44,000		31,220	44,000	62,180
	31,500	38,700		54,920	77,400		54,920	77,400	109,300
	50,450	62,000		88,000	124,000		88,000	124,000	175,100

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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 lb or less. With a pressure drop of 1 lb and a length of run of 1,000 ft, the drop per 100 ft would be $\frac{1}{2}$ lb, while if the total drop were $\frac{1}{2}$ lb, the drop per 100 ft would be $\frac{1}{4}$ lb. In the first instance the pipe could be sized according to Column D for $\frac{1}{16}$ lb per 100 ft, and in the second case, the pipe could be sized according to Column C for $\frac{1}{32}$ lb. 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.

ONE-PIPE GRAVITY AIR-VENT SYSTEMS

1. For the steam main and dripped runouts to risers where the steam and condensate flow in the same direction, use $\frac{1}{4}$ -lb 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.

With a $\frac{1}{32}$ -lb drop the sizing would be the same as for $\frac{1}{4}$ lb 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.

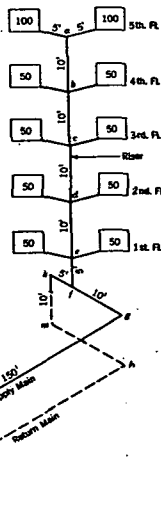
Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of $\frac{1}{4}$ lb the drop per 100 ft will be slightly less than $\frac{1}{16}$ lb. It would be well in this case to use $\frac{1}{24}$ lb, and this would result in the theoretical sizes indicated in Table 9. These theo-

TABLE 9. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 1

PART OF SYSTEM	SECTION OF PIPE	RADIATION SUPPLIED (Sq Ft)	THEORETICAL PIPE SIZE (INCHES)	PRACTICAL PIPE SIZE (INCHES)
Branches to radiators..	-----	100	2	2
Branches to radiators..	-----	50	1 1/4	1 1/4
Riser.....	a to b	200	2	2
Riser.....	b to c	300	2 1/2	2 1/2
Riser.....	c to d	400	2 1/2	2 1/2
Riser.....	d to e	500	3	3
Riser.....	e to f	600	3	3
Branch to riser.....	f to g	600	3 1/2	3 1/2
Supply main.....	g to h	600	3	3
Branch to supply main	h to j	600	2 1/2	3
Dry return main.....	f to k	600	1 1/4	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. 1. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

From Boiler or Source of Supply
To Boiler or Source of Supply



retical 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.

Notes on Gravity One-Pipe Air-Vent Systems

1. Pitch of mains should be not less than 1/4 in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should not be less than 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 be not less than half its diameter at its largest part.
4. Supply mains, branches to risers, or risers, should be dripped where necessary.

TWO-PIPE GRAVITY AIR-VENT SYSTEMS

The method employed in determining pipe sizes for two-pipe gravity air-vent systems is similar to that described for one-pipe systems except that the steam mains never carry radiator condensation. The drop allowable per 100 ft of equivalent run is obtained by taking the equivalent length to the farthest radiator as double the actual distance, and then dividing the allowable or desired total drop by the number of hundreds of feet in the equivalent length. Thus in a system measuring 400 ft from the boiler to the farthest radiator, the approximate equivalent length of run would be 800 ft. With a total drop of 1/2 lb the drop per 100 ft would be 1/8 or 1/16 lb; therefore, Column D would be used for all steam mains where the condensation and steam flow in the same direction. If a total drop of 1/4 lb is desired, the drop per 100 ft would be 1/32 lb

and Column B would be used. If the total drop were to be 1 lb, the drop per 100 ft would be 1/8 lb and Column E would be used.

For mains and riser runouts that are not dripped, and for radiator runouts where in all three cases the condensation and steam flow in opposite directions, Column I should be used, while for the steam risers Column H should be used unless the drop per 100 ft is 1/4 lb or 1/32 lb, when Columns B or C should be substituted so as not to exceed the drop permitted.

On an overhead down-feed system the main steam riser should be sized by reference to Column H, but the down-feed steam risers supplying the radiators should be sized by the appropriate Columns B through G, since the condensation flows downward with the steam through them. The riser runouts, if pitched down toward the riser as they should be, are sized the same as the steam mains, and the radiator runouts are made the same as in an up-feed system.

In either up-feed or down-feed systems the returns are sized in the same manner and on the same pressure drop basis as the steam main; the return mains are taken from Columns O, R, U, X, or AA according to the drop used for the steam main; and the risers are sized by reading the lower part of Table 8 under the column used for the mains. The horizontal runouts from the riser to the radiator are not usually increased on the return lines although there is nothing incorrect in this practice. The same notes apply that are given for one-pipe gravity systems.

TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, (3) because with large variation 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, 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 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.

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 correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed 1/8 lb to 1/4 lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over 1/8 lb divided by 4, or 1/32 lb. 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 $\frac{1}{32}$ lb. 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 $\frac{1}{32}$ lb. 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 be not less than $\frac{1}{4}$ in. in 10 ft.
2. Pitch of horizontal runouts to risers and radiators should be not 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 supply main smaller than 2 in., and when the supply main is 3 in. or over at the boiler or pressure reducing valve it should be not less than $2\frac{1}{2}$ in. at the far end.
4. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return. The drip for a vapor system may be connected into the dry return through a thermostatic drip trap.

VACUUM SYSTEMS

Vacuum systems are usually employed in large installations and have total drops varying from $\frac{1}{4}$ to $\frac{1}{2}$ lb. 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 go to 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 $\frac{1}{2}$ lb divided by 12, or $\frac{1}{24}$ lb. In this case the steam main would be sized from Column *C*, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of $\frac{1}{24}$ lb). 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*; 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 $\frac{1}{8}$ -lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.
2. Pitch of mains should be not less than $\frac{1}{4}$ in. in 10 ft.
3. Pitch of horizontal runouts to risers and radiators should be not 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.
4. In general it is not considered desirable to have a supply main smaller than 2 in. When the supply main is 3 in. or over, at the boiler or pressure reducing valve, it should be not less than $2\frac{1}{2}$ in. at the far end.
5. When necessary, the supply main, supply riser, or branch 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 Chapter 31 under *Up-Feed Vacuum Systems*.

ATMOSPHERIC SYSTEMS

The sizing of the supply and return piping on atmospheric systems is practically identical with the sizing used for vacuum systems and the same notes apply, except that no lift can be made in the return line.

SUB-ATMOSPHERIC SYSTEMS

Any properly pitched, correctly sized vacuum system without a lift may be used as a sub-atmospheric system when the proper equipment is substituted for the ordinary vacuum pump, traps, and controls. On new systems manufacturers usually recommend a drop on the steam line of between $\frac{1}{4}$ and $\frac{3}{8}$ lb for the total run, and suggest adding 25 ft to the total equivalent length of run to insure that the steam gets through to the last radiator.

The same notes apply to these systems as for vacuum systems, except that no lifts can be made in the returns.

ORIFICE SYSTEMS

The orifice systems can be operated with any piping system suitable for vacuum operation, according to experienced designers. Because these systems vary considerably in detail, it is advisable to consult the manufacturer of the particular system contemplated for recommendations.

The same notes apply to these systems as to vacuum systems, except that lifts cannot be made in the returns of orifice systems if a vacuum pump is used.

HIGH PRESSURE STEAM

When steam heating systems are supplied with steam from a high pressure plant, one or more pressure-reducing valves are used to bring the pressure down to that required by the heating system. It has been considered good practice to make the pressure reductions in steps not to exceed 50 lb in each case. For example, in reducing from 100-lb gage to 2-lb gage, two pressure reducing valves would be used, the first reducing the pressure from 100-lb gage to 50 lb and the second reducing the pressure from 50-lb gage to 2-lb gage. Valves are available that will reduce 100 lb in one step, and it is questionable whether two valves are now required for initial pressures of 150 lb or less.

The pressure-reducing valve, or pressure-regulator as it is sometimes termed, has ratings which vary 200 to 400 per cent. Some of these ratings are based on arbitrary steam velocities through the valve of 5,000 to 10,000 fpm and it is assumed that the valve when wide open has the same area as the pipe on the inlet opening of the valve. At times it is considered desirable to keep the steam velocity in the high pressure section of the piping and the low pressure section constant. The velocity through the valve port is obviously a function of the pressure drop across the valve. It is well known that steam flowing through an orifice increases its velocity until the pressure on the outlet side is reduced to 58 per cent of the absolute pressure on the inlet side, and that with further reduction of pressure on the outlet side little change in velocity will be obtained. As practically all pressure-reducing valves used for steam heating work

lower the steam pressure to less than 58 per cent of the inlet pressures, only the maximum velocity through such valves need be considered. If it is assumed that the valve, when fully open, has an area equal to that of the inlet pipe size; that the steam is flowing into a pressure less than 58 per cent of the initial pressure, that the orifice efficiency is approximately 70 per cent, and that 20 per cent more is allowed for a factor of safety, then the pressure reducing valves will have the working capacities shown in Table 10. If the valve, when fully open, does not give an orifice area equal to that of the pipe on the inlet side, then the capacities will be proportional to the percentage of opening secured, taking the pipe area as 100 per cent. More frequently difficulty is encountered from the use of pressure reducing valves which are too large in size instead of being

TABLE 10. CAPACITIES OF PRESSURE-REDUCING VALVES
(100-LB GAGE DOWN TO ANY PRESSURE—52 LB OR LESS)

INLET NOMINAL PIPE DIAMETER (INCHES)	POUNDS STEAM PER HOUR AT 100-LB GAGE	EQUIVALENT DIRECT RADIATION Sq Ft AT 1/4 LB	EQUIVALENT DIRECT RADIATION Sq Ft AT 1/2 LB
1/2	866	3,464	2,598
3/4	1,576	6,304	4,728
1	2,459	9,836	7,377
1 1/4	4,263	17,052	12,689
1 1/2	5,808	23,232	17,424
2	9,564	38,256	28,692
2 1/2	13,623	54,492	40,869
3	21,041	84,104	63,123
3 1/2	28,213	112,852	84,039
4	36,285	145,140	108,855
5	56,971	227,884	170,913
6	82,336	329,344	247,008

Formula:
$$\frac{A \times V \times 3600 \times .50}{144 \times 3.88} = \text{pounds per hour passed by orifice.}$$

where

- A = area of inlet pipe, square inches.
V = velocity of steam through orifice (approximately 870 fps).
.50 = 70 per cent efficiency of orifice less 20 per cent for factor of safety.
144 = square inches in 1 sq ft.
3600 = seconds in one hour.
3.88 = cubic feet per pound at 100-lb gage.

too small. Where valves are large in size, the valve tends to work close to the seat, causing it to cut out in a relatively short time, as well as being noisy in operation.

Most exact regulation of pressure on steam heating systems is secured from diaphragm-operated valves controlled by a pilot line from the low pressure pipe, taken off the low pressure main at least 15 ft from the reducing valve. The reducing valves operating on the proportional-reduction principle will give a variation of steam pressure on the low pressure side if the initial pressure varies between considerable limits. The so-called dead-end valve is used for reduced pressures where the line has not sufficient condensing capacity at all times to condense the leakage that might occur with the ordinary valve. Single-disc valves do not give as close regulation as double-disc valves, but the single disc is preferable where dead-end valves are necessary, such as on short runs to thermo-

statically controlled hot water heaters, central fan heating units and unit heaters.

The correct installation (Fig. 2) of a pressure-reducing valve includes a pressure-reducing valve with a gate valve on each side, a by-pass controlled by a globe valve, a pressure gage on the low pressure side, and a safety valve on the low pressure main at some point, usually within a reasonable distance of the pressure-reducing valve. Pressure-reducing valves should have expanded outlets for sizes greater than 2 in. Where the steam main is of still larger diameter than the expanded outlet, and in cases where straight valves are used, an increaser is placed close against the outlet of the valve to reduce the velocity immediately after passing through the valve. Strainers are recommended on the inlets of all pressure-reducing valves. A pressure gage may be located on the high-pressure line near the valve if desired.

Owing to the large variation in steam demand on the average heating system, it is generally advisable to use two pressure-reducing valves con-

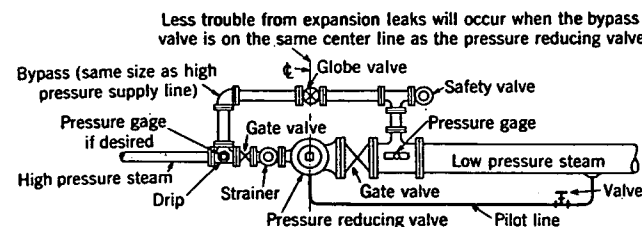


FIG. 2. TYPICAL PRESSURE-REDUCING VALVE INSTALLATION

nected in parallel. One valve should be large enough for the maximum load and the other should have a diameter approximately half that of the first. The smaller valve can be used most of the time, for it will give much better regulation than the larger one on light or normal loads.

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.

EXPANSION IN STEAM AND RETURN LINES

Because all steam and return lines expand and contract with changes in temperature, provision should be made for such movement. The expansion in steam supply pipes is normally taken at 1 1/4 to 1 1/2 in. per 100 ft and in return lines at one-half or two-thirds of this amount. It may be calculated accurately if the temperature rise and fall can be determined with reasonable certainty (Page 616, Chapter 34). The temperature at the time of erection often has a greater expansion effect on piping than the temperature in the building after it has been put into service.

Expansion may be taken care of by any, or all, of three different methods, namely, (1) the spring in the pipe including offsets and expansion bends, (2) the turning of the pipe on its threads and swing joints, and (3) the use of expansion joints.

By the first scheme, which is the most popular method where space permits, the pipe is offset, or *broken*, around rooms or corners, and is hung so that the spring in the pipe at right angles to the expansion movement is sufficient to absorb the expansion. If conditions do not lend themselves to this treatment, regular expansion bends of the *U* or offset type may be used. In tight places such as pipe tunnels the expansion joint is preferable. See additional material on pipe expansion bends in Chapter 34.

On riser runouts and radiator runouts the swing joint is used almost without exception. On high vertical risers the pipes may be reversed every five to ten stories; that is, the supply is carried over to the adjacent return riser location and the return riser is run over to the former supply riser location, thus making horizontal offsets in each line. Corrugated copper expansion joints also are used on risers but must be made accessible in case future replacement becomes necessary.

PIPING CONNECTIONS AND DETAILS

Piping connections may be classified into two groups: *first*, those suitable for any system of steam heating; *second*, those devised for certain systems which cannot be satisfactorily applied to any other type. There are also various details that apply to piping on the steam side which cannot be used on the returns. An installation that is designed and sized correctly and installed with care may be rendered defective by the use of improper connections, such as runouts that do not allow for expansion, thermostatic traps unprotected from scale, pressure-reducing valves without strainers, and lack of drips at required points.

BOILER CONNECTIONS

Supply

Boiler headers and connections have the largest sizes of pipe used in a system. Cast-iron, horizontal-type, low pressure heating boilers usually have several tapped outlets in the top, the manufacturers recommending their use in order to reduce the velocity of the steam in the vertical uptakes from the boiler and to permit entrained water to return to the boiler instead of being carried over into the steam main where it must be cared for by dripping. Steel heating boilers usually are equipped with only one steam outlet but many engineers believe that better results are obtained by specifying that such boilers have two. The second outlet, usually located 3 or 4 ft back of the regular one, reduces the velocity 50 per cent in the steam uptake.

Fig. 3 shows a type of boiler connection that was used for many years and one with which some boilers are now piped. The uptakes are carried as high as possible, turned horizontally and run out to the side of the boiler and then are connected together into the main boiler runout which drops into the top of the boiler header through a boiler stop valve. No drips are provided on this type of runout except a very small one which is sometimes installed on the boiler side of the stop valve. Fig. 4 shows a

type of boiler connection which is regarded as superior to that shown in Fig. 3 and which is the type illustrated in the system diagrams in Chapter 31. This type is similar to that shown in Fig. 3 except that the horizontal branches from the uptakes are connected into the main boiler runout, and the steam is carried toward the rear of the boiler. The branch to the building or boiler header is taken off *behind* the last horizontal boiler connection. At the rear end of this main runout, a large size drip, or balance pipe, is dropped down into the boiler return, or into the top of the Hartford Loop, which is described in a following paragraph. As a result, any water carried over from the boiler follows the direction of steam flow

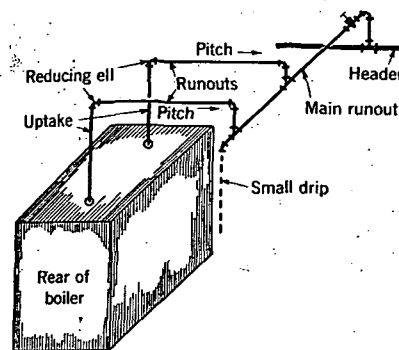


FIG. 3. OLD STYLE STANDARD BOILER CONNECTIONS

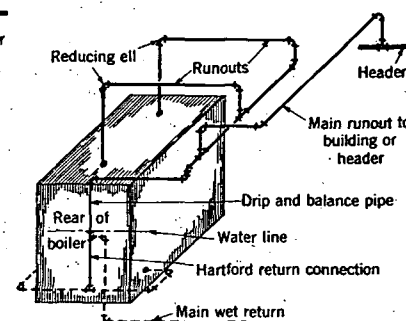


FIG. 4. APPROVED METHOD OF BOILER CONNECTIONS

toward the rear and is discharged into the rear drip, or balance pipe, without being carried over into the system.

Return

Cast-iron boilers are generally provided with return tapplings on both sides; but steel boilers often are equipped with only one return tapping. A boiler with side return tapplings will usually have a more effective circulation if both tapplings are used. Check valves generally should not be used on the return connection to steam heating boilers from one and two pipe gravity systems because they are not always dependable inasmuch as a small piece of scale or dirt lodged on the seat will hold the tongue open and make the check useless. These valves also offer a certain amount of resistance to the returns coming back to the boiler, and in gravity systems will raise the water line in the far end of the wet return several inches. However, if check valves are omitted and the steam pressure is raised with the boiler steam valve closed, the water in the boiler will be blown out into the return system with the accompanying danger of boiler damage. These objections are largely overcome with the Hartford return connection.

*See method of calculating height above water line for gravity one-pipe systems in Chapter 31.

Hartford Return Connection

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford Connection, or the Underwriters Loop, is recommended.

Fig. 5 shows this connection for a two-boiler installation. For a single boiler installation the connection is made as is indicated for one boiler. The essential features of construction of a Hartford Loop 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 connection about 2 in. below the normal boiler water line from the return main to the boiler steam and return balance connection.

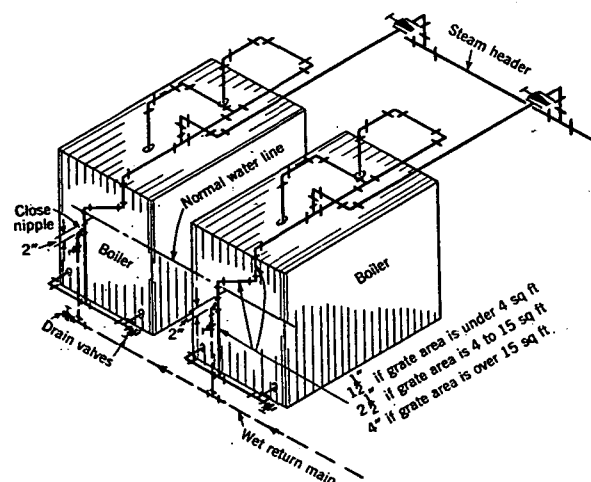


FIG. 5. THE HARTFORD RETURN CONNECTION

Sizing Boiler Connections

Little authentic information is available on the sizing of boiler runouts and steam headers. Although many 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

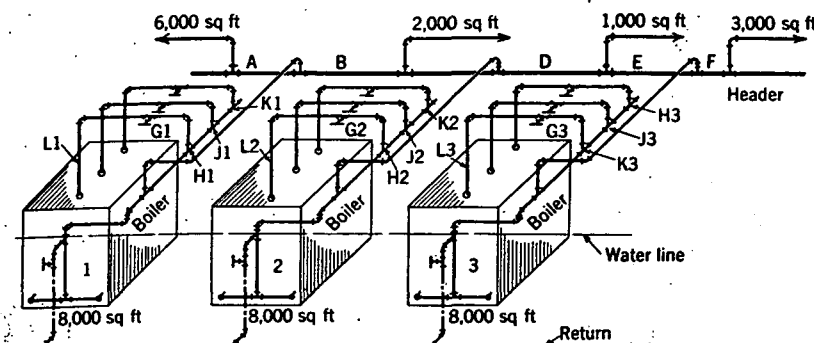


FIG. 6. BOILER STEAM HEADER AND CONNECTIONS

boiler and the horizontal main or runout is compensated for by the use of reducing ells (Figs. 3 and 4).

The following example illustrates the sizing of the boiler connections shown in Fig. 6.

Example 4. Determine the size of boiler steam header and connections (Fig. 6) if there are three boilers, two to carry 50 per cent of the load each, and the third to be used as a spare. The steam mains are based on $\frac{1}{8}$ -lb drop per 100 sq ft of equivalent direct radiation (EDR).

Solution:

Size of Boiler Header

WHEN OPERATING ON BOILERS	LOAD ON VARIOUS PORTIONS OF HEADER						MAXIMUM LOAD
	A	B	C	D	E	F	
Nos. 1 and 2	6000	0	2000	4000	3000	3000	6000
Nos. 2 and 3	6000	6000	8000	2000	3000	3000	8000
Nos. 3 and 1	6000	0	2000	2000	3000	3000	6000
Max. Load	6000	6000	<u>8000</u>	4000	3000	3000	8000

8000 sq ft @ $\frac{1}{8}$ lb per 100 ft = 6 in. main. (See Table 7.)

Size of Boiler Runouts

The three runouts

$$G_1, G_2, G_3 = \frac{8000}{3} = 2667 \text{ sq ft each @ } \frac{1}{8} \text{ lb per 100 ft} = 4 \text{ in. pipe.}$$

$$H_1, H_2, H_3 = 2667 \text{ sq ft each @ } \frac{1}{8} \text{ lb per 100 ft} = 4 \text{ in. pipe}^4 \text{ (See Table 7).}$$

$$J_1, J_2, J_3 = 5333 \text{ sq ft each @ } \frac{1}{8} \text{ lb per 100 ft} = 5 \text{ in. pipe}^4 \text{ (See Table 7).}$$

$K_1, K_2, K_3 = 8000 \text{ sq ft each @ } \frac{1}{8} \text{ lb per } 100 \text{ ft} = 6 \text{ in. pipe}^4 \text{ (See Table 7).}$

The uptakes from the boiler probably would be 6 in. pipe with a 6 in. X 4 in. reducing ell at top.

*Note.—As K_1 , K_2 , K_3 all carry 8000 sq ft and are 6 in. pipe, the whole runout including J_1 , J_2 and J_3 and H_1 , H_2 and H_3 and the leads from the boiler headers to the main steam header would also be made 6 in. pipe.

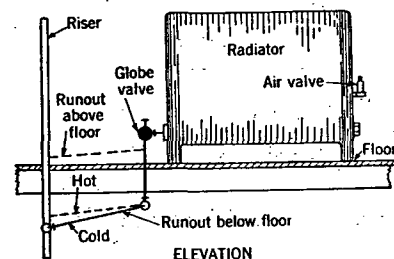
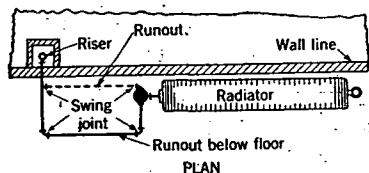


FIG. 7. TYPICAL ONE-PIPE RADIATOR CONNECTIONS (UP-FEED OR DOWN-FEED)

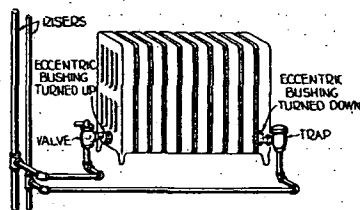


FIG. 8. CONNECTIONS TO STEAM-TYPE RADIATOR FOR TWO-PIPE GRAVITY SYSTEM, UP-FEED OR DOWN-FEED

Note.—Steam-type radiators should not be used on any except gravity one-pipe and gravity two-pipe systems.

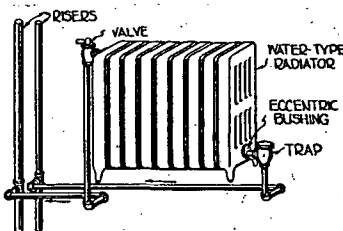


FIG. 9. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS FROM UP OR DOWN-FEED RISERS

Note.—Suitable for up-feed or down-feed atmospheric, vapor, vacuum, sub-atmospheric and orifice systems.

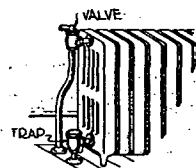


FIG. 10. TOP AND BOTTOM RADIATOR CONNECTIONS FROM UP-OR DOWN-FEED RISERS. (NOT TO EXCEED 8 TO 10 SECTIONS.)

Note.—Suitable for up-feed or down-feed atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems. Opposite end connections always preferable.

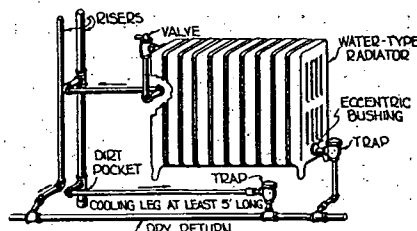


FIG. 11. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS WITH HEEL OF DOWN-FEED RISER DRIPPED INTO DRY RETURN

Note.—Suitable for down-feed only. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

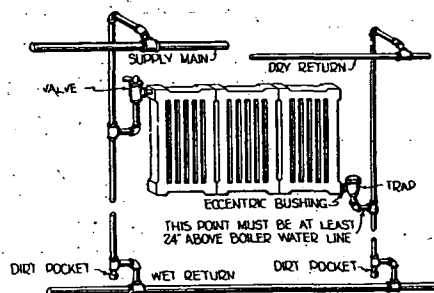


FIG. 12. CONNECTIONS TO RADIATOR HUNG ON WALL

Note.—For up-feed with radiators below level of steam main. For atmospheric and vapor systems. Not suitable for vacuum, sub-atmospheric, or orifice systems.

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 size being obtained from the Chart for Pressure Drop for Various Rates of Flow of Water, Fig. 3, Chapter 35. The relative boiler loads should be considered, as in the case of gravity return connections.

Radiator Connections

Radiator connections are important on account of the number of repetitions which occur in every heating installation. They must be properly pitched and they must be arranged to allow not only for movement in the riser but, in frame buildings, for the shrinkage of the building. In a three story building this sometimes amounts to 1 in. or more. The simplest connection is that for the one-pipe system where only one radiator connection is necessary. Where the radiator runouts are located on the ceiling or under the floor, sufficient space usually is available to make a good swing joint with plenty of pitch, but where the runouts must come above the floor the vertical space is small and the runouts can project out into the room only a short distance. Fig. 7 illustrates two satisfactory methods of making runouts on a one-pipe gravity air vent system of either the up-feed or down-feed type, the runout below the floor being indicated in full lines and the runout above the floor in dotted lines. Sometimes it is necessary to set a radiator on pedestals; or to use high legs, in order to obtain sufficient vertical distance to accommodate above-the-floor runouts. Particular attention must be given to the riser expansion as it will raise the runout and thereby reduce the pitch.

Similar connections for a two-pipe system of the gravity air vent type are illustrated in Fig. 8 for the old steam type radiator. If the water type is used, the supply tapping is at the top instead of at the bottom, the runouts otherwise remaining as shown in Fig. 8. A satisfactory type of radiator connection for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of both the up-feed and down-feed types is shown in Fig. 9.

While short radiators, not exceeding 8 to 10 sections, may be supplied and returned from the same end as indicated in Fig. 10, the top-and-bottom-opposite-end method is to be preferred in all cases where it can be used. On down-feed systems of the atmospheric, vapor, vacuum, sub-atmospheric, and orifice types, the bottom of the supply riser must be

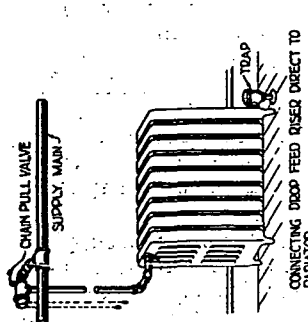


FIG. 13. CONNECTING DROP FEED RISER DIRECT TO RADIATOR BY TURNING VALVE ON ITS SIDE

Note.—Suitable for up-feed with radiators below level of steam main. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

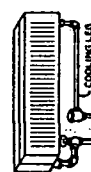


FIG. 16. HORIZONTAL SECTIONAL FIN-TYPE HEATING UNIT

Note.—Suitable for up-feed or down-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

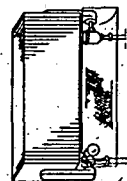


FIG. 16. FIN-TYPE HEATING UNIT VALVES BEHIND GRILLE

Note.—Suitable for up-feed or down-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

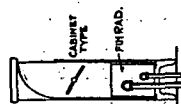


FIG. 17. FIN-TYPE HEATING UNIT CONCEALED IN CABINET

Note.—Suitable for any two-pipe system.

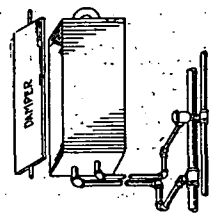


FIG. 14. CONVECTOR CONNECTIONS SAME END

Note.—Suitable for up-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

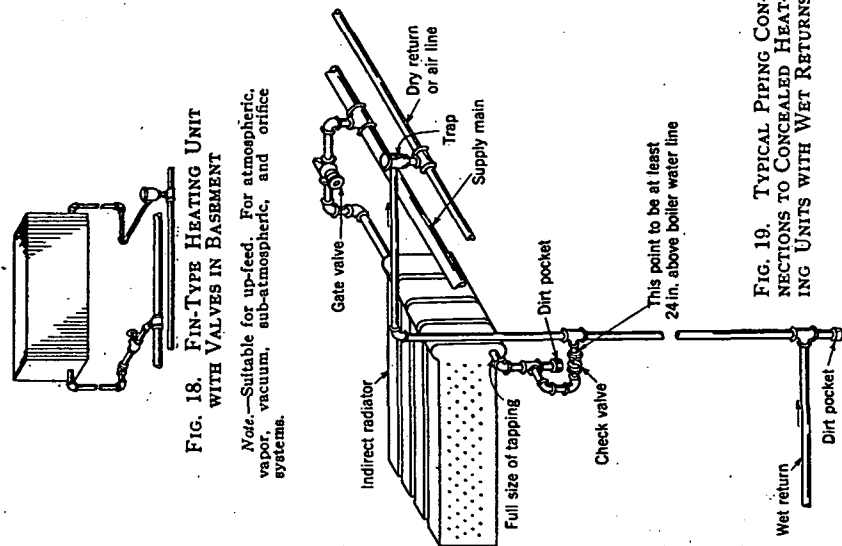


FIG. 18. FIN-TYPE HEATING UNIT WITH VALVES IN BASEMENT

Note.—Suitable for up-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

FIG. 19. TYPICAL PIPING CONNECTIONS TO CONCEALED HEATING UNITS WITH WET RETURNS

dripped into the return somewhat as illustrated in Fig. 11. On up-feed systems of the vapor and atmospheric types, where radiators in the basement are located below the level of the steam main, the drop to the radiator is dripped into the wet return and an air line is used to vent the return radiator connection into an overhead return line, as illustrated in Fig. 12. When the radiator stands on the floor below the main, the drip on the steam branch down to the radiator may be omitted if an overhead valve, as shown in Fig. 13, is used. This method is also suitable for vacuum, sub-atmospheric, and orifice systems.

Convactor Connections

Convectors often are installed without control valves, a damper being used to shut off the flow of air to retard the heat transfer from the convector even though it is still supplied with steam. The piping connections for a convector with the inlet and outlet at the same end are shown in Fig. 14. There is no valve on the steam side but there is a thermostatic trap on the return. The damper for control is shown immediately above the convector. This piping is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of the up-feed type. A similar unit with connections on opposite ends and suitable for the same systems is shown in Fig. 15. This unit has no damper but requires a valve on the steam connection for control. When valves must be located so as to be accessible from the supply air grille, the arrangement usually takes the form indicated in Fig. 16. Convectors with damper control, installed in cabinets or under window sills, usually are connected as shown in Fig. 17. A convector located in the basement and supplying air to a room on the floor above may be piped as pictured in Fig. 18 for all systems except gravity one-pipe or two-pipe systems.

Vapor systems with heating units in the basement where the returns are wet would be treated as in Fig. 19. Similar heating units where a dry return is available would be connected as shown in Fig. 20. If the dry return were on a vacuum, atmospheric, sub-atmospheric or orifice system, the treatment would be identical.

Pipe Coil Connections

Pipe coils, unless coupled in a correct manner, often give trouble from short circuiting and poor circulation. The method of connecting shown in Fig. 21 is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

Indirect Air Heater Connections

Heating units for central fan systems have simple connections on the steam side. The steam main is carried into the fan room and has a single branch tapped off for each row of heating units. Each of these main branches is split into as many connections as need be made to each row, governed by the number of stacks and the width of the stacks. Each stack must have at least one steam connection, and wide stacks are more evenly heated with two steam connections, one at each end.

The piping shown in Fig. 22 is for small stacks and has the steam connected at only one end. On the return side all of the returns are collected

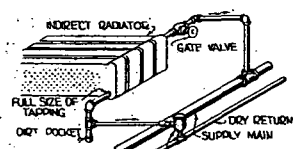


FIG. 20. TYPICAL PIPING CONNECTIONS TO INDIRECT RADIATORS WITH DRY RETURN

Note.—Suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

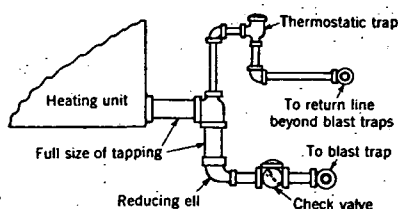


FIG. 23. HEATING UNIT RETURN CONNECTION WITH SEPARATE AIR LINE

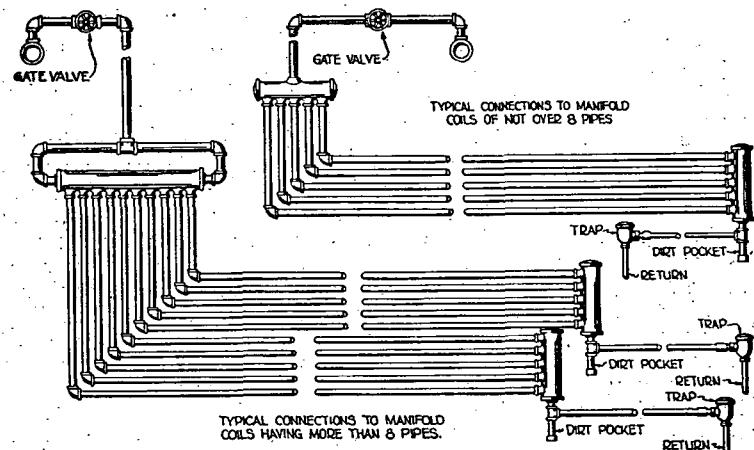


FIG. 21. TYPICAL PIPE COIL CONNECTIONS

Note.—Suitable for up-feed or down-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

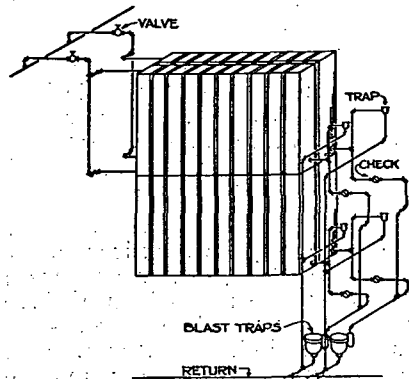


FIG. 22. CONNECTIONS FOR HEATING UNITS OF CENTRAL FAN SYSTEMS

Note.—Suitable for atmospheric and vacuum systems.

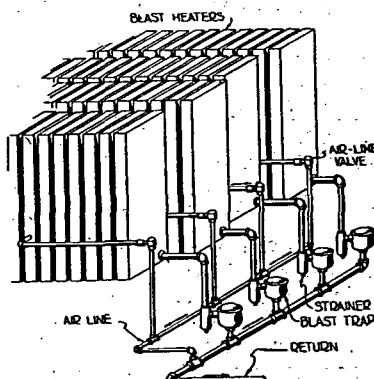


FIG. 24. TYPICAL CONNECTIONS TO CENTRAL FAN SYSTEM HEATING UNITS EXCEEDING 12 SECTIONS

Note.—Suitable for vacuum and atmospheric systems.

together through check valves and are passed through blast traps which are connected to the vacuum return or to an atmospheric return. The air from the stacks, in the case illustrated, passes up into a small air line and through a thermostatic trap into a line connecting into the return beyond the blast trap. It is important to use a nipple the full size of the outlet tapping on the stack and to reduce the pipe size to the normal return size required, by the use of a reducing ell, as indicated in Fig. 23.

Where the stacks contain some thirteen or more sections, an auxiliary air tapping is made to the lower portion of one of the middle sections, in the manner illustrated in Fig. 24, to prevent air collecting at this point. Thermostatic control as applied to such heating units in modern practice

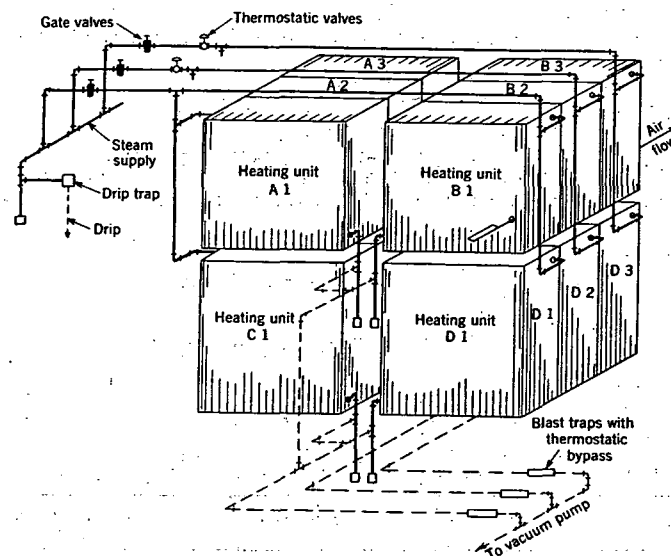


FIG. 25. TYPICAL PIPING FOR ATMOSPHERIC AND VACUUM SYSTEMS WITH THERMOSTATIC CONTROL (CENTRAL FAN SYSTEM)

consists of a thermostatic valve located in each main branch from the steam line so that each valve will open or close a complete row of stacks across the entire face of the heating unit. The stack closest to the outside air intake usually is not equipped with a thermostatic valve. A gate valve on the steam pipe to the first coil is operated manually to supply steam continuously in freezing weather. Good practice demands that the returns be connected in parallel with the steam supplies, with a separate steam trap for each bank of coils having a separately valved steam supply. This arrangement is illustrated in Figs. 22 and 25, for blast traps having external thermostatic bypasses and integral thermostatic bypasses, respectively.

A method of connecting a unit heater to a one pipe air-vent steam heating system is illustrated in Fig. 26.

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.

$$EDR = \frac{Q \times 60 \times (t_1 - t_e)}{55.2 \times 240} = \frac{Q \times (t_1 - t_e)}{220.8} \quad (3)$$

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, degrees Fahrenheit.

t_1 = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.

60 = the number of minutes in one hour.

55.2 = the number of cubic feet of air heated 1 F by 1 Btu.

240 = the number of Btu in 1 sq ft of EDR.

Example 5. Assume that the heating units shown in Fig. 25 are 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

^aOne quarter of total row load.

^bOne half of stack load if two steam connections are made; otherwise, same as stack load.

DRIPPING

Any 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 steam flow. Any steam main in any heating system can be elevated if dripped (Fig. 27). Steam mains also may be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for the condensation (Fig. 28). Return mains may be carried past doorways or other obstructions by using the scheme illustrated in Fig. 29; in vacuum systems it is well to have a gate valve in the air line.

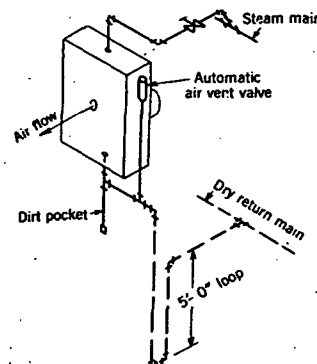


FIG. 26. UNIT HEATER CONNECTED TO ONE PIPE AIR VENT SYSTEM WITH DRY RETURN

Note.—If connection can be made to a wet return, no 5 ft loop is required.

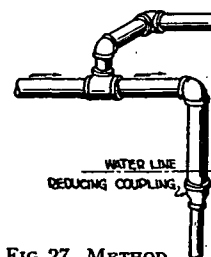


FIG. 27. METHOD OF DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

Note.—Suitable for vapor and atmospheric systems.

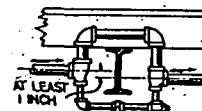


FIG. 28. LOOPING MAIN AROUND BEAM

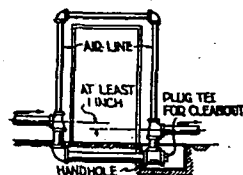


FIG. 29. LOOPING DRY RETURN MAIN AROUND OPENING

Note.—Suitable for any dry return line and any return line carrying air.

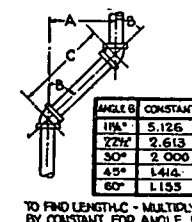


FIG. 30. CONSTANTS FOR DETERMINING PROPER LENGTH OF OFFSET PIPE

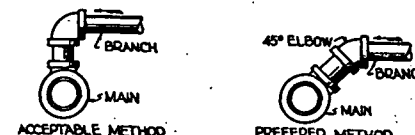


FIG. 31. ACCEPTABLE AND PREFERRED METHODS OF TAKING BRANCH FROM MAIN

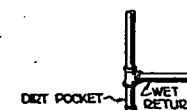


FIG. 32. DIRT POCKET CONNECTION

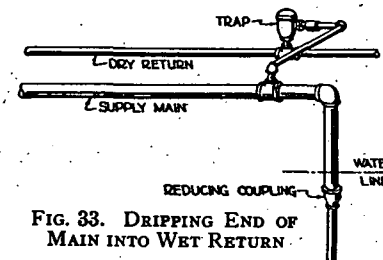


FIG. 33. DRIPPING END OF MAIN INTO WET RETURN

Note.—Suitable for vapor systems.

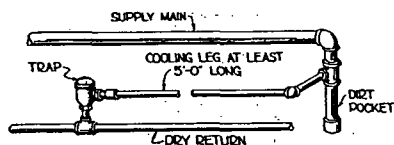


FIG. 34. DRIPPING END OF MAIN INTO DRY RETURN. (A GATE VALVE IS RECOMMENDED AT THE INLET SIDE OF THE TRAP)

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.

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. 30.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 31, 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 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.

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. 32.

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. 33. 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. 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. 34) 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 bottom of the steam risers are dripped in the manner shown in Fig. 35.

PROBLEMS IN PRACTICE

1 • What factors determine the size of steam piping and the allowable limit of capacity?

Factors which determine the size of steam piping are the desired initial pressure and the allowable drop in pressure which is permissible to maintain a pressure in the farthest

radiator. The length of run in sizing piping is important and it is generally considered as the distance along the piping from the source of steam supply to the farthest radiator, with allowances for resistance of elbows and valves expressed in terms of equivalent length.

2 • When the size of pipe is still undetermined, what arbitrary percentage is usually added to the actual length to obtain the equivalent length?

Usually 100 per cent; in other words, the actual length is doubled to allow for the added drop produced by the valves, tees, elbows, and other fittings.

3 • What are the major factors to be considered in determining the flow of steam in pipes?

- The initial steam pressure available and the total pressure drop allowable between the source of steam supply and the end of the return system. The pressure drop should never exceed one half of the initial pressure.
- The maximum steam velocity allowable. When condensate is flowing against the steam, the velocity must not be so great as to produce water hammer, or hold up water in parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which disturbances take place depends upon:
 - Size of pipe.
 - Whether pipe is vertical or horizontal.
 - Pitch or grade of pipe.
 - Quantity of water flowing against steam.
- The equivalent length of run from the source of steam supply to the farthest heating unit, with allowance for friction in pipe fittings and valves.

4 • Name three fundamental considerations in designing the piping system for steam heating.

- Provision for the distribution of suitable quantities of steam to the various heating units.
- Provision for the return of condensate from the radiators and piping to the boiler.
- Provision of means for expelling air from the radiators and piping.

5 • Why is the proper reaming of the ends of pipe necessary?

The capacities of pipes depend upon the free area available for flow. In cutting the pipe this area may be restricted by a burr, which may decrease the capacity of a pipe more than 25 per cent in the smaller pipe sizes.

6 • a. What are the major factors to be considered when selecting a pressure-reducing valve?

b. How should such valve be installed?

- The initial pressure of the steam must be considered along with the desired reduced pressure. The connected load to be supplied must be known in square feet of equivalent direct radiation or in pounds of steam per hour. For operation with a continuous load, a semi-balanced or double seated valve operated by a diaphragm gives good results. Where the load is intermittent, as in process work or with thermostatically controlled blast heaters, a so-called dead end or single seated valve should be used.
- The pressure reducing valve should be installed in a horizontal line with a gate valve on each side, and with a by-pass operated by a valve. The pressure balancing pipe from the diaphragm chamber should be connected into the top or side of the low pressure main not less than 15 ft from the reducing valve.

7 • What is the usual expansion allowance and how it is compensated for in heating system supply risers?

The expansion of low pressure steam piping is normally taken as $1\frac{1}{4}$ to $1\frac{1}{2}$ in. per 100 ft of pipe. With a five story building a double swing connection between the riser and the main will suffice. In buildings between 5 and 10 stories high the riser should be anchored near its center and have double swing connections to the main. For taller buildings expansion loops or riser offsets are used which are capable of handling a length of riser reaching 5 stories in either direction from the joint. The risers are anchored at each alternate 5 stories. All radiators must have double swing connections, and those connected above where the riser is anchored must be given greater pitch to insure their having proper grade when the riser is heated.

8 ● Why should all boiler steam supply tappings be used full size?

In order to operate at low steam velocities so the water in suspension can separate from the steam and remain in the boiler.

9 ● What is the Underwriters Loop or the Hartford Connection?

An arrangement of piping on the returns to low pressure boilers wherein the return line is raised up nearly to the water line of the boiler and is then dropped back and connected to the boiler return inlet; the high point is connected by a balance pipe to the steam runout from the boiler on the boiler side of all stop valves. With this loop no check valve is required, and water cannot be backed out of the boiler and into the return at a point lower than the invert of the pipe at the top of the loop.

10 ● What are the important factors in making radiator connections?

Connections to radiators should be made as direct as possible, of proper size, with ample pitch of piping and allowance for expansion.

11 ● Why should careful attention be given to proper dripping and drainage of steam piping?

The steam mains and risers must be quickly drained of condensate and where necessary vented of air in order to obtain a sufficient supply of steam to the radiators. Proper drainage is also necessary to insure a noiseless heating system.

12 ● What is the limit of pressure drop usually recommended in a vacuum system?

Not over $\frac{1}{8}$ lb (2 oz) per 100 ft of equivalent run, and not over 1 lb total drop.

13 ● When steam and condensation are flowing in the same direction, what is the maximum total pressure drop which should be used?

The maximum total pressure drop should not exceed one half of the initial steam pressure.

14 ● What does a proper installation of a pressure reducing valve include?

A strainer in front of the pressure reducing valve; a gate valve in front of the strainer; a gate valve after the reducing valve; a by-pass around the two gate valves, strainer, and pressure reducing valve; and a globe valve in the by-pass. Sometimes a safety valve on the low pressure side and pressure gages on both sides are installed. The high pressure line should be dripped just before the high pressure steam enters the pressure reducing valve assembly.

15 ● Will a pressure reducing valve which is reducing the steam pressure from 100 lb gage to 50 lb gage pass more or less steam than the same valve when reducing the steam pressure from 100 lb gage to 5 lb gage?

The valve will pass practically the same volume of steam in each case as the velocity of steam flowing through an orifice shows no material increase after the reduced absolute pressure has fallen to 58 per cent of the initial absolute pressure. Because of its greater density, the weight of steam passed will be greater in the case of the reduction to 50 lb gage.

Chapter 33

HOT WATER HEATING SYSTEMS AND PIPING

One- and Two-Pipe Systems, Selecting Pipe Sizes, Forced Circulation, Effect of Variations in Pipe Sizes, Gravity Circulation, Mechanical Circulation, Expansion Tanks, Installation Details

A HOT water heating system is one in which water is the medium by which heat is carried through pipes from the boiler to the heating units. There are two general types, namely, *forced circulation* and *gravity circulation* systems. In the former the pressure head maintaining flow is produced mechanically, whereas in the latter the pressure head is produced by the differences in weight of the water in the flow and in the return risers.

The fundamental rule in the design of a hot water system is that the total friction and resistance head in any circuit must equal the pressure head causing the water to flow in the same circuit.

In designing a hot water heating system, it is necessary to determine:

1. The heat losses of the rooms or spaces to be heated. (See Chapter 7.)
2. The size and type of boiler. (See Chapter 25.)
3. The location, type, and size of heating units. (See Chapter 30.)
4. The method of piping.
5. Suitable pipe sizes.
6. The type and size of circulating pump (if forced circulation).
7. The type and size of expansion tank.

The unit, a square foot of equivalent direct radiation, EDR, has been used for many years for rating purposes in both steam and hot water systems, but its use, especially in hot water systems, has always resulted in complications and confusion. It is the plan of THE GUIDE to eventually eliminate this empirical expression and to substitute a logical unit based on the Btu. The Mb, the equivalent of 1000 Btu, and the Mbh, the equivalent of 1000 Btu per hour, which have been approved by the A.S.H.V.E., are used in this chapter on hot water systems to replace the square foot of radiation formerly used.

ONE- AND TWO-PIPE SYSTEMS

Pipe systems may be divided into two general types, namely, *two-pipe* and *one-pipe* systems. In a two-pipe system the piping is arranged so that the water flows through only one radiator during a circuit through the system, so that all radiators are supplied with water at practically the same temperature as that in the boiler. In a one-pipe system, the water flows through more than one radiator during its circuit. In that case, the

first radiator receives the hottest water; the second radiator, somewhat cooler water; the third one, still cooler; and so on. As the temperature of the water supplied to a radiator is lowered, the size of the radiator must be increased and, consequently, the total heating surface for a one-pipe system must be greater than that for a two-pipe system for the same service.

Two-pipe systems may be divided into two classes, *direct return* systems (Fig. 1), and *reversed return* systems (Fig. 2). In a direct return system the water returns to the heater by a direct route after it has passed through its radiator and, as a result, the paths through the three radiators shown in Fig. 1 are of unequal lengths, the path through the first radiator being the shortest and that through the third radiator, the longest. In a reversed return system, the water returns to the heater by an indirect route after it has passed through the radiators, so that the paths leading through the three radiators shown in Fig. 2 are practically of equal length.

The reversed return system has an advantage over the direct return system in that it is more likely to function satisfactorily even though the

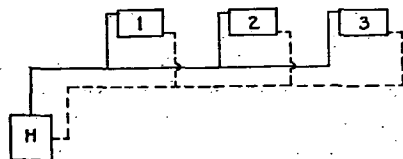


FIG. 1. A DIRECT RETURN SYSTEM

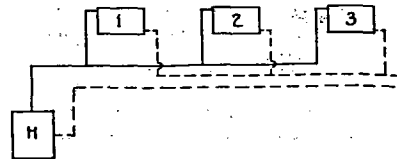


FIG. 2. A REVERSED RETURN SYSTEM

pipe system is not accurately designed. For example, if in Fig. 2 all pipes are of one size, each of the three radiators will receive approximately the same quantity of hot water because the three paths are practically of equal length, whereas in Fig. 1, if all pipes are of the same size, Radiator 1 will receive more water than the others because the path through it is shorter than those through the other radiators. As a result, Radiator 1 will be filled with water at a higher average temperature than the remaining two radiators, and will therefore dissipate more heat. To prevent this unequal distribution of heat it is necessary to throttle the paths through Radiators 1 and 2 so that the friction heads of the three paths are equal when each radiator receives its proper quantity of water.

A comparison of Fig. 1 and Fig. 2 may suggest that a reversed return system requires considerably longer mains than a direct return system. This is not always the case. For example, note the reversed return system of Fig. 3.

PIPE SIZES

The pressure heads available in forced circulation systems are much larger than those 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 pipes of a heating system are reduced in size, the necessary increase in

the velocity of the water increases the cost of operating the circulating pump. There is an optimum velocity of the water in a heating system for which the sum of the cost of the system and the cost of its operation is a minimum. This velocity should be determined by calculation for the particular system under consideration.

Since the velocities in forced circulation systems are higher than those in gravity circulation systems, and since the friction heads in a heating system vary almost as the squares of the velocities, 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

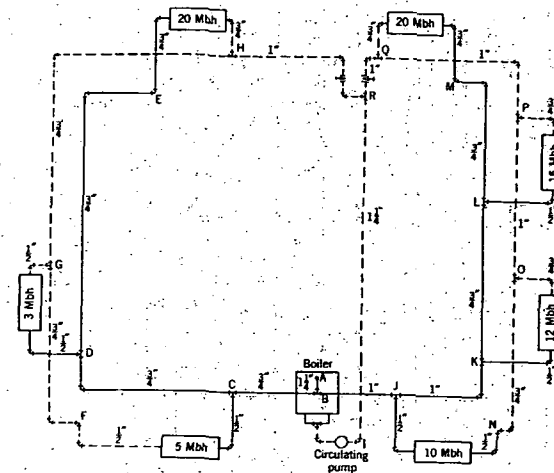


FIG. 3. A FORCED CIRCULATION REVERSED RETURN SYSTEM*

*Note that the numbers on the radiators indicate thousands of Btu per hour (Mbt/h) and not square feet.

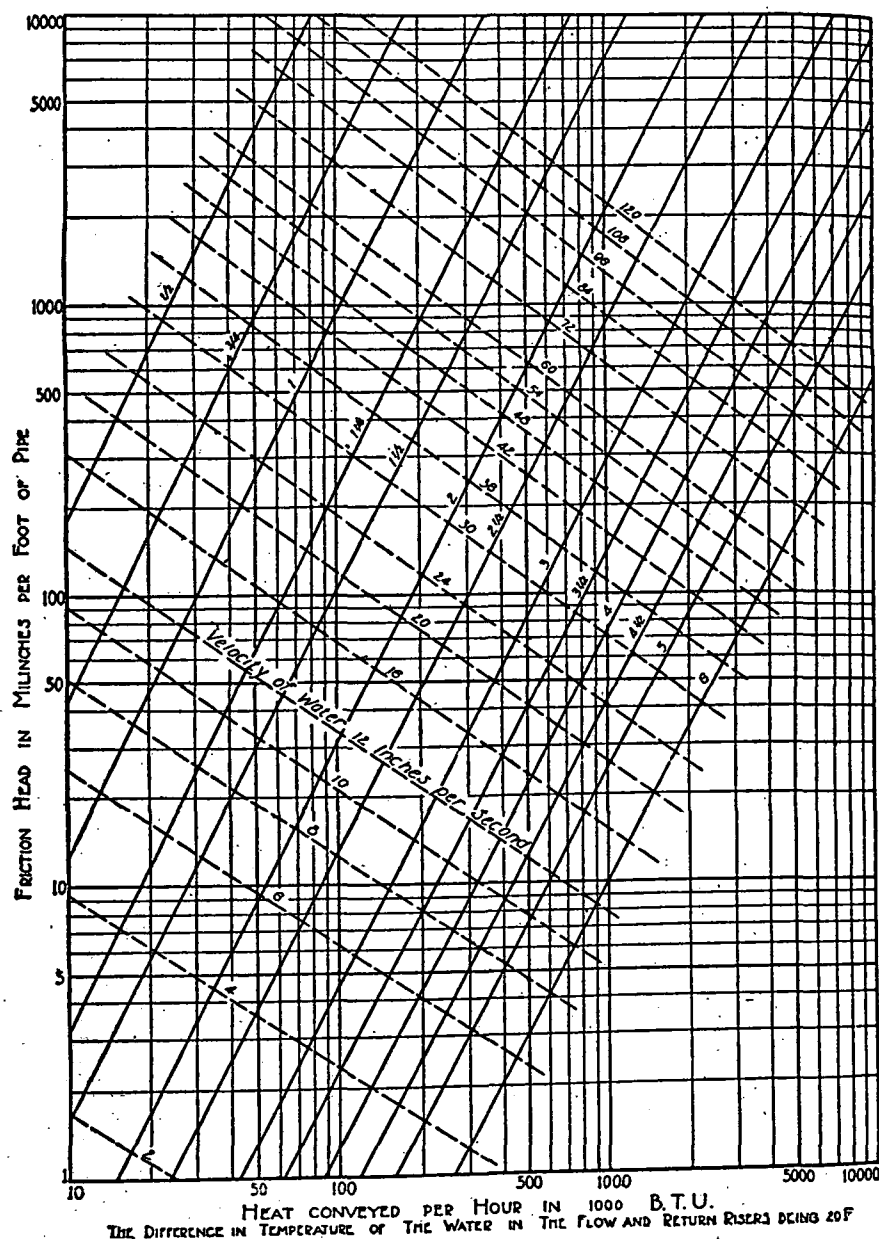
is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

FORCED CIRCULATION

The following examples will illustrate the procedure to be followed in designing forced circulation systems:

Example 1. Assume that the longest path through 7 radiators shown in Fig. 3 consist of 200 ft of mains, 25 ft of radiator connections, 1 boiler, 1 radiator, 1 radiator valve, 1 stop cock, 12 elbows, and 2 tees. Also assume that the short branch main contains the same number of fittings and that the main is 150 ft long with 7 ft of radiator connections. Design the piping for this system.

Solution. The friction heads of boiler, radiator valve and tee may be expressed in terms of the friction head in 1 elbow according to the values given in Table 1. Having done this, the longest circuit consists of 225 ft of pipe and 34 elbow equivalents. The friction head of 1 elbow is approximately equivalent to that in a pipe having a length equal to 25 diameters. Assume that the average pipe size in this case will be 1 in.



branch *BC* and 57 in *BJ*. It will be noted that 28 Mbh is slightly over the capacity of a $\frac{3}{4}$ in. pipe, therefore use a $\frac{3}{4}$ in. pipe in *BC* and a 1 in. in Section *HR*. Section *CD* should have a capacity for 23 Mbh, therefore use a $\frac{3}{4}$ in. pipe in *CD* and *GH*. A capacity of 20 Mbh is not sufficiently under the capacity of a $\frac{3}{4}$ in. pipe to warrant the use of a $\frac{1}{2}$ in. pipe, so use $\frac{3}{4}$ in. in Sections *DE* and *FG*. The radiator branches are sized accordingly, with $\frac{1}{2}$ in. up to 11,000 Btu (11 Mbh) and $\frac{3}{4}$ in. up to 25,000 Btu (25 Mbh).

Due to the difference in the equivalent lengths of each circuit, a static head of 6 in. on the pump would produce a greater velocity in the shortest branch than in the longer branch and consequently a higher friction loss per ft. This variation is usually a negligible factor in most installations and can usually be overlooked. Should the variation in friction head be insufficient to allow the use of smaller pipes, this factor should be taken into consideration. For purposes of illustration in this example, assume that Section *AB* is 4 ft, and then $4 \times 240 = 960$ milinches. $71,800 - 960 = 70,840$ milinches. $70,840$ divided by 231 (total equivalent length of short branch) = 306 milinches per ft which is 66 milinches per ft more than is available in the longer circuit. Therefore approximately 12 per cent more capacity is available in the pipes which will change the pipe size only a slight amount. If it is necessary to correct this variation, generally a stop cock may be placed in the return line of the short branch and adjusted after complete installation.

However, if the variation be of sufficient magnitude, the pipes in the shorter branch should be sized accordingly. In this case the pipe size should be selected according to a frictional loss of 240 milinches per ft. Due to the fact that Section *BJ* requires a capacity of 57 Mbh, which is slightly over 1 in., use a 1 in. pipe in *BJ* and *QR*. Section *JK* has a

TABLE 2. CAPACITIES OF PIPES IN Mbh (1000 BTU PER HOUR) AND Velocities of Water in Pipes in Inches per Second for FORCED CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 2 FT AND FOR A MAXIMUM TEMPERATURE DROP OF 10 F^a

1	2	3	4	5	6	7	8	9
PIPE SIZE (INCHES)	EQUIVALENT LENGTH OF PIPE (FEET) ^b	EQUIVALENT TOTAL LENGTH OF PIPE IN FEET IN LONGEST CIRCUIT						
		100	150	200	250	300	350	400
		UNIT FRICTION HEAD, IN MILINCHES						
		240	160	120	96	80	69	60
$\frac{1}{2}$	1	6.2 15	4.8 12	4.1 10	3.4 9	2.9 8	2.6 7.5	2.4 7
$\frac{3}{4}$	2	13.2 18	10.3 14	8.6 12	7.3 11	6.2 10	6.0 9	5.5 8.5
1	2.3	25.0 22	19.2 17	16.3 15	14.4 13	12.5 12	12.0 11	11.1 10.5
1 $\frac{1}{4}$	3.0	52.8 27	40.8 21	34.8 18	31.2 16	27.8 15	26.4 14	24.0 13
1 $\frac{1}{2}$	3.5	79.2 30	60.7 23	51.2 20	45.6 18	40.8 16	40.0 15	36.0 14
2	4.0	153.8 36	120.0 28	104.0 24	93.5 22	86.4 20	81.5 18	73.8 17
2 $\frac{1}{2}$	6.0	250.0 41	192.0 32	164.5 28	149.0 25	139.2 22	135.8 21	122.5 19
3	6.5	444.0 48	348.0 37	294.0 32	270.0 29	254.0 26	240.0 24	223.0 22

^aFor other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

^bApproximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

load of 47 Mbh which is approximately 1 in. Therefore use 1 in. in *JK* and *PQ*. *KL* carries 35 Mbh which is approximately half-way between $\frac{3}{4}$ and 1 in. Use $\frac{3}{4}$ in. in *KL* and 1 in. in *OP*. Section *LM* requires a capacity of 20 Mbh, therefore use $\frac{3}{4}$ in. in *LM* and *NO*. Size radiator branches as previously described.

Many times a $\frac{1}{2}$ in. pipe will prove to be too large, but at the present time there seems to be a slight aversion in practice against the use of pipes of smaller sizes, especially for hot water installations.

If a number of heating systems are to be designed for similar conditions, i.e., for a total friction head of 6 ft and a temperature drop through the radiators of 20 F when the maximum quantity of heat is being delivered to the building, a table such as Table 3 may be prepared from the data of Fig. 4. Having this table, the pipe sizes for the system of Example 1 can be easily selected. For example, for Sections *AB* and *RJ*, each supplying 85 Mbh, the equivalent pipe length of the system is 299 ft. In the table the length shown nearest to this length is 300 ft. In the 300-ft column, a 1-in. pipe is too small and a 1 $\frac{1}{4}$ -in. pipe is too large. The 1 $\frac{1}{4}$ -in. pipe will therefore be selected. For other systems, it will be economical to operate with different friction heads, and tables may be prepared similar to Tables 2 and 4, which are based on total friction heads of 2 and 18 ft, respectively.

TABLE 3. CAPACITIES OF PIPES IN Mbh (1000 BTU PER HOUR) AND Velocities of Water in Pipes in Inches per Second for FORCED CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 6 FT AND FOR A MAXIMUM TEMPERATURE DROP OF 10 F^a

1	2	3	4	5	6	7	8
PIPE SIZE (INCHES)	EQUIVALENT LENGTH OF PIPE (FEET) ^b	EQUIVALENT TOTAL LENGTH OF PIPE IN FEET IN LONGEST CIRCUIT					
		200	300	400	600	800	1000
		UNIT FRICTION HEAD, IN MILINCHES					
		360	240	180	120	90	72
$\frac{1}{2}$	1	7.4 18	6.0 15	5.0 13	3.8 10	3.4 9	3.1 7.5
$\frac{3}{4}$	2	15.8 22	12.7 18	10.8 16	8.4 12	7.7 11	6.7 9
1	2.5	30.0 27	24.0 22	20.4 19	15.8 15	13.9 13	12.5 11
1 $\frac{1}{4}$	3.3	64.8 33	52.5 26	44.4 23	33.6 18	30.0 16	26.8 14
1 $\frac{1}{2}$	4.0	96.0 37	76.8 31	64.8 26	50.1 20	44.7 18	40.8 15
2	5.0	192.0 44	153.0 36	130.0 30	100.1 24	90.0 21	78.0 18
2 $\frac{1}{2}$	6.0	300.0 50	244.0 41	206.0 35	161.0 26	144.0 24	130.0 21
3	7.5	550.0 58	436.0 48	368.0 42	287.0 32	249.0 27	228.0 24

^aFor other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

^bApproximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

Example 2. Design a direct return two-pipe forced circulation system for the layout shown in Fig. 5 assuming a temperature drop of 10 F through the radiation. For this system the length of the pipe line from the boiler to the highest radiator on the farthest riser and back to the boiler is about 250 ft. There are about 16 elbow equivalents having an equivalent pipe length of about 50 ft, so the total equivalent pipe length is about 300 ft.

Solution. The same pipe size tables may be used as those developed for the reversed return system of Fig. 3, Table 3 which provides for a friction head of 6 ft.

Referring to the column for an equivalent total length of 300 ft for Sections AB and KA, each supplying 117.6 Mbh, it will be found that a 1½-in. pipe is too small and a 2-in. pipe is too large. Consequently, a 1½-in. pipe is selected for the flow line AB, and a 2-in. pipe for the return line KA. For Sections BC and JK, each supplying 88 Mbh, a 1½-in. pipe is only slightly too small and it is selected. The remaining pipe sizes are selected in a similar manner and recorded in Fig. 5. For a temperature drop of 10 F, 24.5 gpm of water must be circulated. The pump to select is one which has its highest efficiency when it is delivering 24.5 gpm against a 6-ft head.

To secure a correct distribution of hot water among the several risers it is necessary, as previously stated, to introduce special resistances to balance the several risers, as follows:

The first riser is 80 ft nearer the boiler than the fifth riser. In order that the two may be balanced, i.e., that they may operate under equal pressure heads, resistance must be added to the first riser equal to the friction head in the 80 ft of flow main from B to F plus that in the 80 ft of return main from G to K.

TABLE 4. CAPACITIES OF PIPES IN Mbh (1000 BTU PER HOUR) AND Velocities of Water in Pipes in Inches per Second FOR FORCED CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 18 FT AND FOR A MAXIMUM TEMPERATURE DROP OF 10 F^a

PIPE SIZE (INCHES)	EQUIVALENT LENGTH OF PIPE (FEET) ^b	EQUIVALENT TOTAL LENGTH OF PIPE IN FEET IN LONGEST CIRCUIT				
		200	400	600	800	1000
		UNIT FRICTION HEAD, IN MILINCHES				
		1080	540	360	270	216
½	1.0	12.7 32	8.6 23	7.2 18	6.2 15	5.5 13
¾	2.0	27.5 40	18.7 28	15.1 22	13.7 19	11.5 17
1	2.5	55.0 48	36.8 34	30.0 27	26.4 23	22.6 20
1¼	3.0	122.0 59	81.5 42	66.0 33	58.3 28	50.5 25
1½	4.0	182.0 66	122.0 46	98.2 37	86.2 31	74.2 27
2	5.0	371.0 80	252.0 56	201.0 45	180.0 38	151.0 33
2½	7.0	598.0 91	407.0 65	323.0 51	287.0 43	240.0 38
3	9.0	1110.0 107	790.0 76	598.0 60	527.0 51	443.0 44

^aFor other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 F, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

^bApproximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

It will be noted from Table 3 that the unit friction head is about 240 milinches per foot. The total friction head in the flow and return mains between the first and fifth risers is therefore 160×240 or 38,400 milinches, or a little more than 3 ft, which must be supplied by additional resistance in the first riser to prevent its having an advantage over the fifth riser.

This resistance can be supplied by a calibrated and adjusted modulating valve or by an orifice resistor in a union. If the orifice resistor is to be used, its size may be selected from Table 5 as follows:

The lower part of the first flow riser supplies 28.8 Mbh. According to Table 3, it should be a 1-in. pipe and would have a velocity of 22 in. per second, if it were supplying 24 Mbh. Since it is supplying 28.8 Mbh, the velocity will be about 26 in. per second. From Table 5 it will be found that for a 1-in. pipe and a velocity of 24 in. per second, an 0.45-in. orifice will produce a loss of head of 37,000 milinches. For a velocity of 26 in. per second, the loss of head will be somewhat more, probably about 43,000 milinches; the difference between it and the required resistance will be about 10 per cent, which is permissible, and the 0.45-in. orifice is selected.

The sizes of the orifice resistors for the second, third, and fourth risers are selected in a similar manner and found to be 0.45 in., 0.50 in., and 0.55 in., respectively.

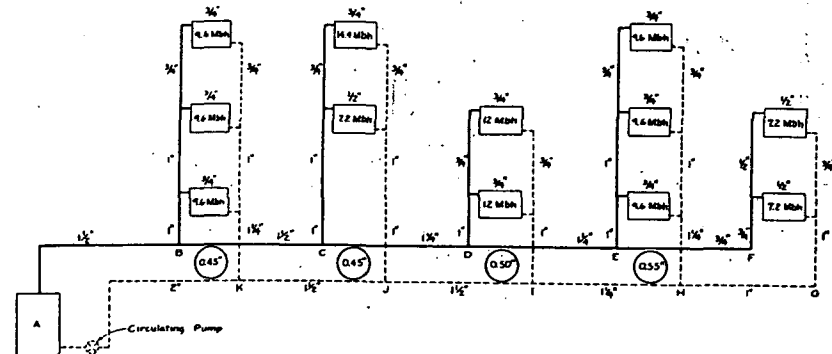


FIG. 5. A FORCED CIRCULATION DIRECT RETURN SYSTEM

If the design of the system of Fig. 5 is to be extremely refined, the gravity pressure heads produced by the risers should be taken into consideration. With water at 220 F and 210 F, respectively, in the risers, the gravity head is 50 milinches per foot of water column or 25 milinches per foot of flow and return pipe. The pump pressure head in this case is 240 milinches per foot of pipe, and the gravity head, being only one tenth as large as the pump head, may be neglected without serious error. This is generally done.

Temperatures of 220 F and 210 F would be used only during the coldest weather for which the system is designed. At other times the temperatures would be lower, the temperature drop smaller, and the gravity heads smaller. The pump pressure head remains constant throughout the season if the pump is operated at a constant speed and, consequently, the gravity head is generally less than one-tenth of the pump head.

Effect of Variations in Pipe Sizes

The pipe sizes for the several parts of the system selected from the tables are only approximately correct but the resulting error should be

TABLE 5. FRICTION HEADS (IN MILINCHES) OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN UNIONS

DIAMETER OF ORIFICES (INCHES)	VELOCITY OF WATER IN PIPE IN INCHES PER SECOND									
	2	3	4	6	8	10	12	18	24	36
$\frac{3}{4}$-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
$1\frac{1}{4}$-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
$1\frac{1}{2}$-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 $1\frac{1}{4}$ -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 $\frac{3}{4}$ -in., 1-in., and $1\frac{1}{4}$ -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).

negligible as may be seen from the following study. Assume, as an extreme case, that the error in pipe size is so large that the water flows twice as fast through one of the radiators as through the others. This would make the friction head through this radiator almost four times as large as those through the other radiators. The result would be that the water, in flowing through the radiator, would cool 5 F instead of 10 F. The mean water temperature in the radiator would then be $217\frac{1}{2}$ F instead of 215 F, and the mean temperature difference, water to air, would be $147\frac{1}{2}$ F instead of 145 F. The heat dissipated by the radiator would therefore be about 2 per cent more than calculated. It is evident that this difference in heat dissipation is smaller than the difference between

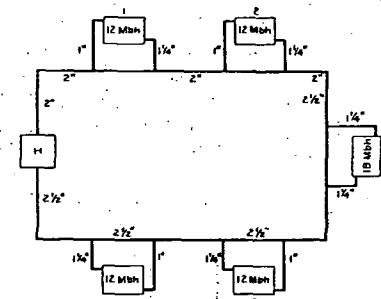


FIG. 6. A ONE-PIPE GRAVITY CIRCULATION SYSTEM

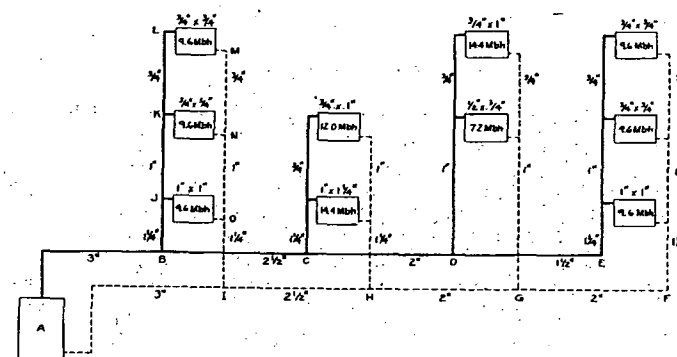


FIG. 7. A TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEM

the calculated heat losses and the actual heat losses, and also smaller than the average difference between the calculated radiator sizes and the nearest stock sizes selected.

GRAVITY CIRCULATION

For gravity circulation, the one-pipe system shown in Fig. 6 and the two-pipe direct return system shown in Fig. 7 are probably in most common use.

The one-pipe system has the disadvantage that the radiator nearest the

boiler is the only one which receives water at approximately the temperature at which it leaves the boiler. All other radiators receive cooler water and must be proportionally increased in size, so the total heating surface in the system is considerably larger than that in a corresponding two-pipe system.

The pipe sizes in gravity circulation systems may be varied. As the pipe sizes are decreased, the temperature drop through the radiators, which produces circulation, is increased and it becomes necessary to increase the temperature of the water leaving the boiler so that the mean temperature in the radiator remains constant. For example, Fig. 8 shows diagrammatically an elementary heating system which will function with either 1½-in. or 1-in. pipe. The radiator is required to deliver 27 Mbh, and the circuit consists of 30 ft of pipe and 20 elbow equivalents.

If 1½-in. pipe is used, the system will operate correctly if the water temperatures in the flow and return risers are 200 F and 180 F, respectively. The mean water temperature in the radiators will then be 190 F and, if the radiator is located in air having a temperature of 70 F, the size of the radiator must be sufficient to deliver 27 Mbh under these conditions.

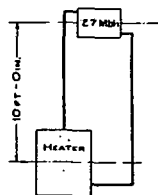


FIG. 8. AN ELEMENTARY SYSTEM

If 1-in. pipe is used, the system will function correctly with water temperatures in the flow and return risers of 210 F and 170 F, or of 200 F and 160 F. In the first case, the mean water temperature is again 190 F and the same size radiator may be used as with the 1½-in. pipe, but the temperature of the water leaving the boiler must be raised from 200 F to 210 F. In the second case, the temperature of the water leaving the boiler is the same as for the 1½-in. pipe, but the mean water temperature in the radiator is lowered from 190 F to 180 F, and theoretically the size of the radiator should be increased about 12½ per cent to deliver the required 27 Mbh (See Table 3, Chapter 6, 1933 GUIDE).

This indicates the extent to which pipe sizes and radiator sizes may be decreased by increasing the temperatures of the water in the boiler, as is possible in closed systems and in open systems in which the open expansion tank is located sufficiently high to secure a pressure in the boiler equal to that existing in the boiler of the closed system.

Example 3. Design a one-pipe gravity circulation system for the layout shown in Fig. 6. Assume that the main circuit consists of 150 ft of pipe, 7 elbows, and one boiler.

Solution. Replace the boiler by 3 elbow equivalents and assume that the size of the main will be about 2 in. According to Table 6, Column 2, a 2-in. elbow is equivalent to 4 ft of pipe, and the total equivalent length of the main will be about 150 plus 40, or 190 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main and that the temperature drop in the system is to be 35 F, Table 6 may be used to determine the size of the mains. Note from Column 8, for a 200-ft

CHAPTER 33—HOT WATER HEATING SYSTEMS AND PIPING

TABLE 6. CAPACITIES OF MAINS IN Mbh, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 0.6 IN., A TEMPERATURE DROP OF 35 F, WHEN THE MAINS ARE 4 FT ABOVE THE CENTER OF THE BOILER

1	2	3	4	5	6	7	8	9	10	11	
PIPE SIZE (INCHES)	EQUIVALENT LENGTH OF PIPE (FEET)	EQUIVALENT TOTAL LENGTH OF PIPE IN FEET IN LONGEST CIRCUIT									
		75	100	125	150	175	200	250	300	350	
		UNIT FRICTION HEAD, IN. MILINCHES									
		8.0	6.0	4.8	4.0	3.4	3.0	2.4	2.0	1.7	
1½	3.0	43.0	37.5	33.0	30.0	27.0	25.0	22.2	20.2	18.7	
2	4.0	83.0	72.0	63.0	57.0	51.0	48.0	42.0	38.0	35.0	
2½	4.5	140.0	115.0	100.0	90.0	81.5	75.4	67.2	61.0	56.0	
3	5.0	234.0	204.0	175.5	160.0	143.0	133.0	110.0	107.5	100.0	
3½	5.5	347.0	300.0	260.0	236.0	214.0	200.0	177.0	160.0	146.0	
4	6.0	490.0	422.0	370.0	334.0	297.0	278.0	248.0	223.0	205.0	

*Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

length, that a 2-in. main will supply 48 Mbh and a 2½-in. main, 75.4 Mbh. Since the system to be designed is to supply 66 Mbh, a 2-in. pipe is too small and a 2½-in. pipe too large. The solution is to use some 2-in. and some 2½-in. pipe. Since the 2½-in. is nearer the correct size than the 2-in., select 2-in. pipe for the first 50 or 60 ft out of the boiler and 2½-in. for the remaining pipe back to the boiler.

Tables 7 and 8 may be used to design the radiator risers and connections. According to Table 7, for 12 Mbh the flow riser should be ¾ in. and the return riser 1 in., and the riser branches should be 1 in. and 1¼ in., respectively. Note that according to Table 8, both radiator tappings should be 1 in. To simplify the construction, select 1-in. flow

TABLE 7. MAXIMUM CAPACITIES OF RISERS^a IN Mbh, AND Velocities of Water in Pipes in Inches Per Second for ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A DROP OF 35 F THROUGH EACH RADIATOR

PIPE SIZE (INCHES)		EQUIVALENT LENGTH OF PIPE (FEET)	1ST FLOOR ^b			2ND FLOOR	3RD AND 4TH FLOORS
Flow	Return		Mbh	Vel. (In. per Sec.) ^d		Mbh	Mbh
				Flow	Return		
1/2	1/2	1.0				5	6.2
1/2	3/4					6.4	8.0
3/4	3/4	1.5	9	2.3	2.3	10.1	14.0
3/4	1		12	3.2	2.0	12.8	17.1
1	1	2.0	18	2.5	2.5	20	26.0
1	1 1/4		21	3.0	2.0	25.2	34
1 1/4	1 1/4	3.0	26	3.0	3.0	43	55
1 1/4	1 1/2		34	4.0	2.5		
1 1/2	1 1/2	3.5	48	3.0	3.0		

^aThis table is based on pressure heads of 450, 1800, 3150, and 4500, respectively, for the first, second, third, and fourth floor radiators, and on friction heads of 200 milinches for the first floor radiators and connections, and 700 milinches for all other radiators and their connections.

^bThe riser branches, the piping which connects the risers to the mains, are to be one size larger than the risers.

^cApproximate length of pipes in feet equivalent to one elbow in friction head. This value varies with the velocity.

^dVelocities apply to the riser branches.

risers with 1-in. riser branches and 1-in. radiator tappings. Also select $1\frac{1}{4}$ -in. return risers with $1\frac{1}{4}$ -in. riser branches, and $1\frac{1}{4}$ -in. radiator tappings. Similarly, for 18 Mbh, select $1\frac{1}{4}$ -in. flow and return risers and riser branches, and $1\frac{1}{4}$ -in. radiator tappings.

To develop a rule for determining radiator sizes, assume a system similar to that of Fig. 6, in which the total temperature drop is to be 35 F and which is equipped with 7 radiators, all radiators dissipating equal quantities of heat. The mean temperature of the water in the radiators will be reduced 5 F for each successive radiator. If the mean temperature of the water in the first radiator is 200 F, the mean temperature of the water in the seventh radiator will be 170 F, and, according to Table 3, Chapter 6, of the 1933 GUIDE, the heat dissipation of these two radiators will be to each other as 868 is to 617, or as 140 is to 100, and therefore if the last radiator is to dissipate as much heat as the first, its size must be 40 per cent larger.

Example 4. Design a two-pipe, direct return, gravity circulation system for the layout shown in Fig. 7. Assume that the main circuit from the boiler to the farthest flow riser and from the farthest return riser back to the boiler consists of 160 ft of pipe, 6 elbows, and 1 boiler.

TABLE 8. MAXIMUM CAPACITIES OF RADIATOR CONNECTIONS IN Mbh, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TEMPERATURE DROP OF 35 F THROUGH EACH RADIATOR

PIPE SIZE		EQUIVALENT LENGTH OF PIPE (FEET) ^a	1ST FLOOR	2ND, 3RD, AND 4TH FLOORS
Flow	Return		Mbh	Mbh
$\frac{1}{2}$	$\frac{1}{2}$	1.0	4.1	5.9
$\frac{1}{2}$	$\frac{3}{4}$		5.2	7.5
$\frac{3}{4}$	$\frac{3}{4}$	1.5	7.0	10.5
$\frac{3}{4}$	1		9.1	13.0
1	1	2.0	12.5	17.8
1	$1\frac{1}{4}$		17.5	23.2
$1\frac{1}{4}$	$1\frac{1}{4}$	3.0	23.8	33.2

^aApproximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

Solution. Replacing the boiler by 3 elbow equivalents and assuming that the largest size of the main will be about 3 in., the total equivalent length of the main will be 160 plus 45, or 205 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main, and that the temperature drop will be 35 F for the system, the pressure head caused by the difference in weight between the water in the flow and return risers joining the mains to the boiler will be about 0.6 in. of water, or about one-fourth of the pressure head produced by the circulating pump selected for the system of Fig. 3.

Table 6 may be used to determine the size of the main as follows: Refer to Column 8 and note that for Sections AB and IA, which supply 105.6 Mbh, a 3-in. pipe is too large and a $2\frac{1}{2}$ -in. pipe is too small; hence, select $2\frac{1}{2}$ in. for Section AB and 3 in. for Section IA. For Sections BC and HI, which supply 76.8 Mbh, a $2\frac{1}{2}$ -in. pipe is almost exactly the correct size and is selected for both sections.

For the forced circulation system of Fig. 5, the pressure head produced by the circulating pump is used to force the water through the mains and also through the risers. Gravity circulation systems have two distinct pressure heads. One is produced by the difference in weight of the water in the flow and return risers adjacent to the boiler, and is the boiler pressure head, which in this case is 0.6 in. The other pressure head is produced by the difference in weight of the water in the flow and return risers adjacent to the radiators, and is the radiator pressure head. If the temperature drop through the radiators is about 35 F, and if the story heights of the building are 9 ft and the distance from the center of the first floor radiator to the average level of the main is 3 ft, the

radiator pressure head of the first floor radiator is about 450 milinches and the pressure heads of the radiators on the upper floor are 1350 milinches greater than those on the next lower floors.

Tables 6 and 7 are based on the assumption that the boiler pressure head must be equal to the friction head in the mains, and that the several radiator pressure heads must be equal to the respective radiator and riser friction heads.

To design the radiator risers, use Table 7 and begin with the set nearest the boiler. The first floor risers must supply 28.8 Mbh. According to the table, $1\frac{1}{4}$ -in. flow and return risers will supply 26.0 Mbh; if the return riser is increased to $1\frac{1}{2}$ in., the capacity will be increased to 34.0 Mbh. This is considerably larger than necessary, and $1\frac{1}{4}$ -in. flow and return risers are selected. However, it must be remembered that the riser branches, which are the connections from the flow and return mains to the flow and return risers, are to be one size larger than the risers.

The second floor risers must supply 19.2 Mbh. According to the table, the capacity of 1-in. flow and return risers is 20.0 Mbh, and that size is selected.

The third floor risers must supply 9.6 Mbh. If a $\frac{1}{2}$ -in. flow and a $\frac{3}{4}$ -in. return riser are used, the capacity will be 8.0 Mbh; if both risers are $\frac{3}{4}$ in., the capacity will be 14.0 Mbh. The $\frac{3}{4}$ -in. pipe is selected for both risers.

To design the radiator connections, use Table 8 and note that for the first floor radiator connections the capacity of a $\frac{3}{4}$ -in. flow and 1-in. return is 9.1 Mbh, and that of a 1-in. flow and a 1-in. return is 12.5 Mbh. The former is more nearly the correct size, but since it is difficult to secure a good flow through first floor radiators, the 1-in. flow and return connection is selected. For the two upper floors, the capacity of a $\frac{3}{4}$ -in. flow and return connection is 10.5 Mbh, and that size is used.

As explained in the design of the forced circulation system of Fig. 5, the two-pipe direct return system of Fig. 7 will not function correctly unless its four sets of risers are balanced among themselves. This necessary balancing is accomplished by adding resistances to all risers, except the one farthest from the boiler, equal to the excess boiler pressure heads available for those risers above the boiler pressure head available for the farthest riser. For example, the first set of risers is 60 ft nearer the boiler than the last set. Since the flow and return mains are designed for a friction head of 3 milinches per foot (See Table 6, Column 8), the boiler pressure head available for the first set of risers is 360 milinches in excess of that available for the fourth set. The velocity in the riser branch is 3 in. per second (See Table 7) and, therefore, according to Table 5, an 0.65-in. orifice in a $1\frac{1}{4}$ -in. union should be used. This will provide a resistance of about 420 milinches. In the same manner it is found that for the second set of risers a resistance of 240 milinches is required and that an 0.70-in. orifice in a $1\frac{1}{4}$ -in. union will provide a resistance of 285 milinches. For the third set of risers, a resistance of 120 milinches is required and an 0.60-in. orifice in a 1-in. union will provide sufficient resistance.

MECHANICAL CIRCULATION

Circulating pumps for hot water systems may be used to provide the motive head for forced circulation systems as already described, or to improve the operation of gravity-designed systems. Small specially-designed centrifugal pumps installed on a by-pass with the necessary gate or check valves near the point where the return main enters the heater may be employed. Specially-designed, electrically-driven, propeller-type circulating pumps or units may also be employed. The latter are usually installed directly in the return main and are available for all commercial pipe sizes used for hot water heating. The motor switch may be under manual control, automatic control using thermostatic elements,

or tied in with the oil or gas burner switch which starts and stops the burner. For large capacities these units may be installed in multiple.

For exceptionally large installations such as central heating plants, circulating pumps of the centrifugal single stage type, having an average operating efficiency of 70 per cent against heads up to 125 ft, are sometimes used. It is generally advisable to install the pumps in duplicate to provide for contingencies and to insure continuous operation. In such cases each pump should be made equal to the maximum capacity required.

EXPANSION TANKS

When water at ordinary temperatures is heated or cooled, its volume is increased or decreased. This variation in the volume of the water in a heating system is generally provided for by means of an expansion tank into which the water can flow from the system during the heating-up periods and from which it can flow back into the system during the cooling-down periods.

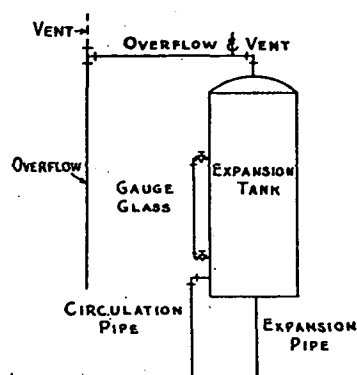


FIG. 9. AN OPEN EXPANSION TANK

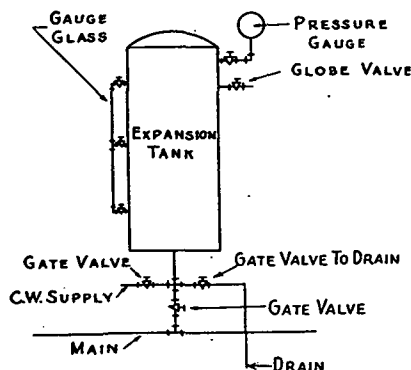


FIG. 10. A CLOSED EXPANSION TANK

The expansion tank may be open or closed. In an open expansion tank (Fig. 9), the water is subjected to atmospheric pressure and can expand freely without a material increase in pressure. In a closed expansion tank (Fig. 10), the water is subjected to the pressure of the compressed air within the tank, and as the water expands, the volume of the air in the tank is decreased and its pressure increased.

The open expansion tank must be placed at a sufficient elevation above the highest radiator to prevent boiling when the water in that radiator is at the highest temperature to which it is to be heated. For example, if the water is to be heated to 225 F on extremely cold days, the absolute pressure on the water in the highest radiator must be at least 19 lb per square inch. This pressure will be secured if the open expansion tank is located 15 ft above the highest radiator. If a closed expansion tank is used and is located 30 ft below the highest radiator, an absolute pressure of about 32 lb per square inch must be maintained in the expansion tank if the water in the highest radiator is to be heated to 225 F without danger of boiling.

The type of expansion tank used in a heating system, whether open or closed, has no influence on the operation of the system. The only function performed by the expansion tank is to provide for the variation in the volume of the water in the system, and at the same time to maintain a sufficient pressure in the system to prevent boiling when the water is at the highest temperature for which the system is designed. The use of an expansion tank may be dispensed with when the heating system is allowed to float on the water system, *i.e.*, when the connection between the heating system and the water system is kept open so that the water system replaces the expansion tank.

The capacity of the expansion tank should be at least twice the increase in volume produced when the water in the system is heated from its normal to its maximum temperature. When 25 gal of water are heated from 40 F to 200 F, the volume of water increases to 26 gal. A safe rule, therefore, is to make the water capacity of the expansion tank equal to 10 per cent of the capacity of the heating system.

In a forced circulation system, the expansion tank should be connected to the return main near the circulating pump. In a gravity circulation system, the expansion tank should be connected to the flow riser so that air liberated from the water in the boiler may escape through the expansion tank, except where it is desired to maintain a temperature higher than 212 F, in which case the connection should be in the return main to prevent possible boiling in the expansion tank.

The expansion tank should be protected so that the water in the tank or in the connecting pipe lines cannot freeze. If such water should freeze and the water in the system be heated to cause further expansion, the resulting force will burst the boiler or some other portion of the system.

INSTALLATION DETAILS

The detailed installation of the pipe system should be governed by four fundamental rules:

1. All piping must be pitched either up or down so that all gases which are liberated from the water can move freely to a vented section of the system. Whenever practicable, the pipe line should be pitched so that gases flowing to a vent will flow in the same direction as the water. When a pipe system cannot be installed without creating *air pockets*, that is, sections in the system from which liberated gases cannot escape, such sections must be provided with automatic air relief valves or with air valves which may be operated manually when necessary, or trapped into a pressure tank.
2. All piping must be arranged so that the entire system can be drained, either to permit alterations or repairs, or to prevent freezing if the system is not to be operated during a cold period.

It is well to install a gate valve and union in every riser near the main to permit the draining of individual risers without draining the entire system. It is also well, in large installations, to divide the system into branches and to provide each branch with unions and valves so that any one branch can be drained without disturbing the remaining ones.

The dividing of large heating systems into branches or zones and providing each zone with individual valves has the further advantage of permitting a varying temperature control. For example, if a building is equipped with a forced circulating system and if the south rooms are on one branch of the main and the north rooms are on a separate branch, the valves may be set so that the water will circulate through the north branch with a temperature drop of, say, 10 F, and through the south branch with a temperature drop of, say, 20 F, thus delivering less heat to the south rooms than to the north

rooms. This arrangement is especially valuable when the regulating valves are controlled thermostatically by the temperatures in the two zones, because no matter how accurately the heating system may have been designed, the heat demand of any group of rooms varies with sunshine and with wind velocity, and these intermittent variations can be provided for only by the individual control made possible by changing the valve settings controlling the heat supplied to particular groups of rooms.

3. 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.

4. The pipe system must be installed so that each circuit has its correct friction head. To bring this about, it is necessary in some cases to minimize the friction, *i.e.*, to make the pipe line as short as possible and to provide as few fittings as possible; and in other cases it is necessary to increase the length of the pipe and the number of fittings so that, for every circuit, the friction head will be equal to the available pressure head.

The connections from the boiler to the mains should be short and direct, to reduce the friction head. It is frequently possible to avoid an elbow and to reduce the length of the

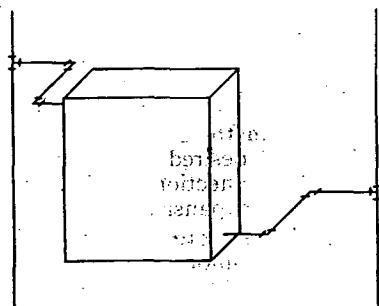


FIG. 11. METHOD OF CONNECTING RADIATOR TO ALLOW FOR EXPANSION OF PIPE

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 flow main should always be covered; the return main should be covered except where it is to provide the heating surface for the basement.

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.

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, but in most cases both connections should be at an angle of 45 deg. This method shortens the lines and substitutes 45-deg ells for 90-deg ells.

Preferably, connection of the flow riser to a radiator should be to the upper tapping; and connection of the return riser to a radiator should be to the lower tapping. When hot water enters at the top of a radiator it will distribute itself along the entire length of the radiator, and as it cools it will settle gradually to the bottom; the cool water may then be taken out of the radiator at either end.

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 and low velocities it makes little difference whether the water leaves at the end at which it enters or at the opposite end.

The connections of the risers to the radiators should be such that provision is made for the vertical expansion of the risers. This can be accomplished as indicated in Fig. 11 by using one tee and two ells for each connection. These connections should be pitched upward or downward, whichever may be necessary to prevent the formation of air pockets and to permit draining.

PROBLEMS IN PRACTICE

1 ● Will altering a hot water heating system from an open to closed type system (a) increase the circulation and (b) give more heat?

a. No. Tests conducted by the A.S.H.V.E. indicate that there is little, if any difference in the circulation when the system is under pressure. The difference in temperature between the supply and return, and the friction are the governing factors.

b. With a closed system the water may be carried at a higher temperature without boiling which permits warmer radiators.

2 ● What tends to prevent or to retard the circulation of water in hot water heating systems?

In both gravity flow and forced circulation systems, the friction which must be overcome when the water is flowing through pipes, fittings, valves, heaters, and radiators tends to prevent or retard circulation. For a given pipe the friction varies approximately as the 1.7 power of the velocity, and for given fittings, valves, heaters, and radiators, the friction varies approximately as the square of the velocity. It is therefore sufficiently accurate to express the friction in fittings, valves, heaters, and radiators in terms of the friction in one standard elbow, as shown in Table 1.

3 ● In the elementary heating system, Fig. 8, what is the pressure head maintaining the circulation if the water in the return riser is at 180 F and that in the flow riser is at 200 F?

It is found, from Table 7, Chapter 1, that 180 F water weighs 60.61 lb per cu ft and 200 F water weighs 60.13 lb per cu ft. The pressure head is independent of the size of the pipe. If the two risers were each 1 ft square, the water in the flow riser would weigh 601.3 lb and that in the return riser would weigh 606.1 lb. Thus the water in the return riser would weigh 4.8 lb more than that in the flow riser. Consequently, the resulting pressure head is 4.8 lb per square foot.

Pressure heads are generally expressed in feet, or inches, or milinches of water of a given temperature. In this case we are dealing with water at both 180 F and 200 F, so the pressure head is expressed in terms of 190 F water. Such water weighs 60.39 lb per cu ft, and to secure a pressure of 4.8 lb per square foot, it is necessary to have a column of water having a weight of 4.8 divided by 60.39 = 0.0795 ft, or 0.9540 in., or 954 milinches. This is the pressure head which maintains the circulation.

4 ● In the elementary system of Question 3, if the radiator dissipates 14,000 Btu per hour, what is the velocity of the water in the pipe line, if the pipes are 1 in. in diameter? What, if they are ¾ in. in diameter?

Since the temperature drop through the radiator is from 200 F to 180 F or 20 F, every pound of water flowing through the radiators delivers 20 Btu; consequently, 14,000 divided by 20 = 700 lb of water, or for 190 F water, 700 divided by 60.39 = 11.59 cu ft of water must flow through the radiator and through the pipe lines every hour.

The interior area of a 1-in. pipe is 0.864 sq in. The velocity in the 1-in. pipe is 11.59 divided by 0.864 and multiplied by 144 = 1932 ft per hour or 6.44 in. per second.

For ¾-in. pipe, the interior area is 0.533, and the velocity is 6.44 multiplied by 0.864 and divided by 533 = 10.44 in. per second.

5 ● If, in the elementary heating system of Question 3, a 1-in. pipe line is used, what would be the friction head?

If the radiator is connected as shown in Fig. 11, with the heater connected to provide freedom of expansion, the heating circuit may be assumed to consist of a heater, 25 ft of pipe, 8 elbows, 1 radiator valve, and 1 radiator. From Table 1 it appears that the heater and radiator are equivalent, in friction, to 6 elbows; hence, the circuit may be placed equal to 25 ft of pipe and 14 elbows.

From the diagram of Fig. 4 it appears that the friction head for a 1-in. pipe and a velocity of 6.44 in. per second is about 25 milinches per foot. For 25 ft of pipe, the friction head will be 625 milinches.

It appears from Table 1 that the friction head in one elbow is $\frac{v^2}{2g}$, or in this case 0.54 multiplied by 0.54 and divided by 64.4 = 0.0045 ft or 54 milinches. Hence, for the 14 elbows the friction is 756 milinches. For the entire circuit, the friction head is the sum of the 625 milinches of the pipe plus the 756 milinches of the elbows, or 1381 milinches which equal 1.381 in.

6 • If the elementary heating system of Question 3 is installed with a 1-in. pipe line, how will it function?

It is found from the answer to Question 3 that the pressure head is 954 milinches and from the answer to Question 5 that the friction head is 1381 milinches when the water is flowing with such velocity that the specified 14,000 Btu will be delivered with a 20 F temperature drop through the radiators. Since the pressure head is smaller than the friction head, the system will not function as planned for the water will flow through the system more slowly and remain in the radiator longer. The temperature drop through the radiator will be more than 20 F, and the difference in the weight of the water in the return and flow risers will be greater than that intended. The final result will be that the pressure head will become equal to the friction head at a value somewhere between 954 and 1381 milinches. Since the average water temperature in the radiator will be less than 190 F, the radiator should be larger than the size given in Question 4.

7 • Should a hot water heating system be designed to embody small pipes or large pipes?

As pipe sizes in gravity circulation heating are reduced, the friction head is increased and it is necessary to increase the temperature drop through radiators; this lowers the average temperature of the water in the radiators and necessitates an increase in the size of the radiators, so whereas the cost of the pipe in a system is reduced, the cost of the radiators is increased. For each installation there is a definite pipe size which entails maximum economy.

As pipe sizes in forced circulation systems are reduced, friction heads are increased so a circulating pump of greater size or capacity is required. Thus, by decreasing the size of the piping, both the first cost of the circulating pump and the cost of its operation are increased. There is a definite pipe size for every installation which is most economical. For each installation of both types of systems there is a definite pipe size entailing maximum economy which can be determined by a series of comparative calculations.

8 • What should be the size of the radiators for the elementary heating system of Question 3 in which the water enters the radiator with a temperature of 200 F and leaves with a temperature of 180 F? The average temperature of the water in the radiator is, approximately, 190 F.

If test results are available for the particular radiators to be used, and for the temperatures named, the size of the radiators should be selected from them. If no such test results are to be had, but if test results are available for the type of radiator to be used when it is supplied with 215 F steam and placed in a 70 F room, the required size may be determined by the following ratio: The required size is to the corresponding steam radiator size as $(215 - 70)^{1.2}$ is to $(190 - 70)^{1.2}$. This ratio works out to 1.28. Hence, the radiators should be 28 per cent larger under the conditions prescribed than are corresponding radiators under standard conditions. It is immaterial whether a radiator is filled with steam or with water, as long as the average temperature of its outer surface is the same in both cases.

Chapter 34

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.

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 or butt-weld process. While the lap-weld process produces a better weld than the butt type, lap-weld pipe is 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 considered to be more corrosion-resisting than ordinary steel pipe and therefore its somewhat higher

first cost can 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, such as molybdenum or manganese, has been claimed to possess more resistance to corrosion than plain steel pipe and it is 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.

DIMENSIONS

The *IPS* dimensions of commercial pipe universally used at the present time conform to the recommendations made by a Committee of the *A.S.M.E.* in 1886. Pipe up to 12 in. in diameter is made in certain definite sizes designated by nominal internal diameter which is somewhat different from the actual internal diameter, depending on the wall thickness required. There are three weights of wrought-iron and steel pipe commonly used, known as *standard-weight*, *extra-strong*, and *double-extra-strong*. Because of the necessity of maintaining the same external diameter in all three weights for the same nominal size, the added wall thickness is obtained by decreasing the internal diameter. The term *full-weight*, when applied to sizes below 8 in., means that the pipe is up to the nominal weight per foot. When applied to sizes between 8 and 12 in., inclusive, it often indicates that the pipe has the heaviest of several wall

thicknesses listed. 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 the *standard-weight* and *extra-strong* pipe, demands for pipe for higher pressures and temperatures in industry resulted in the use of a multiplicity of wall thicknesses for all sizes. Even in heating installations, the erection of piping by welding was deemed to warrant the use of pipe lighter than standard weight. For these reasons, a *Sectional Committee on Standardization of Wrought Iron and Wrought Steel Pipe and Tubing* functioning under the procedure of the *American Standards Association* was appointed to standardize the dimensions and materials of pipe.

The proposed pipe standard recommended by that sectional committee has set up several schedules of pipe including standard-weight and extra-strong thicknesses which are now included in Schedules 40 and 60, respectively. The schedules approved by the Sectional Committee are given in Tables 1 and 3 and the corresponding weights in Tables 2 and 4.

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:

- Class *K*—Designed for underground services and general plumbing service.
- Class *L*—Designed for general plumbing purposes.
- Class *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 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 5. 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 one hundred degrees 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* of that material, 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 6.

The elongation values in Table 6 were computed from the following formula:

$$L_t = L_o \left[1 + a \left(\frac{t - 32}{1000} \right) + b \left(\frac{t - 32}{1000} \right)^2 \right] \quad (1)$$

TABLE 1. DIMENSIONS OF WELDED AND SEAMLESS STEEL PIPE

NOMINAL PIPE SIZE	OUTSIDE DIAM.	NOMINAL WALL THICKNESSES FOR SCHEDULE NUMBERS									
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160
1/8	0.405				0.068*		0.095*				
1/4	0.540				0.088*		0.119*				
3/8	0.675				0.091*		0.126*				
1/2	0.840				0.109*		0.147*				0.187
3/4	1.050				0.113*		0.154*				0.218
1	1.315				0.133*		0.179*				0.250
1 1/4	1.660				0.140*		0.191*				0.250
1 1/2	1.900				0.145*		0.200*				0.281
2	2.375				0.154*		0.218*				0.343
2 1/2	2.875				0.203*		0.276*				0.375
3	3.500				0.216*		0.300*				0.437
3 1/2	4.000				0.226*		0.318*				
4	4.500				0.237*		0.337*	0.437			0.531
5	5.563				0.258*		0.375*	0.500			0.625
6	6.625				0.280*		0.432*	0.562			0.718
8	8.625	0.250	0.277*	0.322*	0.406	0.500*	0.593	0.718	0.812		0.906
10	10.75	0.250	0.307*	0.365*	0.500*	0.593	0.718	0.843	1.000	1.125	
12	12.75	0.250	0.330*	0.406	0.562	0.687	0.843	1.000	1.125	1.312	
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 O. D.	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 O. D.	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 O. D.	30.0	0.312	0.500	0.625							

All dimensions are given in inches.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12.5 per cent under nominal thicknesses.

*Thicknesses marked with asterisk in Schedules 30 and 40 are identical with thicknesses for standard-weight pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for extra-strong pipe in former lists.

The Schedule Numbers indicate approximate values of the expression $1000 \times P/S$.

where

L_t = length at temperature t degrees Fahrenheit, feet.

L_o = length at 32 F, feet.

t = final temperature, degrees Fahrenheit.

a and b are constants as follows:

METAL	a	b
Cast-Iron.....	0.005441	0.001747
Steel.....	0.006212	0.001623
Wrought-Iron.....	0.006503	0.001622
Copper.....	0.009278	0.001244

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.

TABLE 2. NOMINAL WEIGHTS OF WELDED AND SEAMLESS STEEL PIPE

NOMINAL PIPE SIZE INCHES	SCH. 10 PLAIN ENDS	SCH. 20 PLAIN ENDS	SCHEDULE 30		SCHEDULE 40		SCH. 60 PLAIN ENDS	SCH. 80 PLAIN ENDS	SCH. 100 PLAIN ENDS	SCH. 120 PLAIN ENDS	SCH. 140 PLAIN ENDS	SCH. 160 PLAIN ENDS
			Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings						
1/8					0.25*	0.25*		0.32*				
1/4					0.43*	0.43*		0.54*				
3/8					0.57*	0.57*		0.74*				
1/2					0.86*	0.86*		1.09*				1.31
3/4					1.14*	1.14*		1.48*				1.94
1					1.68*	1.69*		2.18*				2.85
1 1/4					2.28*	2.29*		3.00*				3.77
1 1/2					2.72*	2.74*		3.64*				4.86
2					3.66*	3.68*		5.03*				7.45
2 1/2					5.80*	5.82*		7.67*				10.0
3					7.58*	7.62*		10.3*				14.3
3 1/2					9.11*	9.21*		12.5*				
4					10.8*	10.9*		15.0*		19.0		22.6
5					14.7*	14.9*		20.8*		27.1		33.0
6					19.0*	19.2*		28.6*		36.4		45.3
8		22.4	24.7*	25.0*	28.6*	28.8*	35.7	43.4*	50.9	60.7	67.8	74.7
10		28.1	34.3*	35.0*	40.5*	41.2*	54.8*	64.4	77.0	89.2	105.0	116.0
12		33.4	43.8*	45.0*	53.6	55.0	73.2	88.6	108.0	126.0	140.0	161.0
14 O. D.	36.8	45.7	54.6	63.3	63.3		85.0	107.0	131.0	147.0	171.0	190.0
16 O. D.	42.1	52.3	62.6	82.8	82.8		108.0	137.0	165.0	193.0	224.0	241.0
18 O. D.	47.4	59.0	82.0	105.0	105.0		133.0	171.0	208.0	239.0	275.0	304.0
20 O. D.	52.8	78.6	105.0	123.0	123.0		167.0	209.0	251.0	297.0	342.0	374.0
24 O. D.	63.5	94.7	141.0	171.0	171.0		231.0	297.0	361.0	416.0	484.0	536.0
30 O. D.	99.0	158.0	197.0									

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*The weights marked with asterisk in Schedules 30 and 40 are identical with weights for standard-weight pipe in former lists; those in Schedules 60 and 80 are identical with weights for extra-strong pipe in former lists.

The Schedule Numbers indicate approximate values of the expression $1000 \times P/S$.

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 extensively in low-pressure steam and hot water heating systems and in hot water supply lines. The swivel joints absorb the expansive movement of the pipe by the turning of threaded joints. In many cases the straight pipe in the offset of a swivel joint is sufficiently

TABLE 3. DIMENSIONS OF WELDED WROUGHT-IRON PIPE

NOMINAL PIPE SIZE	OUTSIDE DIAMETER	NOMINAL WALL THICKNESSES FOR SCHEDULE NUMBERS					
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80
1/8	0.405	-----	-----	-----	0.070*	-----	0.098*
1/4	0.540	-----	-----	-----	0.090*	-----	0.122*
3/8	0.675	-----	-----	-----	0.093*	-----	0.129*
1/2	0.840	-----	-----	-----	0.111*	-----	0.151*
3/4	1.050	-----	-----	-----	0.115*	-----	0.157*
1	1.315	-----	-----	-----	0.136*	-----	0.183*
1 1/4	1.660	-----	-----	-----	0.143*	-----	0.195*
1 1/2	1.900	-----	-----	-----	0.148*	-----	0.204*
2	2.375	-----	-----	-----	0.158*	-----	0.223*
2 1/2	2.875	-----	-----	-----	0.208*	-----	0.282*
3	3.5	-----	-----	-----	0.221*	-----	0.306*
3 1/2	4.0	-----	-----	-----	0.231*	-----	0.325*
4	4.5	-----	-----	-----	0.242*	-----	0.344*
5	5.563	-----	-----	-----	0.263*	-----	0.383*
6	6.625	-----	-----	-----	0.286*	-----	0.441*
8	8.625	-----	-----	0.283*	0.329*	-----	0.510*
10	10.75	-----	-----	0.313*	0.372*	0.510*	0.606
12	12.75	-----	-----	0.336*	0.414	0.574	0.702
14 O. D.	14.0	0.250	0.312	0.375	0.437	0.625	0.750
16 O. D.	16.0	0.250	0.312	0.375	0.500	0.687	-----
18 O. D.	18.0	0.250	0.312	0.437	0.562	0.750	-----
20 O. D.	20.0	-----	0.375	0.500	0.562	-----	-----

All dimensions are given in inches.

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12.5 per cent under the nominal thickness.

*Thicknesses marked with an asterisk in Schedules 30 and 40 are identical with thicknesses for standard-weight pipe in former lists; those in Schedules 60 and 80 are identical with thicknesses for extra-strong pipe in former lists.

The Schedule Numbers indicate approximate values of the expression $1000 \times P/S$.

flexible to take up the expansion without developing enough thrust to produce swiveling in the threaded joint. This is preferable since continued turning in the threaded joint may in time result in a leak, particularly when the pressure is high. The amount of elongation which a swivel joint can take up is controlled by the length of the swing piece employed and by the lateral displacement which is permissible in the long pipe runs.

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 complicated¹. 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 the following formula:

$$L = 6.16 \sqrt{D \Delta} \quad (2)$$

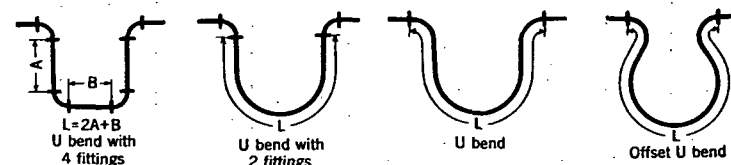
where

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

FIG. 1. MEASUREMENT OF L ON VARIOUS PIPE BENDS

stress of 16,000 lb per square inch. When square type bends are used, the width of the bend should not exceed about two times the height. 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.

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.

It is especially necessary with light-weight radiators so to anchor the piping and so to give it freedom for expansion that no strain therefrom shall be allowed to distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters they usually emit sounds of distress which are exceedingly troublesome.

¹Piping Handbook, by Walker and Crocker, and A Manual for the Design of Piping for Flexibility by the Use of Graphs, by E. A. Wert, S. Smith, and E. T. Cope, published by The Detroit Edison Company.

PIPE THREADS

All threaded pipe for heating and ventilating installations uses the American Standard taper pipe thread which is made with a taper of 1 in 16 measured on the diameter of the pipe so as to secure a tight joint. Threads of fittings are tapped to the same taper. The number of threads per inch varies with the different pipe sizes. All threaded pipe should be made up with a thread paste suitable for the service under which the pipe is to be used.

TABLE 4. NOMINAL WEIGHTS OF WELDED WROUGHT-IRON PIPE

NOMINAL PIPE SIZE (INCHES)	SCHED. 10	SCHED. 20	SCHEDULE 30		SCHEDULE 40		SCHEDULE 60	SCHEDULE 80
	Plain Ends	Plain Ends	Plain Ends	Threads and Couplings	Plain Ends	Threads and Couplings	Plain Ends	Plain Ends
1/8	-----	-----	-----	-----	0.25*	0.25*	-----	0.32*
1/4	-----	-----	-----	-----	0.43*	0.43*	-----	0.54*
3/8	-----	-----	-----	-----	0.57*	0.57*	-----	0.74*
1/2	-----	-----	-----	-----	0.86*	0.86*	-----	1.09*
3/4	-----	-----	-----	-----	1.14*	1.14*	-----	1.48*
1	-----	-----	-----	-----	1.68*	1.69*	-----	2.18*
1 1/4	-----	-----	-----	-----	2.28*	2.29*	-----	3.00*
1 1/2	-----	-----	-----	-----	2.72*	2.74*	-----	3.64*
2	-----	-----	-----	-----	3.66*	3.68*	-----	5.03*
2 1/2	-----	-----	-----	-----	5.80*	5.82*	-----	7.67*
3	-----	-----	-----	-----	7.58*	7.62*	-----	10.3*
3 1/2	-----	-----	-----	-----	9.11*	9.21*	-----	12.5*
4	-----	-----	-----	-----	10.8*	10.9*	-----	15.0*
5	-----	-----	-----	-----	14.7*	14.9*	-----	20.8*
6	-----	-----	-----	-----	19.0*	19.2*	-----	28.6*
8	-----	-----	24.7*	25.0*	28.6*	28.8*	-----	43.4*
10	-----	-----	34.3*	35.0*	40.5*	41.2*	54.8*	54.4
12	-----	-----	43.8*	45.0*	53.6	55.0	73.2	88.6
14 O. D.	36.0	44.8	53.6	-----	62.2	-----	87.6	104.0
16 O. D.	41.3	51.4	61.4	-----	81.2	-----	111.0	-----
18 O. D.	46.5	57.9	80.5	-----	103.0	-----	136.0	-----
20 O. D.	-----	77.0	103.0	-----	115.0	-----	-----	-----

Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

*Weights marked with an asterisk in Schedules 30 and 40 are identical with weights for standard-weight pipe in former lists; those in Schedules 60 and 80 are identical with weights for extra-strong pipe in former lists.

The Schedule Numbers indicate approximate values of the expression $1000 \times P/S$.

HANGERS

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 when expansion takes place, or the bottom support should be designed to withstand the entire load.

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 3 1/2 in.,

TABLE 5. STANDARD DIMENSIONS, WEIGHTS, AND DIAMETER AND WALL THICKNESS TOLERANCES FOR COPPER WATER TUBES*
(All Tolerances Plus and Minus)

NOMINAL SIZE, IN.	ACTUAL OUTSIDE DIAMETER, IN.	PERMISSIBLE VARIATION IN MEAN OUTSIDE DIAMETER, IN.		WALL THICKNESS, IN.						WEIGHT PER FT LB		
				Class K		Class L		Class M				
				Annealed	Hard Drawn	Nominal	Per- missible Variation	Nominal	Per- missible Variation	Nominal	Per- missible Variation	Class K
3/8	0.500	0.0025	0.001	0.049	0.004	0.035	0.0035	0.025	0.0025	0.269	0.198	0.144
1/2	0.625	0.0025	0.001	0.049	0.004	0.040	0.0035	0.028	0.0025	0.344	0.285	0.203
3/4	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	0.464
1 1/4	1.375	0.004	0.0015	0.065	0.0045	0.055	0.0045	0.042	0.0035	1.04	0.884	0.681
1 1/2	1.625	0.0045	0.002	0.072	0.005	0.060	0.0045	0.049	0.004	1.36	1.14	0.94
2	2.125	0.005	0.002	0.083	0.005	0.070	0.005	0.058	0.0045	2.06	1.75	1.46
2 1/2	2.625	0.005	0.002	0.095	0.005	0.080	0.005	0.065	0.0045	2.92	2.48	2.03
3	3.125	0.005	0.002	0.109	0.005	0.090	0.005	0.072	0.0045	4.00	3.33	2.68
3 1/2	3.625	0.005	0.002	0.120	0.005	0.100	0.005	0.083	0.005	5.12	4.29	3.58
4	4.125	0.005	0.002	0.134	0.006	0.110	0.005	0.095	0.005	6.51	5.38	4.66
5	5.125	0.005	0.002	0.160	0.006	0.125	0.006	0.109	0.005	9.67	7.61	6.65
6	6.125	0.005	0.002	0.192	0.006	0.140	0.006	0.122	0.005	13.87	10.20	8.91

*From Standard Specifications for Copper Water Tube of the American Society for Testing Materials, A.S.T.M. Designation B88-33.

and the latter for the larger sizes, 4 in. and above. Screwed fittings of 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.

As in the case of pipe, there are several weights of fittings manufactured. Recognized American Standards for the various weights are as follow:

Cast-iron pipe flanges and flanged fittings for 25 lb (sizes 4 in. and larger), 125 lb, and 250 lb maximum saturated steam pressure.

Malleable iron screwed fittings for 150 lb maximum saturated steam pressure.

Cast-iron screwed fittings for 125 and 250 lb maximum saturated steam pressure.

Steel flanged fittings for 150 and 300 lb maximum steam service pressure.

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.

TABLE 6. 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 Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe
		-20	0	0	0	0	664.3	500	3.847	4.296	4.477	6.110
		0	0.127	0.145	0.152	0.204	795.3	520	4.020	4.487	4.677	6.352
		20	0.255	0.293	0.306	0.442	945.3	540	4.190	4.670	4.866	6.614
		40	0.390	0.430	0.465	0.655	1115.3	560	4.365	4.860	5.057	6.850
29.39		60	0.518	0.593	0.620	0.888	1308.3	580	4.541	5.051	5.268	7.123
28.89		80	0.649	0.725	0.780	1.100	1525.3	600	4.725	5.247	5.455	7.388
27.99		100	0.787	0.898	0.939	1.338	1768.3	620	4.896	5.437	5.660	7.636
26.48		120	0.926	1.055	1.110	1.570	2041.3	640	5.082	5.627	5.850	7.893
24.04		140	1.051	1.209	1.265	1.794	2346.3	660	5.260	5.831	6.067	8.153
20.27		160	1.200	1.368	1.427	2.008	2705	680	5.442	6.020	6.260	8.400
14.63		180	1.345	1.528	1.597	2.255	3080	700	5.629	6.229	6.481	8.676
6.45		200	1.495	1.691	1.778	2.500		720	5.808	6.425	6.673	8.912
	2.5	220	1.634	1.852	1.936	2.720		740	6.006	6.635	6.899	9.203
	10.3	240	1.780	2.020	2.110	2.960		760	6.200	6.833	7.100	9.460
	20.7	260	1.931	2.183	2.279	3.189		780	6.389	7.046	7.314	9.736
	34.5	280	2.085	2.350	2.465	3.422		800	6.587	7.250	7.508	9.992
	52.3	300	2.233	2.519	2.630	3.665		820	6.779	7.464	7.757	10.272
	74.9	320	2.395	2.690	2.800	3.900		840	6.970	7.662	7.952	10.512
	103.3	340	2.543	2.862	2.988	4.145		860	7.176	7.888	8.195	10.814
	138.3	360	2.700	3.029	3.175	4.380		880	7.375	8.098	8.400	11.175
	180.9	380	2.859	3.211	3.350	4.628		900	7.579	8.313	8.639	11.360
	232.4	400	3.008	3.375	3.521	4.870		920	7.795	8.545	8.867	11.625
	293.7	420	3.182	3.566	3.720	5.118		940	7.989	8.755	9.089	11.911
	366.1	440	3.345	3.740	3.900	5.358		960	8.200	8.975	9.300	12.180
	451.3	460	3.511	3.929	4.096	5.612		980	8.406	9.196	9.547	12.473
	550.3	480	3.683	4.100	4.280	5.855		1000	8.617	9.421	9.776	12.747

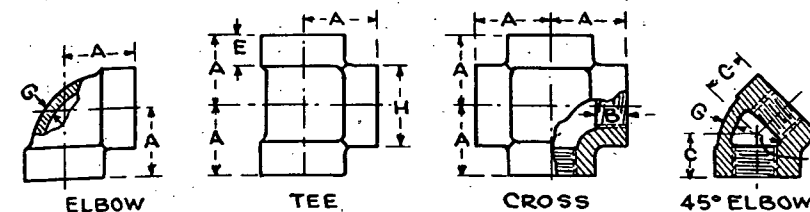
^aFrom *Piping Handbook*, by Walker and Crocker. This table gives the expansion from -20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 - 0.593 = 1.926 in.

Screwed fittings include: nipples or short pieces of pipe of varying lengths; couplings, usually of wrought iron only; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close 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 is shown in Fig. 2.

TABLE 7. TENTATIVE AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45 DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON SCREWED FITTINGS



NOMINAL PIPE SIZE	A	C	B	E	F INSIDE DIAMETER OF FITTING		G METAL THICKNESS, MIN.	H OUTSIDE DIAMETER OF BAND, MIN.
	CENTER TO END, ELBOWS, TEES AND CROSSES	CENTER TO END, 45 DEG ELBOWS	LENGTH OF THREAD MIN.	WIDTH OF BAND, MIN.	Min.	Max.		
1/4	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
3/8	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
1/2	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
3/4	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
1 1/4	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1 1/2	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
2 1/2	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
3 1/2	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	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50	5.97	1.88	1.88	12.750	12.850	0.800	15.47
14 O.D.	10.40		2.00	2.00	14.000	14.100	0.880	16.94
16 O.D.	11.82		2.20	2.20	16.000	16.100	1.000	19.30

All dimensions given in inches.

Fittings for copper pipe of IPS dimensions are available in screwed or soldered types of connection.

The compression type fitting is generally limited to smaller size tubing while the flared and soldered types are used in large and small sizes. While no effort has been made to standardize dimensions of flared tube fittings, manufacturers have quite generally used S.A.E. standard dimensions. Flared tube fittings are widely used in refrigeration work and the use of S.A.E. dimensions and a 45 degree flare renders most fittings

interchangeable, although for refrigeration use, thread fits and tolerances on thread gages must be maintained within close limits.

Ammonia pipe fittings made of cast iron are extensively used in handling refrigerants in larger installations. Until recently, no standard dimensions were adhered to in the manufacture of ammonia flanged fittings or companion flanges with the result that fittings of different manufacturers were not interchangeable. A subcommittee of A.S.A Sectional Committee B16 has prepared proposed American Standard dimensions for ammonia flanged fittings and companion flanges for maximum service pressure of 300 lb per sq in. which will be available soon.

Thread Connections

Threads used for fittings are the same American Standard taper pipe threads as those used for pipe, and unless otherwise ordered, right-hand

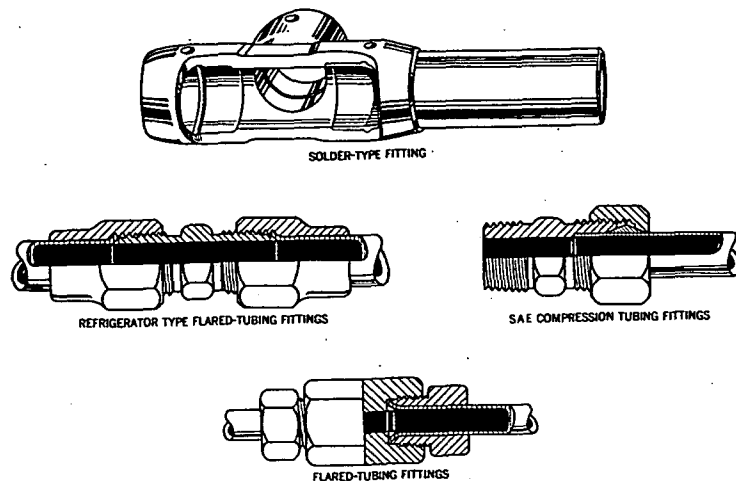


FIG. 2. COPPER OR BRASS TUBING FITTINGS

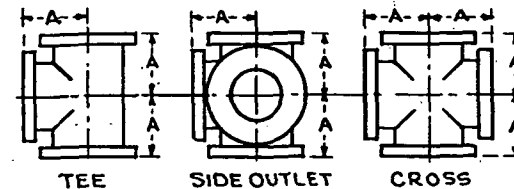
threads are used. To facilitate drainage, some elbows have the thread tapped at an angle to provide a pitch of the connecting pipe of $\frac{1}{4}$ in. to the foot. These elbows are known to the trade as pitched elbows and are commercially available. Malleable iron fittings, like brass fittings, are 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.

Flanged fittings are generally used in the best practice for connecting all piping above 4 in. in diameter. While screwed fittings may be used for the larger sizes and are satisfactory under the proper working conditions, it will be found difficult either to make or to break the joints in these large sizes.

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 lb pressure and below are normally furnished with a plain face, while the 250 lb cast-iron fittings are supplied with a $\frac{1}{16}$ -inch raised face. The standard facing for steel flanged fittings for 150 and 300 lb is a

TABLE 8. AMERICAN STANDARD DIMENSIONS OF TEES AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON FLANGED FITTINGS



NOMINAL PIPE SIZE a-b	A	AA	DIAMETER OF FLANGE	THICKNESS OF FLANGE, MIN.	METAL THICKNESS OF BODY, MIN.
	CENTER TO FACE TEES AND CROSSES b-c	FACE TO FACE TEES AND CROSSES b-c			
1	3½	7	4¼	⅞	⅞
1¼	3¾	7½	4⅝	⅞	⅞
1½	4	8	5	⅞	⅞
2	4½	9	6	⅞	⅞
2½	5	10	7	⅞	⅞
3	5½	11	7½	⅞	⅞
3½	6	12	8½	⅞	⅞
4	6½	13	9	⅞	⅞
5	7½	15	10	⅞	⅞
6	8	16	11	1	⅞
8	9	18	13½	1⅞	⅞
10	11	22	16	1⅞	⅞
12	12	24	19	1⅞	⅞
14 O.D.	14	28	21	1⅞	⅞
16 O.D.	15	30	23½	1⅞	1
18 O.D.	16½	33	25	1⅞	1⅞
20 O.D.	18	36	27½	1⅞	1⅞
24 O.D.	22	44	32	1⅞	1⅞
30 O.D.	25	50	38¼	2⅞	1⅞
36 O.D.	28	56	46	2⅞	1⅞
42 O.D.	31	62	53	2⅞	1⅞
48 O.D.	34	68	59½	2⅞	2

All dimensions given in inches.

aSize of all fittings listed indicates nominal inside diameter of port.

bTees, side outlet tees, and crosses, 18 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.

cTees 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.

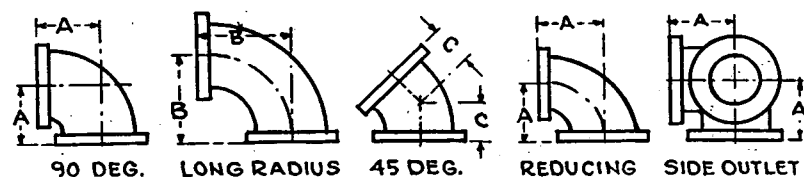
$\frac{1}{16}$ -inch raised face although these fittings are obtainable with a variety of facings. The gasket surface of the raised face may be finished smooth or may be machined with concentric or spiral grooves often referred to as serrated face or phonograph finish, respectively.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed

fittings are given in Table 7, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Tables 8 and 9.

For low temperature service not to exceed about 220 F, a number of paper or vegetable fiber gasket materials will prove satisfactory; for plain

TABLE 9. AMERICAN STANDARD DIMENSIONS OF ELBOWS FOR 125 LB CAST-IRON FLANGED FITTINGS



NOMINAL PIPE SIZE ^a	A CENTER TO FACE ELBOW b-b	B CENTER TO FACE LONG RADIUS ELBOW b-b	C CENTER TO FACE 45 DEG ELBOW c	DIAMETER OF FLANGE	THICKNESS OF FLANGE, MIN.	METAL THICKNESS OF BODY, MIN.
1	3 1/2	5	1 3/4	4 1/4	7/16	7/16
1 1/2	3 3/4	5 1/2	2	4 5/8	1/2	7/16
2	4	6	2 1/4	5	9/16	7/16
2 1/2	4 1/2	6 1/2	2 1/2	6	5/8	7/16
3	5	7	3	7	1 1/16	7/16
3 1/2	5 1/2	7 3/4	3 1/2	7 1/2	3/4	7/16
4	6	8 1/2	4	8 1/2	13/16	7/16
5	6 1/2	9	4 1/2	9	1 1/16	1 1/2
6	7 1/2	10 1/4	5	10	1 1/8	1 1/2
8	8	11 1/2	5 1/2	11	1 1/8	1 1/2
10	9	14	6 1/2	13 1/2	1 1/8	5/8
12	11	16 1/2	7 1/2	16	1 3/8	3/4
14 O.D.	12	19	8 1/2	19	1 1/2	1 1/4
16 O.D.	14	21 1/2	9 1/2	21	1 3/8	1 1/8
18 O.D.	15	24	11	23 1/2	1 7/8	1 1/2
20 O.D.	16 1/2	26 1/2	15	25	1 7/8	1 1/2
24 O.D.	18	29	18	27 1/2	1 7/8	1 1/2
30 O.D.	22	34	21	32	1 7/8	1 1/2
36 O.D.	25	41 1/2	24	38 3/4	2 1/8	1 1/2
42 O.D.	28	49		46	2 3/8	1 1/2
48 O.D.	31	56 1/2		53	2 3/8	1 1/2
	34	64		59 1/2	2 3/4	2

All dimensions given in inches.

^aSize of all fittings listed indicates nominal inside diameter of port.

^bReducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows corresponding to the size of the larger opening.

^cSpecial 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.

^dSide outlet elbows shall have all openings on intersection center-lines.

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 tem-

perature conditions, but preferably only with a relatively 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 a competitive 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.

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 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. Industry will not accept the employment of mechanics of undetermined ability nor on the basis of past experience. Neither does industry accept the statement that a weld is only as good as the workman who makes it. The control Codes now in process of adoption will be the law governing the use of the welding process. These Codes prohibit individual practices contrary to their specified procedure and rules of control, and this is predicated upon the sound requirement that the employer must assume full responsibility for the deposited weld.

It is advisable that this management responsibility be included in all welding specifications and that authoritative standards of workmanship also be specified. The standards of workmanship for this industry are as set forth in the *Standard Manual on Pipe Welding* of the Heating, Piping and Air Conditioning Contractors National Association.

A complete line of manufactured steel welding fittings is now available with plain ends machine beveled for welding and with radii similar to short and long radius flanged fittings. Some typical types of these fittings are shown in Fig. 3. They are made in pipe sizes 3/4 to 24 in., standard and extra heavy, in steel, wrought iron, brass, copper, and special alloys.

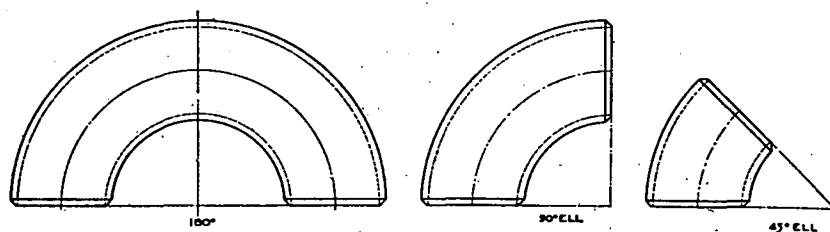
Socket welding fittings of forged steel are also commercially available. These fittings have a machined recess into which the pipe slips. A fillet weld between the pipe and socket edge provides a pressure-tight joint. A

proposed American Standard containing dimensions of steel welding-neck flanges for pressures up to 1500 lb per sq in. has been developed in A.S.A. Sectional Committee B16. Tables 10 and 11 give these dimensions for welding-neck flanges suitable for 150 and 300 lb per sq in. gage pressure.

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 sizes either cast-iron, cast-steel or some of the steel alloys are employed.



a. TYPICAL SHORT RADIUS ELBOWS

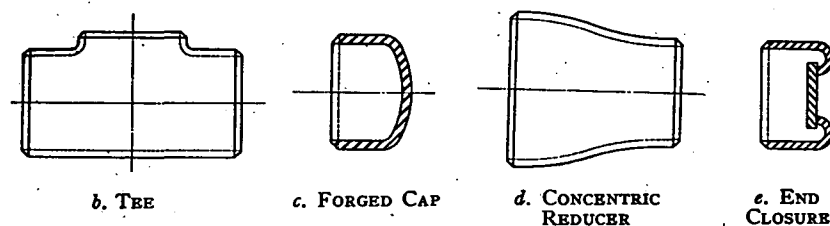


FIG. 3. TYPICAL WELDING FITTINGS

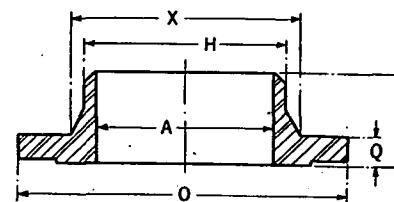
Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed.

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. 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

TABLE 10. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 150 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 500 F, AND 100 LB AT 750 F



NOMINAL PIPE SIZE	DIAMETER OF FLANGE	THICKNESS OF FLG. MIN.	DIAMETER OF HUB	HUB DIAM. BEGINNING OF CHAMFER	LENGTH THRU HUB	DIAM. FOR STANDARD PIPE	DIAM. OF BOLT CIRCLE	NO. OF BOLTS	SIZE OF BOLTS
	O	Q	X	H	Y	A			
1	4 1/4	3/16	1 1/2	1.32	2 3/4	1.05	3 1/8	4	1/2
1 1/2	4 5/8	1/2	2 1/2	1.66	2 1/4	1.38	3 1/2	4	1/2
1 1/2	5	3/4	2 3/4	1.90	2 1/4	1.61	3 3/8	4	1/2
2	6	5/8	3 1/4	2.38	2 1/2	2.07	4 3/4	4	5/8
2 1/2	7	1 1/16	3 3/4	2.88	2 3/4	2.47	5 1/2	4	5/8
3	7 1/2	3/4	4 1/4	3.50	2 3/4	3.07	6	4	5/8
3 1/2	8 1/2	1 1/16	4 1 3/4	4.00	2 1 3/4	3.55	7	8	5/8
4	9	1 1/2	5 1/4	4.50	3	4.03	7 1/2	8	5/8
5	10	1 5/8	6 1/4	5.56	3 1/2	5.05	8 1/2	8	3/4
6	11	1 7/8	7 1/4	6.63	3 1/2	6.07	9 1/2	8	3/4
8	13 1/2	1 1/2	9 1/4	8.63	4	7.98	11 3/4	8	3/4
10	16	1 3/4	12	10.75	4	10.02	14 1/4	12	7/8
12	19	1 3/4	14 3/8	12.75	4 1/2	12.00	17	12	7/8
14 O. D.	21	1 3/4	15 3/4	14.00	5	13.25	18 3/4	12	1
16 O. D.	23 1/2	1 1/2	18	16.00	5	15.25	21 1/4	16	1
18 O. D.	25	1 3/4	19 3/8	18.00	5 1/2	17.25	22 3/4	16	1 1/8
20 O. D.	27 1/2	1 1/2	22	20.00	5 1/2	19.25	25	20	1 1/8
24 O. D.	32	1 3/8	26 3/8	24.00	6	23.25	29 1/2	20	1 1/4

All dimensions given in inches.

A raised face of 1/4 in. is included in thickness of flange minimum.

It is recommended that the taper of the hub should not exceed 6 degrees for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding.

may prevent complete drainage of the pipe line. These objections are of particular importance in heating work.

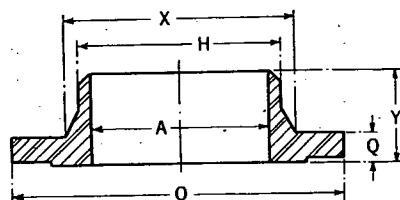
Check valves are automatic in operation and permit flow in only one direction, depending for operation on the difference in pressure between the two sides of the valve. The two principal kinds of check valves are the swing check in which a flapper is hinged to swing back and forth, and the lift check 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 in any position. Standard roughing-in dimensions for angle-type valves are given in Table 12.

Automatic control of steam supply to individual radiators can be

TABLE 11. PROPOSED DIMENSIONS OF STEEL WELDING NECK FLANGES FOR MAXIMUM STEAM SERVICE PRESSURE OF 300 LB PER SQ IN. (GAGE) AT A TEMPERATURE OF 750 F



NOMINAL PIPE SIZE	DIAM. OF FLANGE	THICKNESS OF FLANGE MIN.	DIAM. OF HUB	HUB DIAM. BEGINNING OF CHAMFER	LENGTH THRU HUB	DIAM. FOR STANDARD PIPE	DIAM. FOR EXTRA STRONG PIPE	DIAM. OF BOLT CIRCLE	NO. OF BOLTS	SIZE OF BOLTS
	O	Q	X	H	Y	A	A			
*2	6½	⅞	3½	2.38	2¾	2.07	1.94	5	8	⅝
2½	7½	1	3½	2.88	3	2.47	2.32	5½	8	¾
3	8½	1½	4½	3.50	3½	3.07	2.90	6½	8	¾
3½	9	1½	5½	4.00	3¾	3.55	3.36	7½	8	¾
4	10	1½	5½	4.50	3¾	4.03	3.83	7½	8	¾
5	11	1½	7	5.56	3¾	5.05	4.81	9¼	8	¾
6	12½	1½	8½	6.63	3¾	6.07	5.76	10½	12	¾
8	15	1½	10½	8.63	4½	7.63	7.98	13	12	¾
10	17½	1½	12½	10.75	4½	10.02	9.75	15¼	16	1
12	20½	2	14¾	12.75	5½	12.00	11.75	17¾	16	1½
14 O. D.	23	2½	16¾	14.00	5½	13.25		20¼	20	1½
16 O. D.	25½	2½	19	16.00	5¾	15.25		22½	20	1½
18 O. D.	28	2½	21	18.00	6¼	17.25		24¾	24	1½
20 O. D.	30½	2½	23½	20.00	6¾	19.25		27	24	1½
24 O. D.	36	2½	27½	24.00	6¾	23.25		32	24	1½

*For sizes below 2 inches use dimensions of 600 lb flanges.

All dimensions given in inches.

A raised face of ⅛ in. is included in thickness of flange minimum.

It is recommended that the taper of the hub should not exceed 6 degrees for a reasonable distance back of the chamfer in order to reduce the heat transfer while welding.

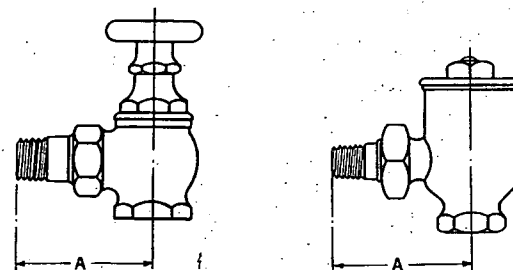
effected by use of direct-acting radiator valves having a thermostatic element at the valve, or near to it. The direct-acting valve is usually an angle-type valve containing a thermostatic element which permits the flow of steam in accordance with room temperature requirements. These valves usually are 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 ⅛-in. hole through the web forming the seat to insure sufficient circulation to prevent freezing when the valve is closed. Valves made particularly

for use in hot water heating systems are of less complex 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

TABLE 12. STANDARD ROUGHING-IN DIMENSIONS ANGLE TYPE VALVES



SIZE OF VALVE	DIMENSION A STEAM AND HOT WATER ANGLE VALVES AND UNION ELBOWS EFFECTIVE JANUARY 1, 1926	DIMENSION A MODULATING VALVES EFFECTIVE JANUARY 1, 1926	DIMENSION A RETURN LINE VACUUM VALVES EFFECTIVE JANUARY 1, 1925
½	2¼	2¾	3¼
¾	2¾	2¾	
1	3	3	
1¼	3½	3½	
1½	3¾	3¾	
2	4¼	4¼	
Tolerance	+⅛	+⅛	

All dimensions given in inches.

Connecting ends shall be threaded and gaged as to threading according to the American (Taper) Pipe Thread Standard, A.S.A. No. B2—1919.

The standardization of the Roughing-in Dimensions of Angle Steam and Hot Water, and Modulating Radiator Valves was made possible by the cooperation of the Manufacturers Standardization Society of the Valves and Fittings Industry.

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 the pressure dies down and a vacuum tends to be formed the air is drawn back into the radiator.

A system supplied with vacuum valves will heat more quickly and stay warm longer than one provided with straight pressure air valves; thus it will effect considerable economy of fuel because the idle period during which no heat is delivered is shortened. Automatic air valves are provided with a float to close them in case the radiator becomes flooded with water because it does not drain properly.

CORROSION²

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 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 Dalton³. The exact application of this law, however, assumes equilibrium conditions which do not always exist under the flow conditions prevailing in a heating system.

²New Light on Heating System Corrosion, by J. H. Walker (*Heating and Ventilating*, May, 1933).

³Some 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).

Distinction should be made between corrosion in heating systems proper and in the condensate discharge lines from other apparatus using steam, such as water heaters, kitchen equipment, and sterilizers. Experience has shown that in heating systems the partial pressures of the gases do not reach such magnitudes as to cause harmful amounts of gas to become dissolved in the condensate when steam supplies are of reasonable purity. In other kinds of steam-using apparatus which are not ordinarily well vented, the gases tend to accumulate in the steam space and to become dissolved in the condensate in appreciable concentrations. Consequently, corrosion is frequently observed in the condensate discharge lines from such apparatus, but this does not necessarily indicate that equally serious corrosion is taking place in the heating system supplied with steam from the same source.

When corrosive conditions are believed to exist, their seriousness should be 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 corrosiveness 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.

Corrosion, if found to exist, can be lessened or overcome by several means. If the steam supply is found to be definitely contaminated, proper chemical treatment of the water, followed by deaeration, is an obvious remedy. The leaks in the piping system, particularly in vacuum systems, should be stopped so far as is practicable.

Some success has been reported with the use of inhibitors, chief among which are oil, sodium silicate, and ammonia. Oil may be fed into the main steam-supply pipe by means of a sight-feed lubricator. The type of oil known as 600-W is usually recommended. In the present state of knowledge on this point, the quantity to be fed can best be determined by trial. The use of sodium silicate, fed in a similar manner, is reported to be successful but it has not been widely used.

The effect of ammonia is to increase the *pH* value of the condensate above the point where corrosion is likely to take place. Speller⁴ reports having injected small quantities of ammonia into a small closed heating system (the entire amount of condensate being returned to the boiler) and finding the *pH* value maintained at a high point for several months without further additions of ammonia. The concentration of ammonia must be kept low to avoid corrosion of brass parts of the system. The use of ammonia is not to be recommended where steam may come in contact with food or other materials.

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

⁴Corrosion in Steam Heating Systems, by F. N. Speller (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

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.

PROBLEMS IN PRACTICE

1 ● What is the meaning of IPS brass pipe?

It means that the brass pipe has the same external diameter as steel pipe in the same nominal pipe size and that the wall thickness is sufficient to allow cutting of threads for use with standard size threaded fittings.

2 ● Why is thin-walled copper pipe made up with sweated joints?

If the pipe were threaded it would be necessary to use at least standard-weight wall thickness on account of the metal removed in threading. Flared ends with coupling nuts may be used, but this construction is expensive and hard to keep tight.

3 ● How are pipes designated in diameters of 12 in. and less?

By weight and nominal size, referring to the approximate inside diameter.

4 ● How are pipe sizes designated in diameters of 14 in. and more?

By wall thickness and outside diameter.

5 ● Why are expansion joints required in steam pipes?

To care for the change in length of the line brought about by a change in temperature.

6 ● What devices are used for taking up expansion?

Expansion joints, swivel joints, and the inherent flexibility of the pipe itself.

7 ● Where are swivel joints principally used?

In branch connections to radiators, and in the risers of multi-story buildings where they are installed between the floor joists.

8 ● Name three grades of American Standard screwed pipe fittings.

125-lb cast-iron, 150-lb malleable iron, and 250-lb cast-iron.

9 ● In what sizes are American Standard cast-iron flanges and flanged fittings for 25-lb saturated steam pressure made?

In nominal sizes from 4 in. to 72 in., inclusive.

Chapter 35

WATER SUPPLY PIPING AND WATER HEATING

Maximum Possible Flow, Maximum Probable Flow, Average Probable Flow, Factor of Usage, Kind of Pipe Used, Sizing of Risers, Sizing of Mains, Sizing of Systems, Hot Water Supply, Hot Water Heating, Hot Water Storage, Swimming Pool Heating Requirements

DOMESTIC water supply systems present the engineer with a design problem that requires combining the somewhat empirical rules and formulæ in use with the more or less exact hydraulic principles involved. Unlike heating and ventilating layouts, there are practically no definite data for estimating the quantity of water likely to be consumed or the probable rate of water flow at any particular moment.

Metered results in one building often show two or three times the metered amount in another building of the same size and with the same type of tenants. In hotels, one riser will often have an almost constant flow that may never be reached by another at peak load. In office buildings, the women's toilets show a far greater daily consumption than those of the men, yet at no time will they approach the hourly consumption of the men's toilet during the first hour of the day. This condition has led to a multiplicity of rules of practice which vary as much as the data used. All must of necessity be based on an assumed rate of consumption and on an assumed probability of simultaneous use, and while the formulæ employed may have been derived on sound technical bases the assumptions are often in error.

To arrive at a safe standard, the approximate rate of flow of each fixture to be supplied must be known and the probable number of fixtures in use at any one time must be assumed. Obviously, the maximum number of fixtures assumed to be in use must be taken at the peak of demand and the lines must be made adequate to supply such a peak regardless of the riser or branch on which the demand may occur. This means that all water piping under the usual conditions will be over-sized.

In tall buildings it is customary to divide the water supply systems, both hot and cold, into sections of 10 to 20 stories. Such zoning¹ or

¹It is impractical to attempt to size piping so as to produce the proper pressure on fixtures at different levels by employing friction, owing to the fact that this friction will be built up to the amount desired only in times of maximum demand and at all other times the friction will be only a fraction of the maximum friction so that the fixtures by this method are subjected to a varying pressure on the water supply line. A much more practical method is to throttle the flow at the fixture, or to use flow regulators, so that the quantity of water delivered will approximate the fixture demands and so that this is accomplished without splashing or noise.

sectionalizing is for the purpose of avoiding excessive pressures on the fixtures in the lower stories of each system. This limits the consideration of water pipe sizes to horizontal mains and to risers not exceeding 20 stories in height or about 200 ft.

For the purpose of this chapter the following terms will be used and should be clearly distinguished from one another:

Maximum Possible Flow: The flow which would occur if the outlets on all fixtures were opened simultaneously. This condition is seldom, if ever, obtained in actual practice except in cases of gang showers controlled from one common valve, and similar conditions.

Maximum Probable Flow: The maximum flow which any pipe is likely to carry under the peak conditions. This is the most important amount to be considered in pipe sizing.

Average Probable Flow: The flow likely to be required through the line under normal conditions.

It is evident that any pipe adequate to take care of the *maximum*

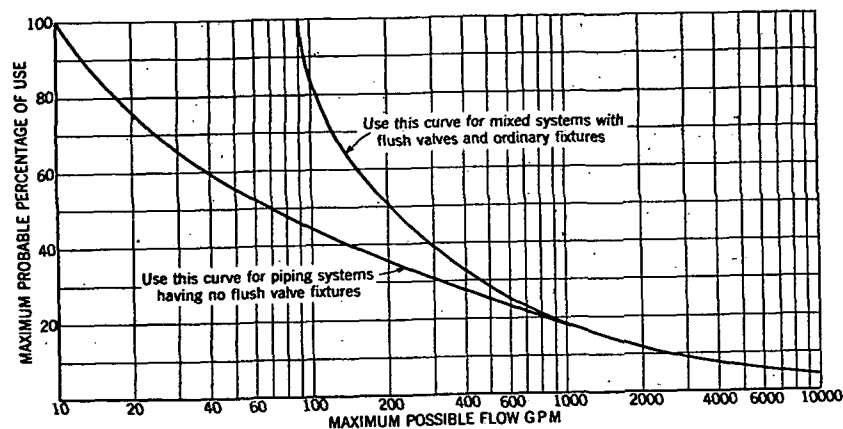


FIG. 1. CHART SHOWING RELATION BETWEEN MAXIMUM POSSIBLE FLOW AND MAXIMUM PROBABLE PERCENTAGE OF USE

probable flow will also be more than able to take care of the *average probable flow*, and hence the latter has no bearing on the pipe size.

MAXIMUM PROBABLE FLOW

There are two factors to be considered in calculating the maximum probable flow, namely, (1) the quantity of water that will flow from the outlets when they are open, and (2) the number of outlets likely to be open *at the same time*. Table 1 shows the maximum approximate rate of flow from each fixture when it is in use, and will serve as a guide in estimating maximum probable flow demands although there is considerable variation in different fixtures and valves. Probably the flow under normal water pressures, or with the pressure properly throttled, will not differ greatly from the values stated. With the aid of this table it is possible to calculate the maximum possible flow with all outlets open in both the hot and cold water lines.

To obtain the maximum probable flow it is necessary to multiply the maximum possible flow by a factor of usage, and this factor varies with the installation and the number of fixtures in the installation. It is evident that with two fixtures it is quite possible that both will at some time be in operation simultaneously. With 200 fixtures, it is unlikely the entire 200 would ever operate at the same time. Consequently, the factor of usage reduces as the number of fixtures becomes greater, all other things being equal.

TABLE 1. APPROXIMATE FLOW FROM FIXTURES UNDER NORMAL WATER PRESSURES

FIXTURES	COLD WATER (GALLONS PER MINUTE)	HOT WATER (GALLONS PER MINUTE)
Water-closets, flush valve.....	45 ^a	0
Water-closets, flush tank.....	10	0
Urinals, flush valve.....	30 ^a	0
Urinals, flush tank.....	10	0
Urinals, automatic tank.....	1	0
Urinals, perforated pipe per foot.....	10	0
Lavatories.....	3	3
Showers, 4 in. heads, ½ in. inlets.....	3	3
Showers, 6 in. heads or larger.....	6	6
Needle bath.....	30	30
Shampoo spray.....	1	1
Liver spray.....	2	2
Manicure table.....	1½	1½
Baths, tub.....	5	5
Kitchen sink.....	4	4
Pantry sink, ordinary.....	2	2
Pantry sink, large bibb.....	6	6
Slop sinks.....	6	6
Wash trays.....	3	3
Laundry tray.....	6	6
Garden hose bibb.....	10	0

^aActual tests on water-closet flush valves indicate 40 gpm as the maximum rate of flow with 30 lb pressure at the valve; this would increase to 60 gpm (about 50 per cent) at 90 lb pressure. The 45 gpm has been taken as an average flow; possibly, with very low pressures just sufficient to operate the flush valve, 30 gpm could be allowed with safety. Urinal flush valves would vary proportionately in the same manner.

In practice all the elements will vary according to conditions; in the case of flush valve closets the duration of flush with the kind and condition of supply apparatus, the interval between flushes with the number of people using the system and their habits; and the length of the rush period with the type of installation and its location. The effect of each of these time elements on the results should be considered in connection with any data on which it is based before passing judgment on the selection of the factor of usage. The longer the duration of the flush the greater is the probability of overlapping flow. In selecting the factor of usage shown in Fig. 1 for systems having flush valves, 10 seconds was chosen as the maximum duration of flush, a value that represents an approximate average as water closets are installed.

While the curve has been calculated for systems composed of water closets alone, it is possible to calculate probabilities for mixed systems of water closets and other smaller fixtures. It has been found however that for two systems both having the same maximum possible flow, one composed entirely of water closets and the other a mixed system of water

closets and smaller fixtures, the probability of a given rate of flow is greater for the system composed of water closets than for the mixed

TABLE 2. SCHEDULE OF SIZES FOR DOWN-FEED RISER (SEE FIG. 2)

ALLOW- ABLE DROP PER LB RISER	POR- TION OF LB RISER	MAXIMUM PROBABLE FLOW, GALLONS PER MINUTE																
		5	10	15	20	25	30	40	50	60	70	80	90	100	125	150	200	250
3.5	T	¾	1	1¼	1½	1¾	2	2	2¼	2½	2¾	3	3	3	3	3	3¾	3¾
20	S	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	2	2	2	2	2	2	2¾	3
30	R	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	Q	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	P	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	O	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	N	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	M	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	L	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	K	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	J	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	I	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	H	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	G	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	F	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	E	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	D	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	C	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	B	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾
30	A	¾	¾	¾	1	1	1¼	1½	1¾	1¾	1¾	1¾	2	2	2	2	2¾	2¾

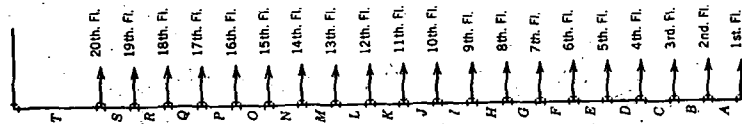


FIG. 2. TYPICAL RISER FOR 20-STORY BUILDING

system. The use of this chart then would produce results which would be on the safe side for mixed systems.

For systems composed entirely of fixtures other than flush valve fixtures the curve has been extended for smaller maximum possible flow values.

This chart applies to a normal building and not to installations where the inmates may all be required, for instance to bathe on certain days of the week and at certain hours of those days; or in schools for example where all the showers in the gymnasium may be used simultaneously after instruction periods. In such special cases a new factor of usage must be developed based on the maximum probable usage under the conditions involved.

Example 1. Assume that in a normal building, such as a residential hotel or an apartment house, there are 50 flush valve water-closets, 50 lavatories, 50 sinks and 50 baths, and that it is desired to determine the maximum probable flow in a line supplying all of these fixtures with both hot and cold water.

Cold Water

50 W. C. x 45 gpm.....	2250 gpm
50 Lavs. x 3 gpm.....	150 gpm
50 Sinks x 4 gpm.....	200 gpm
50 Baths x 5 gpm.....	250 gpm

Maximum possible flow.....2850 gpm

Fig. 1 shows a factor of usage of 9 per cent.

Maximum probable flow of cold water is 2850×0.09 257 gpm

Hot Water

50 Lavs. x 3 gpm.....	150 gpm
50 Sinks x 4 gpm.....	200 gpm
50 Baths x 5 gpm.....	250 gpm

Maximum possible flow..... 600 gpm

Fig. 1 shows a factor of usage of 23 per cent.

Maximum probable flow of hot water is 600×0.23 138 gpm

Total for main supplying cold and hot water ($2850 + 600$) $\times 0.08$ 276 gpm

It should be noted that this is a rate of flow or an instantaneous demand.

KIND OF PIPE USED

Before entering into the actual sizing of pipe, it is necessary to consider the kind of pipe to be used, and to make suitable allowance for corrosion and fouling during the lifetime of the system. For example, if brass, copper or alloy pipe is contemplated, it is probable that the quantities indicated in Example 1 are ample; if galvanized pipe is to be used, then it is quite likely that after a period of say 15 years the area may be decreased as much as 25 per cent and the quantities of water assumed should be increased by 35 per cent to allow for this reduction of area; if the water contains lime it is possible that 50 per cent of the area may be lost and in such cases the flow should be doubled and no branch pipe connected to fixtures should be less than $\frac{3}{4}$ in. In all of the following calculations, the assumption is made that the water is fairly good and that a corrosion resistant type of pipe is to be used.

SIZING A DOWN-FEED RISER

Down-feed systems are commonly used for tall buildings. In sizing a riser arranged for down-feed, the gravity head permits a pressure drop that is almost prohibitive in an up-feed riser. There is a gain in riser head of 0.43×100 or 43 lb per 100 ft of run and hence it is quite permissible to size such a riser on the basis of a pressure drop of 30 lb per 100 ft of run, as the difference between the 43 lb generated and the 30-lb drop under maximum probable demand is ample to take care of the friction caused by the fittings. This method applied to the typical riser shown in Fig. 2 gives the schedule of sizes indicated in Table 2 for any flow from 5 to 250 gal.

SIZING AN UP-FEED RISER

When the riser is an up-feed, the opposite condition occurs; that is, there is a drop in pressure as the top of the riser is approached, due to the natural reduction in the gravity pressure, and to this must be added the pipe friction plus that introduced by the pipe fittings, all of which produce an excessive drop when compared to the conditions existing with a down-feed riser.

To size an up-feed riser the minimum pressure of the street main, or other source of supply, should be ascertained and from this should be subtracted the pressure to be maintained at the highest fixture, namely, 15 lb per square inch, plus the height in feet above the source of water pressure, multiplied by 0.43 to change from feet of head to pounds of pressure. The total length of run from the source of pressure to the farthest and highest fixture should be ascertained, and this should be changed to equivalent length of run to allow for the loss occasioned by

TABLE 3. APPROXIMATE ALLOWANCES FOR FITTINGS AND VALVES IN FEET OF STRAIGHT PIPE

SIZE OF PIPE (INCHES)	TYPE OF FITTING OR VALVE					
	90-Deg Elbow	45-Deg Elbow	Return Bend	Gate Valve	Globe Valve	Angle Valve
1/2	4	3	8	2	48	8
3/4	5	3	10	3	60	10
1	5	3	10	3	60	10
1 1/4	6	4	12	3	72	12
1 1/2	7	5	14	4	84	14
2	7	5	14	4	84	14
2 1/2	10	7	20	5	120	20
3	12	8	24	6	144	24
4	18	13	36	9	216	36
5	25	18	50	13	300	50
6	30	21	60	15	360	60

the pipe fittings. Table 3 gives the additional lengths necessary to allow for the various fittings and valves. The drop allowable in pressure per 100 ft of run may then be obtained by multiplying the surplus pressure (over that required for the gravity head and to supply 15 lb at the fixture) by 100 and by dividing this by the equivalent length of run to the farthest or highest fixture.

Where street water pressures are available the pressure drop through the meter and service pipe must be taken into consideration. Table 4 shows the pressure loss through meters. It also gives the minimum sizes of recommended service and maximum meter deliveries.

Example 2. Assume a street pressure of 60 lb, the height of the highest fixture 50 ft, and the length of the longest run 200 ft. Without knowing the additional length of pipe to be added for the fittings it will be assumed that this is about 100 ft. The surplus pressure which will be available for pressure drop will then be 60 lb - (15 lb + 50 ft × 0.43 lb) = 60 lb - (15 lb + 21.5 lb) = 23.5 lb.

To change this into drop per 100 ft: $\frac{23.5 \text{ lb} \times 100}{200 \text{ ft} + 100 \text{ ft}} = 7.8 \text{ lb per 100 ft.}$

The pipe may then be sized from the maximum probable flow by selecting a size that does not give a drop in excess of 7.8 lb per 100 ft.

It will be seen from Example 2 that it is impossible to size up-feed risers without determining the drop allowable in both the horizontal feed mains and the toilet room branches. Having once ascertained this allowable drop, it is simply a matter of applying it throughout the system.

TABLE 4. PRESSURE LOSS THROUGH WATER DISC METERS^a

A. W. W. A. Standards

RATE OF FLOW GPM	APPROX. PRESSURE LOSS THROUGH METERS, LB PER SQ IN. PIPE SIZE (IN.)						
	1/2	3/4	1	1 1/4	2	3	4
5	1.5	0.5	0.2				
10	6.0	2.0	1.0	0.2			
15	14.0	5.0	2.0	0.6	0.2		
20	25.0	9.0	3.5	1.0	0.4		
25		13.5	5.5	1.5	0.6		
30		19.5	8.0	2.0	0.9		
35			11.0	3.0	1.0		
40			14.0	4.0	1.5		
45			18.0	5.0	2.0		
50			22.0	6.0	2.5	0.7	
75				14.0	5.5	1.5	
100				25.0	10.0	2.8	1.0
125					15.0	4.0	1.5
150					22.0	6.0	2.2
175						8.0	3.0
200						10.4	4.0
250							16.0
300							23.0
350							
400							
500							6.5
600							9.0
800							16.0
1000							25.0

MINIMUM SIZE OF SERVICE RECOMMENDED						SAFE MAXIMUM DELIVERY OF METERS	
RATE OF FLOW GPM	APPROX. MINIMUM PIPE SIZE OF SERVICE, MAIN TO METER (IN.) MAXIMUM LENGTH (FT)					METER SIZE IN.	CAPACITY, GPM BASED ON 25 LB LOSS THROUGH METER
	30	75	100	150	200		
1-20	3/4	3/4	1	1	1	5/8	20
20-30	3/4	1	1	1 1/2	1 1/2	3/4	34
30-50	1	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	53
50-100	1 1/2	1 1/2	2	2	2	2	100
100-150	1 1/2	2	2	2 1/2	2 1/2	3	160
						6	315
						8	500
							1000

^aPressure loss through compound and current meters are less than shown in table. For exact information consult manufacturers.

HORIZONTAL SUPPLY MAINS

The horizontal mains supplying the risers at the top of a down-feed system must be liberally sized unless the house tank is set at a much higher elevation than usual. To provide a gravity head on the highest fixtures of 15 lb per square inch it is necessary for the water line in the house tank to be nearly 40 ft higher, and with the line loss considered this becomes about 45 ft. Such heights are not often practical and as a result the pressure on the highest fixtures either is reduced to 7 lb (which is sufficient to operate a flush valve), or flush tank water-closets are substituted, or a separate cold and hot water supply is installed with a small pneumatic tank to give the increase in pressure necessary. The chief objection to the use of a pneumatic tank is that a separate hot water heater is required and this heater must be located either sufficiently below the highest fixtures to obtain a gravity circulation, or it must be provided with a circulating pump in order to force the hot water to the top floor level.

The most common solution is to place the house tank as high as the structural and architectural conditions will permit and then to use liberally-sized lines between the house tank and the upper fixtures, say for the two top stories, below which the riser sizes may be reduced to those indicated in Fig. 2 and Table 2. Where the house tank is only one story above the top fixtures, flush tank water-closets must be used and the drop in the entire run from the house tank down to the farthest fixture should not exceed 1 lb; the less, the better. This means that if the total equivalent run to the farthest top fixtures supplied is 300 ft, the drop per 100 ft should not exceed $\frac{1 \text{ lb} \times 100}{300}$ or 0.33 lb per 100 ft. The friction

curves shown in Fig. 3 may be used for quickly determining the proper size of pipe to give any desired drop in pounds per 100 ft of equivalent run.

OVERHEAD DISTRIBUTION MAIN

Example 3. Suppose an installation has a house tank in which the water line is 20 ft above the level of the top fixtures to be supplied and that the length of run to the farthest fixtures on this level is 400 ft with the pipe fittings adding another 200 ft, making an equivalent length of 600 ft. What would be the size of main coming out of the tank where a maximum flow rate of 400 gpm may be expected, of the horizontal main where a maximum flow rate of 200 gpm may be expected, and of the riser down to the fixture level where the maximum flow rate is approximately 100 gpm?

Here the level of the water in the house tank is 20 ft above the faucet of the highest fixture and the gravity pressure will be $0.43 \text{ lb} \times 20 \text{ ft} = 8.6 \text{ lb}$ and, if a total pressure drop of 1 lb is assumed, the pressure on the farthest fixture under times of peak load will be $8.6 \text{ lb} - 1 \text{ lb} = 7.6 \text{ lb}$ while the drop per 100 ft of equivalent run will have to be $\frac{1 \text{ lb} \times 100}{600} = 0.1667 \text{ lb}$.

Referring to Fig. 3 it will be noted that where the flow through the main is 400 gpm, an 8 in. pipe would be required; that where the flow is reduced to 200 gpm, a 6-in. pipe would be sufficient; and that where the flow is 100 gpm in the riser branch and riser, a 5-in. size would be correct. Of course these are somewhat excessive flows and the head from the tank is small so that large sizes are to be expected. It would be necessary to carry a 5-in. riser down to the branch to the top floor, then reduce to 4 in. for the branch to the floor below the top, and below this the sizes in Table 2 could be followed. In such a case, flush tank closets should doubtless be substituted.

Had the tank been set 10 ft higher, the head available to be used up in friction, but

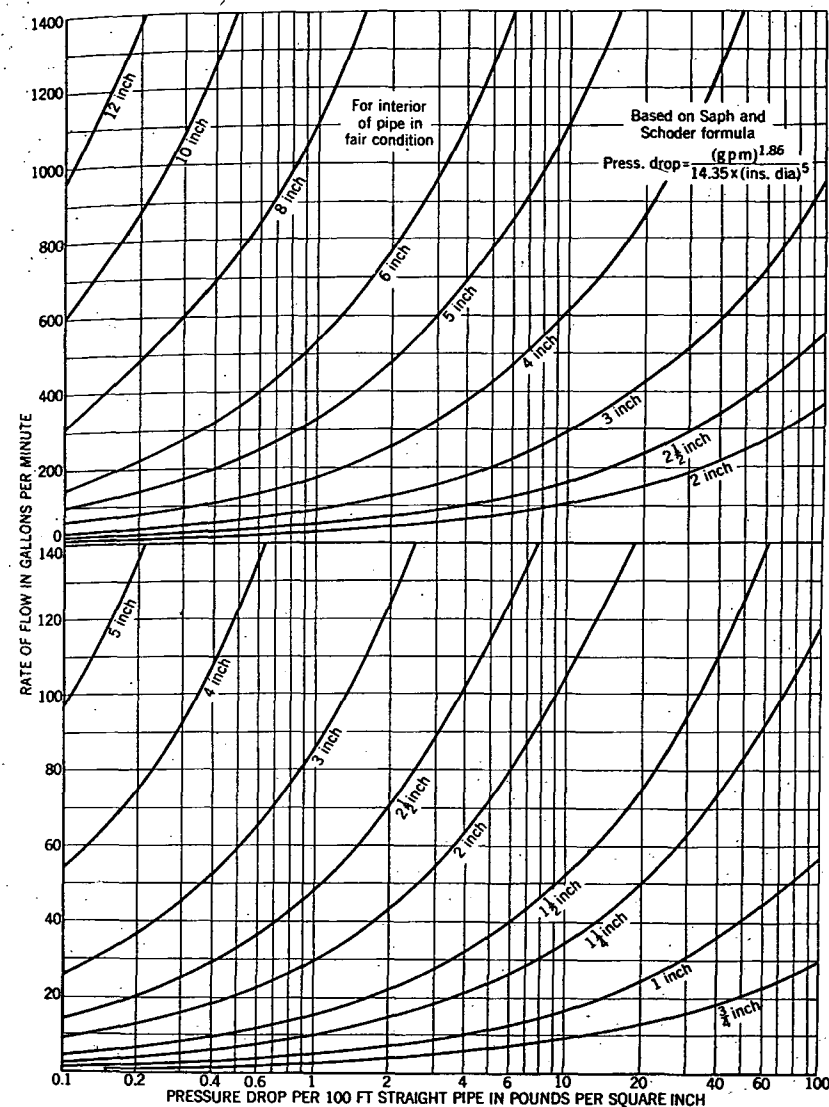
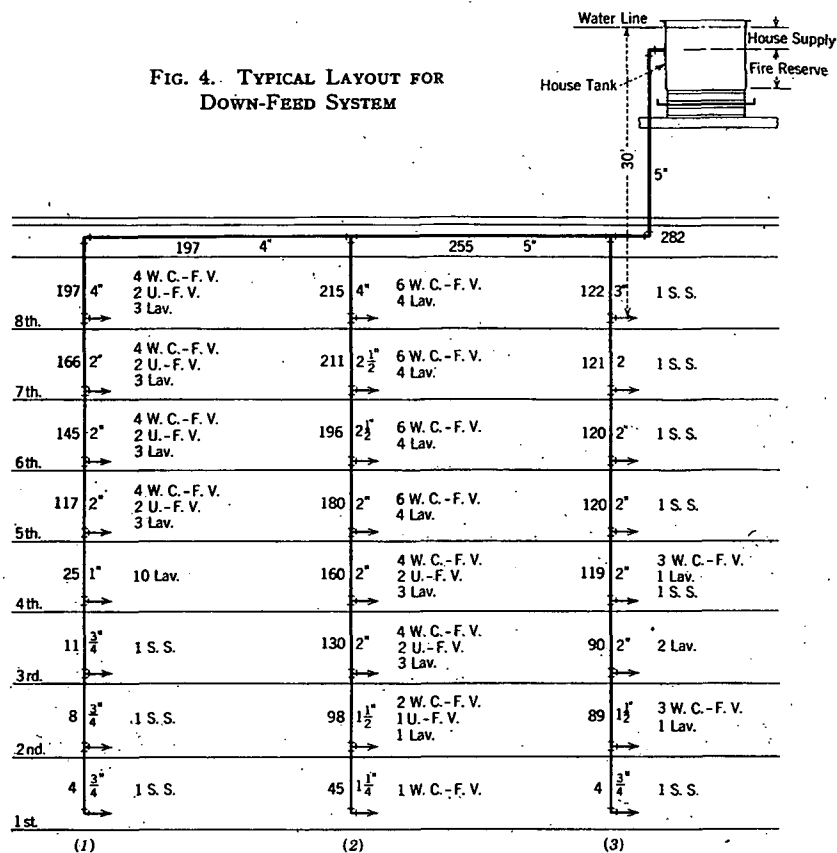


FIG. 3. CHART GIVING PRESSURE DROP FOR VARIOUS RATES OF FLOW OF WATER

still giving the same pressure at the top fixtures, would have been $0.43 \text{ lb} \times 10 \text{ ft}$ or 4.3 lb greater and this, with the 1 lb drop used previously, would give a total allowable drop of $1 \text{ lb} + 4.3 \text{ lb} = 5.3 \text{ lb}$ which, divided by the 600 ft equivalent run gives a drop per 100 ft of $\frac{5.3 \times 100}{600} = 0.9 \text{ lb}$

FIG. 4. TYPICAL LAYOUT FOR DOWN-FEED SYSTEM



and, with this drop, the sizes according to the chart (Fig. 3) are 6 in., 5 in., and 4 in., respectively, while if the run is reduced to 200 ft instead of 600 ft, the allowable drop will be $\frac{5.3 \text{ lb} \times 100}{200} = 2.7 \text{ lb per 100 ft}$. This gives 5 in., 4 in., and 3 in., respectively, for the flows of 400, 200, and 100 gpm.

From Example 3 it is evident that, while the down-feed system possesses certain economies in size for the riser portion, it is quite likely to involve large distribution main sizes, especially when the tank is not elevated to a considerable degree.

SIZING A PIPING SYSTEM

Example 4. Fig. 4 shows a typical layout with three risers extending eight stories and with the fixtures noted on each floor. First this will be solved for a down-feed arrangement assuming that the level of the water in the house tank is 30 ft above the fixtures on the top floor, that the length of run from the tank to the farthest fixture is 200 ft, equivalent length of fittings 100 ft, and the pressure required at the fixture is 7 lb.

TABLE 5. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 1. Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 S. S.	4	4	4	100	4	30	3/4
2nd	1 S. S.	4	4	8	100	8	30	3/4
3rd	1 S. S.	4	4	12	92	11	30	3/4
4th	10 Lav.	3	30	42	58	25	30	1
5th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9	249				
			249	291	40	117	30	2
6th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9	249				
			249	540	27	145	30	2
7th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9	249				
			249	789	21	166	30	2
8th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9	249				
			249	1038	19	197	2	4

The 30-ft head is equal to a static pressure of 0.43×30 or 12.9 lb per square inch and to maintain a pressure of 7 lb at the highest fixtures the drop allowable in pressure is $12.9 - 7.0 \text{ lb}$ or 5.9 lb. As the total equivalent run is 300 ft, this is a drop per 100 ft of 1.97 lb, or practically 2 lb. Therefore, all risers and mains from the top floor back to the tank must be sized on the basis of a drop of 2 lb per 100 ft. Tables 5, 6, 7 and 8 show the schedule for Risers Nos. 1, 2 and 3 with the maximum possible flow taken from Table 1, the percentage of use at the peak taken from Fig. 1, and the maximum probable flow at the peak worked out for each portion of the riser, the riser sizes being taken from Table 2 as far as possible and from Fig. 3 where the amounts exceed the values given in this table; a drop of 30 lb per 100 ft is used except on the riser from the top floor back to the tank where 2 lb per 100 ft is the allowable limit.

The reduction in pipe size which would occur if flush tank water-closets were used on the top floor and only 3 lb pressure used on the fixtures is given in Tables 9 and 10. This illustrates why flush tank closets so frequently are substituted on the uppermost floor when a house tank is the source of water pressure.

If it is now assumed that Riser No. 1 is to be fed from the bottom and the minimum street pressure is 75 lb with the top fixture of the riser 80 ft above the main, the problem would be solved by determining the maximum rate of flow in each portion of the riser as shown in Table 11 and then finding the allowable drop which can be used per 100 ft. The 80 ft of riser height will use up $0.43 \text{ lb} \times 80 = 34.4 \text{ lb}$ and the pressure at the top of the required 15 lb will make the total reduction 49.4 lb, leaving a balance of 25.6 lb which may be used up in friction. If the distance from the street main to the bottom of the riser, which will be assumed to be the farthest one on the horizontal line, is 100 ft, and if the fittings are sufficient to add another 100 ft, as well as the 80 ft of vertical distance up the riser, the total equivalent run will be 280 ft, which will be taken as an even 300 ft.

TABLE 6. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 2. Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 W. C.	45	45	45	100	45	30	1½
2nd	2 W. C. 1 U. 1 Lav.	45 30 3	90 30 3					
			123	168	58	98	30	1½
3rd	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9					
			249	417	31	130	30	2
4th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9					
			249	666	24	160	30	2
5th	6 W. C. 4 Lav.	45 3	270 12					
			282	948	19	180	30	2
6th	6 W. C. 4 Lav.	45 3	270 12					
			282	1230	16	196	30	2½
7th	6 W. C. 4 Lav.	45 3	270 12					
			282	1512	14	211	30	2½
8th	6 W. C. 4 Lav.	45 3	270 12					
			282	1794	12	215	2	4

Then the allowable drop per 100 ft will be $\frac{25.6 \text{ lb} \times 100}{300} = 8.5 \text{ lb}$ and the sizes shown in Fig. 5 are based on this amount of drop. Of course the other risers will have the same maximum flows at the bottom as they formerly had at the top, namely 215 and 122 gal, respectively, for Risers Nos. 2 and 3. Combining these maximum flows in the same manner as pursued in the down-feed system it is seen that the maximum flow between Riser No. 2 and Riser No. 3 is 255 gpm, and between Riser No. 3 and the street main, 282 gpm which at a drop of 8.5 lb gives the main sizes indicated. It will be noted that in determining the maximum flow in an up-feed riser it is necessary to begin at the top floor and work down instead of beginning at the bottom floor and working up as was done in the down-feed sizing.

SIZING UP-FEED AND DOWN-FEED HOT WATER SYSTEMS

Hot water supply systems, when of the circulating type, have a few differences to be considered although the same general principles of sizing apply to these lines as to the cold water lines. Owing to the fact that there are no flush valves on the hot water piping and also because many plumbing fixtures have no hot water connections, the sizes of the hot water piping in general will be considerably less than the cold water piping in the same building. On the other hand it is almost invariably

TABLE 7. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

(Riser No. 3. Fig. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 S. S.	4	4	4	100	4	30	¾
2nd	3 W. C. 1 Lav.	45 3	135 3	142	63	89	30	1½
			138					
3rd	2 Lav.	3	6	148	61	90	30	1½
4th	3 W. C. 1 Lav. 1 S. S.	45 3 4	135 3 4	290	41	119	30	2
			142					
5th	1 S. S.	4	4	294	41	120	30	2
6th	1 S. S.	4	4	298	40	120	30	2
7th	1 S. S.	4	4	302	40	121	30	2
8th	1 S. S.	4	4	306	40	122	2	3

required that a gravity circulation be kept up in such hot water lines and this often has a considerable influence on the size. There are three methods of arranging circulation lines, as follow:

1. By using the plain up-feed with a return carried back from the top of the riser and paralleling it.
2. By carrying a supply riser up in one location thus supplying fixtures on up-feed, then crossing over at the top and coming down past another collection of fixtures and supplying these by a down-feed.
3. By carrying all of the water to the top of the building and dropping risers wherever needed, feeding all hot water on a down-feed system.

TABLE 8. SIZE OF DISTRIBUTION MAIN FOR DOWN-FEED SYSTEMS (SEE FIG. 4)

RISER NO.	MAXIMUM GPM RISER	MAXIMUM GPM MAIN	PROBABLE USE (PER CENT)	PROBABLE GPM	ALLOWABLE DROP LB PER 100 FT	SIZE OF MAIN IN.
1	1038	1038	18	187	2	4
2	1794	2832	9	255	2	4
3	306	3138	9	282	2	5

In the first instance the up-feed riser may be sized for the same pressure drop as used for the cold water riser and, from the top of the riser *just below the top fixture connection*, a return circulation line may be carried back to the main return line in the basement and connected through a check valve, set on a 45-deg angle, and a gate valve; these return circulation lines should never be less than ¾ in., and on the farther half of the risers, not less than 1 in. to favor circulation in the far end. Typical top and bottom connections for such risers are shown in Fig. 6.

TABLE 9. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISERS WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY (SEE FIG. 4)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
<i>Riser No. 1</i>								
7th and below				789	21	166	30	2
8th	4 W. C. 2 U. 3 Lav.	10 10 3	40 20 9 69	858	20	172	3.3	4
<i>Riser No. 2</i>								
7th and below				1512	14	211	30	2½
8th	6 W. C. 4 Lav.	10 3	60 12 82	1594	14	223	3.3	4
<i>Riser No. 3</i>								
7th and below				302	40	121	30	2
8th	1 S. S.	4	4	306	40	122	3.3	3

For the second arrangement of hot water risers (Fig. 7b), circulation lines are run back from the last fixture supplied to the main return circulation line in the same manner as just described, using ¾ in. for the near risers and 1 in. for the far risers. The sizing is much more difficult, as it is necessary to start at the bottom floor of the return riser and work back to the top of this riser and then carry the maximum flow across on to the top of the corresponding supply riser and work down on this riser from the top floor to the bottom. Naturally this gives a much greater flow in the supply riser and aids circulation by reducing pipe friction. The allowable loss per 100 ft in such lines must be made about half that used for the cold water risers which do not have the combined up- and down-travel which the hot water must make.

In the third and most common arrangement (Fig. 7c) all of the water is carried from the tank or heater directly to the top of the building and is there distributed to the risers which are down-feed and may be sized in the

TABLE 10. SUMMARY OF RISER SIZES TO GIVEN MAIN SIZES WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY. (SEE FIG. 4)

RISER No.	MAXIMUM GPM RISER	MAXIMUM GPM MAIN	PROBABLE USE (PER CENT)	PROBABLE GPM	ALLOWABLE DROP LB PER 100 FT	SIZE OF MAIN IN.
1	858	858	20	172	3.3	4
2	1594	2452	10	245	3.3	4
3	306	2758	9	248	3.3	4

regular down-feed manner if the total equivalent run either from the street main or house tank is taken into consideration. The return circulation lines from the bottom of each riser should be arranged in the manner already outlined and any riser not going to the basement to supply fixtures must have these returns carried down to the basement from the termination of the supply riser at whatever level it may end.

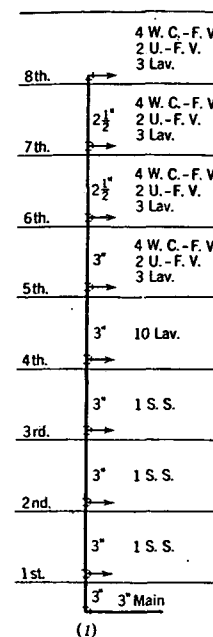


FIG. 5. UP-FEED SYSTEM

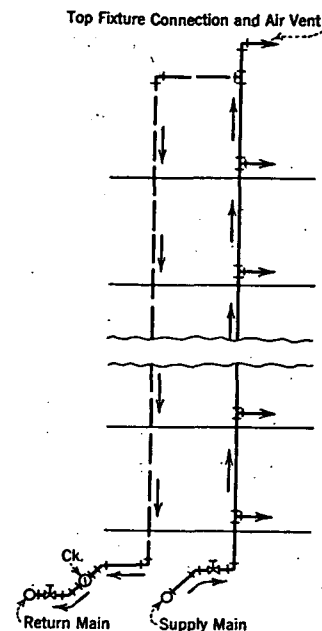


FIG. 6. SUPPLY AND RETURN MAIN CONNECTIONS FOR HOT WATER SUPPLY SYSTEM

All risers, both hot and cold, should be valved at the main with an extra check valve on the hot water return circulation so that the risers may be cut off and repaired when necessary without disturbing the service in the remainder of the system.

HOT WATER SUPPLY

Having designed the service hot water piping, the next step is to furnish some means of heating the water and in this respect it is necessary to pass from the maximum probable flow to the *maximum probable hourly demand*, which is quite different. If an instantaneous heater were used, it would require adequate capacity to provide for the heating of the water as fast as it is drawn and a heater of this type should be sized on the basis of the maximum probable flow with the accompanying heavy drafts on the heating device and with intervals of no draft at all. To balance these inequalities of flow the storage-type heater is often utilized so that the

TABLE 11. TYPICAL CALCULATION OF PIPE SIZES ON UP-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS (SEE FIG. 5)

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXTURE	MAXIMUM GPM ON FLOOR	MAXIMUM GPM ON RISER	PROBABLE USE (PER CENT)	PROBABLE DEMAND RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
8th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9	249	44	109	8.5	2½
7th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9	249	28	139	8.5	2½
6th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9	249	22	164	8.5	3
5th	4 W. C. 2 U. 3 Lav.	45 30 3	180 60 9	249	18	179	8.5	3
4th	10 Lav.	3	30	1026	18	185	8.5	3
3rd	1 S. S.	4	4	1030	18	186	8.5	3
2nd	1 S. S.	4	4	1034	18	187	8.5	3
1st	1 S. S.	4	4	1038	18	188	8.5	3

TABLE 12. SUGGESTED STORAGE TANK SIZES FOR HOMES AND APARTMENTS^a

ALL YEAR SERVICE BASED ON BOILER WATER AT 180 F				SERVICE DURING HEATING SEASON BASED ON BOILER WATER AT 215 F			
Tank Capacity Gal	Piping Connections		Number of Baths or Families	Tank Capacity Gal	Piping Connections		Number of Baths or Families
	Boiler, In.	Tank, In.			Boiler, In.	Tank, In.	
30	1	¾	1	30	1	¾	1
35	1¼	¾	1	40	1	¾	1
40	1¼	¾	1-2	52	1	¾	1
50	1¼	¾	1-2	66	1¼	1	1-2
60	1¼	1	1-2	82	1¼	1	2-3
72	1½	1	2-3	100	1¼	1	3
80	2	1	2-3	120	1¼	1	4
100	2	1¼	3-4	144	1½	1	5
125	2	1¼	4-5	160	2	1¼	6
150	2	1¼	5-6	200	2	1¼	6-7
200	2	1½	6-7	250	2	1½	7-9
250	2½	1½	7-9	300	2	1½	9-11
300	2½	1½	9-11	400	2½	2	11-15
400	3	2	11-15	500	2½	2	15-18
500	3	2	15-18	600	3	2½	18-21

^aSee pages 645 and 705 for further data.

water demand can be heated during periods of light demand and stored up for use during the periods of heavy demand. The total water consumption per person usually varies between 100 and 150 gal per day when laundry and culinary operations for the occupants are carried out on the same premises. The maximum hourly demand under these conditions will be found to be about one-tenth of the average daily consumption.

If one-third of the total water used is hot water and 125 gal per day is assumed as a fair average of consumption per person, it is apparent that each person uses about 40 gal of hot water per day. If one-tenth of this represents the peak hourly load, then 4 gph must be allowed per person for the heaviest demand. If the average occupancy of apartments is 3 persons, the peak hour demand per apartment will be about 12 gph. It is customary to allow 10 gph of heating capacity per apartment. Water in excess of this heating capacity drawn out during the peak hours is

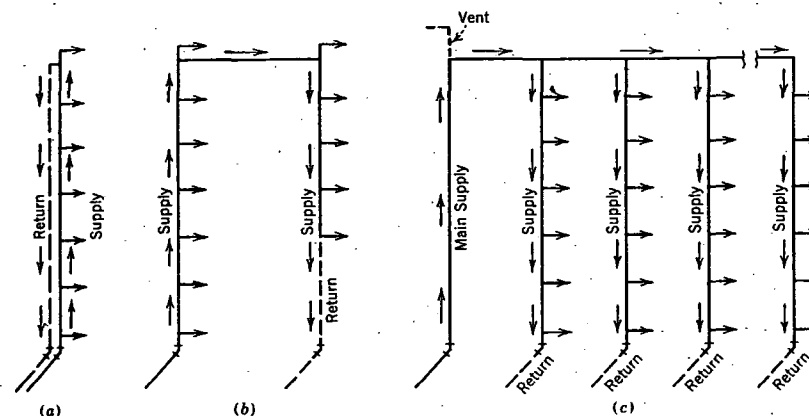


FIG. 7. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

provided for by storage in the hot water tank where this water is heated during hours when the demand is below the average. Table 12 gives suggested storage tank sizes for homes and apartments based on the number of families or baths.

HOT WATER HEATERS

Various types of heaters are available for supplying the hot water for domestic service in buildings. In any hot water supply system the water should be heated to a temperature between 150 and 180 F. Where the hot water requirements include supplies for kitchens, laundries or process work, the higher temperatures are used. In buildings where steam is available throughout the year, the hot water supply is usually taken from this source. In smaller domestic installations the fuel-burning device is generally automatically arranged so that hot water is supplied the entire year and not merely when the boiler is used for heating purposes.

Water is heated by various methods using heat exchangers arranged so that the boiler heating medium gives up its heat to the water in the hot water circulating system. These heat exchangers may be classified as follows:

1. Submerged steam heating coil in storage tank. (Fig. 8).
2. Submerged water heating coil in storage tank. (Fig. 9).
3. Indirect water heater, mounted on side of boiler below water line. (Fig. 9).
4. Submerged indirect water heater, placed in boiler below water line. (Fig. 10).

The efficiency of these heaters may be estimated as nearly 100 per cent as the heat loss from surface radiation of the heater and tank shell when covered with insulating material is generally reduced to a minimum. The capacities of these heaters are usually available from manufacturers

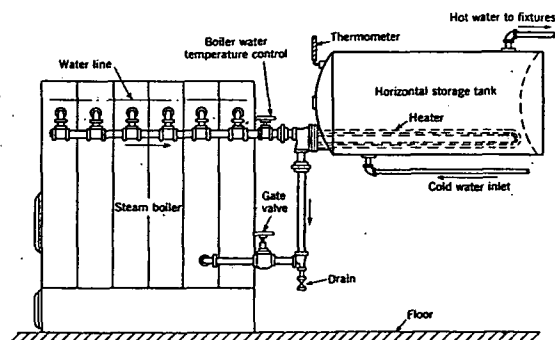


FIG. 8. HOT WATER HEATING COIL SUBMERGED IN STORAGE TANK

rating tables. The area of the inside surface of a heating coil may be determined from the following equation:

$$A = \frac{Q \times 8.33 (t_o - t_i)}{K_o \times t_m} \quad (1)$$

where

A = surface area of coil, square feet.

Q = quantity of water heated, gallons per hour.

t_o = hot water outlet temperature, degrees Fahrenheit.

t_i = cold water inlet temperature, degrees Fahrenheit.

K_o = coefficient of heat transmission, Btu per hour per square foot surface.

For copper or brass coils $K_o = 240$ (steam) and 100 (hot water).

For iron coils $K_o = 160$ (steam) and 67 (hot water).

t_m = logarithmic mean of the difference between the temperature of the heating medium and the average water temperature. t_m is approximately =

$$\left[t_o - \frac{(t_o + t_i)}{2} \right]$$

Equation 1 may also be used for determining revised heating coil ratings under different temperature conditions as stated in the manufacturers ratings. When selecting a water heater, the conditions of operation should be carefully considered, as well as the location of the

storage tank and the piping arrangement between the boiler, heater and tank. It is generally good practice to allow a margin of safety when selecting an indirect heater of the proper size to provide for loss of efficiency due to the accumulation of scaling on the coils and piping. Heat exchangers classified according to (3) and (4) may be used with or without a storage tank, but when tanks are omitted, the indirect water heaters should be increased in size so as to heat the water instantaneously as it is needed.

The storage tank should be installed as high as possible. Horizontal tanks are preferable for all medium size installations and absolutely essential on larger installations. Where possible the storage tank should be installed with the bottom of the tank at or above the boiler water line. Horizontal storage tanks smaller than 18 or 20 in. diameter are not

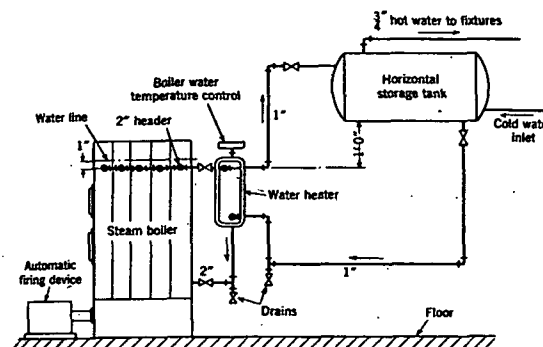


FIG. 9. INDIRECT WATER HEATER MOUNTED ON SIDE OF BOILER

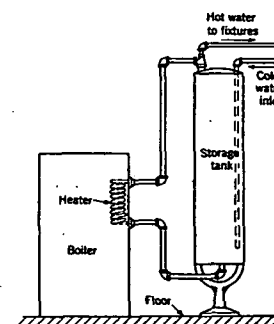


FIG. 10. INDIRECT WATER HEATER PLACED IN BOILER

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.

Pipe sizes between the water heater and boiler should be full size of the heater tapplings (Table 12). When a heater is connected to a horizontal sectional boiler, it is recommended that connections be made to all sections and joined together a few inches below the water line as shown in Fig. 8, so that steaming is prevented in those sections which are not connected to the header.

When a steam coil is used for heating the water, an automatic thermostatic valve may be installed in the steam supply to the coil. The operation of this automatic valve is controlled by a thermostat located in the storage tank which permits the proper amount of steam to enter the coil so as to maintain an even water temperature.

An indirect water heater may be used on either a steam or hot water system, and generally this type of heater is provided with a temperature control device located in the boiler water circulating connection to the water heater. The setting on this thermostatic valve may be as low as

140 F or as high as 180 F and may be readily adjusted to meet particular requirements. With this type of control it is impossible to overheat the hot water supply which is an important safety consideration in some installations. This type of system may also be conveniently used during the non-heating season with the operation of the fuel burning device controlled by the water heater thermostat. (See Fig. 10, Chapter 14.) During the heating season the water heater temperature control functions as a low limit control.

When an indirect water heater is applied to a gravity hot water system, it is necessary to provide a valve in the supply to the heating system to prevent the flow of hot water from the boiler when heat is not required in the house. This valve may be controlled from a room thermostat and the automatic fuel-burning device controlled from the water heater thermostat. To prevent circulation in a forced hot water heating system, flow control valves may be installed in the flow and return lines which act merely as check valves when the circulating pump is not operating. In this arrangement the pump is controlled by the room thermostat and the automatic fuel-burning device is controlled from the water heater thermostat.

STORAGE CAPACITY AND BOILER ALLOWANCES

The amount of storage provided in the hot water tank or heater is somewhat a matter of choice but is usually made ample to carry over the peak shortage which is likely to occur and is based on the assumption that only 75 per cent of the storage capacity will be available, as it has been found that if more than this amount is withdrawn from storage, the tank is so cooled down as to make the balance useless. The general rule may be cited that the less the heating capacity the greater must be the storage, and the greater the storage the less may be the heating capacity down to a point where the heating capacity will fail to be sufficient to heat up the tank storage during the periods of small load.

Example 5. A heater to supply 500 persons will have an average daily use of about $500 \times 40 \text{ gal} = 20,000 \text{ gal}$ and this is an average of $\frac{20,000 \text{ gal}}{24} = 833 \text{ gph}$ but the peak hour will require $\frac{1}{4}$ of 20,000 = 2000 gal and the shortage during the peak hour, if the heating capacity is made to suit the average hourly use of 833 gal, will be $2000 - 833 = 1167 \text{ gal}$ so that the storage capacity, based on 75 per cent being available from this capacity without cooling the tank excessively, will be $\frac{1167}{0.75} = 1556 \text{ gal}$.

Should it be desired to reduce the size of storage tanks and to use a greater heating capacity, it is only necessary to increase the heating capacity to say 1200 gph which then gives $2000 - 1200 = 800 \text{ gal}$ as the shortage during the peak hour, and the necessary storage will be $\frac{800 \text{ gal}}{0.75} = 1067 \text{ gal}$; or the heating capacity can be increased to 1500 gal, leaving a shortage of $2000 - 1500 = 500 \text{ gal}$.

Good design requires that the heating capacity be made as small as possible without introducing undesirable amounts of storage, as the heating capacity directly determines the load on the source of heat.

As indicated in Example 5, the heating load is proportional to the heating capacity and the boiler capacity must be increased for higher heating capacities and may be reduced for smaller heating capacities with greater storage. It may be assumed that a boiler capacity of about 4 sq ft* of equivalent steam heating surface (radiation) must be provided for every gallon of water heated 100 F or from 50 F to 150 F, which is the temperature rise most commonly assumed and required. On this basis it will be seen that the various conditions cited in Example 5 will require additional boiler capacity as follows:

Heating Capacity (gph)	Additional Boiler Capacity (Sq Ft EDR)
833	3332
1200	4800
1500	6000

From this it is apparent that it is less costly to provide ample storage and to reduce boiler capacity than to diminish the storage and supply a greatly increased boiler capacity to compensate.

The boiler allowance value of 4 sq ft of equivalent steam radiation for each gallon of water heated through a temperature range of 100 F is based on an hourly heating rate. When reduced heating capacities are desired for economic reasons of boiler design and selection, engineers frequently recommend that the heating rate be extended over a period of two hours in which case the boiler allowance value would be reduced to 2 sq ft of equivalent steam radiation. Similarly any other heating rate may be established and a corresponding value of boiler allowance determined.

Reliable information based upon the installations of several heaters in existing heating systems indicates varying arbitrary values of boiler allowances to be used. When these values are selected for usage, a careful analysis of the varying factors involved in determining these values should be considered so that the proper heating allowances may be provided.

ESTIMATING HOT WATER DEMAND BY FIXTURES

In buildings where the occupancy is doubtful and only the number of plumbing fixtures can serve as a basis for determining the probable hot water demand, the problem is not so simple owing to the fact that a fixture gives no information as to how heavy a service may be demanded from the fixture and this amount of service is really the governing factor in making an estimate of the probable hot water demand. Table 13 may prove of some value in this respect as it gives the maximum assumed quantity of hot water per hour which will be demanded of any fixture and then gives a percentage of this amount which may be assumed as probable in different types of buildings. Table 14 gives approximate hot water requirements in various types of buildings.

Example 6. Let it be assumed that an apartment house with 20 apartments has 20 baths, 20 lavatories, 20 kitchen sinks and 20 laundry trays; what is the probable maximum hourly demand for hot water?

*Actual requirement for 100-deg temperature difference = $\frac{100 \times 8.33}{240} = 3.33 \text{ sq ft per gallon of water heated}$.

20 Baths at 40 gal and 33 per cent.....	270 gal
20 Lavs. at 20 gal and 25 per cent.....	100 gal
20 Sinks at 30 gal and 33 per cent.....	200 gal
20 Trays at 50 gal and 60 per cent.....	600 gal
Total.....	1170 gal
Probable peak use at one time.....	35 per cent
Probable actual peak demand.....	409 gph

If three persons are assumed to an apartment the total daily use of hot water should approximate $20 \times 3 \times 40 \text{ gal} = 2400 \text{ gal}$ and if the peak hour is 10 per cent of this amount, the peak hour by this method shows a probable demand of one-tenth of 2400 gal, which indicates that the values in Table 13 are safe.

SWIMMING POOL HEATING REQUIREMENTS

Swimming pools present a problem of hot water heating demand which is frequently overestimated. Few outdoor swimming pools require water heating, and in some cases they require the addition of cold water to regulate the temperature. The recirculation system of a swimming pool consists of the pumps, hair and lint catchers, and filters together with all necessary pipe connections to the inlets and outlets of the pool. The water heater, the sterilizing equipment and suction cleaner are usually installed or connected to the recirculation system and may be considered as integral parts of the system.

The recirculation system and all its component parts should be designed to provide the required volume of circulation so that the water turnover ratio is at least two times per day and where heavy loads are anticipated the turnover ratio should be increased to three times or more. Many states have regulations prescribing the circulation turnover.

The water heaters for swimming pools are usually instantaneous steam

TABLE 13. ORDINARY MAXIMUM HOURLY DEMAND FOR HOT WATER FOR VARIOUS FIXTURES IN GALLONS AND PROBABLE PERCENTAGE OF USAGE

TYPE OF BUILDING	LAVATORIES		BATHS	SHOWERS	SLOP SINKS	KITCHEN SINKS	PANTRY SINKS	FOOT BATHS	WASH TRAYS	AV. MAX. USE*
	Private	Public								
MAXIMUM PROBABLE USAGE GPH	20	20	40	300	30	30	20	20	50	
Probable Usage in Per Cent of Maximum Ordinary Use										
Apt. house	25	50	33	67	67	33	50	25	60	35
Club	25	75	50	67	67	67	100	25	80	60
Gym.	25	100	100	100	—	—	—	100	—	80
Hospital	25	75	50	33	67	67	100	25	80	45
Hotel	25	100	50	33	100	67	100	25	80	70
Industrial	25	150	100	100	67	67	—	100	—	90
Laundries	25	100	—	—	33	—	—	—	100	100
Office building	25	75	—	—	50	—	—	—	—	20
Baths	25	150	150	100	50	—	—	—	—	100
Residences	25	—	50	33	50	33	50	50	60	50
Schools	25	75	—	100	67	33	100	50	—	25
Y. M. C. A.	25	100	100	100	67	67	100	100	80	75

*Percentage of fixtures likely to be demanding maximum probable usage at any one time.

coil heaters. These heaters should be sized so that they will have sufficient capacity to heat the water delivered by the circulating pump 15 F per hour.

The water temperature in a pool is usually maintained at about 72 F. A few states have regulations prohibiting higher water temperatures than 70 F. The room temperature should be approximately 5 F higher, but not more than 8 F higher nor less than 2 F lower, than the water temperature.

Example 6. Assume a swimming pool 75 ft long, 30 ft wide with an average depth of 6 ft. If the water is to be heated from a temperature of 50 to 65 F, what capacity heater and steam consumption is required with a turnover ratio of two times per day?

Pool volume: $75 \times 30 \times 6 \times 7.5 = 100,000 \text{ gal}$.

With a turnover ratio of twice in 24 hr, the heating capacity is: $\frac{100,000}{2 \times 24} = 2080 \text{ gal per hour}$.

The steam consumption would be: $\frac{2080 \times 8.33 (65 - 50)}{970} = 268 \text{ lb steam per hour}$.

Regulation of swimming pool temperatures is essential for successful operation and economy. It is therefore recommended that the steam supply to the heater be provided with a bypass which may be used for pool filling and initial heating and that a smaller bypass be installed with an automatic control valve having the capacity to heat the circulation water approximately 5 F per hour.

TABLE 14. HOT WATER CONSUMPTION IN VARIOUS TYPES OF BUILDINGS FOR DIFFERENT PURPOSES

TYPE OF BUILDING	CONDITIONS	GALLONS
Hotels	Room with basin only	10 (per day)
	Room with bath	40 (per day)
	(Transient)	40 (per day)
	(Men)	60 (per day)
	(Mixed)	80 (per day)
Public Buildings	(Women)	80 (per day)
	Two-room suite and bath	80 (per day)
	Three-room suite and bath	100 (per day)
Public Buildings	Public bath or lavatory	150 (per day per fixture)
	Public shower	200 (per day per fixture)
	Public lavatory with attendant	200 (per day per fixture)
Industrial Buildings	Per office employee	2 (per day)
	Per factory employee	5 (per day)
	Cleaning floors	3 (per 1000 sq ft per day)
Restaurants	\$0.50 Meals	0.5 (per customer with hand washing)
		1.0 (per customer with machine washing)
	\$1.00 Meals	1.0 (per customer with hand washing)
		2.0 (per customer with machine washing)
	\$1.50 Meals	1.5 (per customer with hand washing)
		4.0 (per customer with machine washing)

PROBLEMS IN PRACTICE

1 • The heating capacity of an indirect water heater is 100 gal per hour, using steam at 215 F and raising the water from a temperature of 50 to 150 F. Determine the heating capacity of the same water heater using water at a temperature of 180 F for the heating medium.

Using Equation 1, and because the surface area of the water heater is the same for each condition, the two conditions may be equated as follows:

$$\frac{100 \times 8.33 (150 - 50)}{240 \left[215 - \frac{(150 + 50)}{2} \right]} = \frac{Q \times 8.33 (150 - 50)}{100 \left[180 - \frac{(150 + 50)}{2} \right]}$$

$Q = 28.98$ gal per hour, capacity of heater using water at a temperature of 180 F.

2 • Why is it impractical to size water supply piping so pipe friction will produce an equal pressure on each fixture?

Because the friction would be built up only in periods of maximum flow and at all other times it would be only a fraction of that required.

3 • What is the purpose of zoning water supply systems in tall buildings?

To avoid excessive pressures in the lower stories.

4 • Define the maximum possible flow, the maximum probable flow, and the average probable flow.

The maximum possible flow is the flow which would occur if all of the outlets on the system were opened at one and the same time. The maximum probable flow is the flow which will occur with probable peak conditions. The average probable flow is the flow likely to occur under a normal condition of use.

5 • What is the factor of usage?

This is the percentage of the maximum possible flow which is likely to occur at peak load.

6 • How many feet higher than the uppermost fixtures must the water line in a house tank be to provide about 15 lb per square inch pressure at the fixture outlet?

Allowing for pipe losses, about 45 ft.

7 • What methods of hot water circulation commonly are employed with hot water supply systems?

- Upfeed risers with returns having no connections paralleling the risers.
- Upfeed risers with returns in other locations, and with connections taken off both supply and return.
- One main upfeed riser, without connections, supplying all downfeed risers for all fixtures.

8 • Which method of hot water supply generally is the most satisfactory?

The single main upfeed riser supplying drop risers for all fixtures.

9 • How much of the water stored in a hot water storage tank really is available for use?

About 75 per cent, because when only 25 per cent of the original water remains in the tank it has been so cooled down by the entering water that it is too cold for satisfactory use.

10 • In cases of intermittent demand, does a large hot water storage tank increase or decrease the steam load for water heating?

It decreases the steam load in cases of intermittent demand but causes no change in the steam load if the demand is constant.

Chapter 36

INSULATION OF PIPING

Heat Losses from Bare Pipes, Steam and Hot Water Lines, Low Temperature Pipe Insulation, Pipe Sweating, Heat Losses from Pipe Surfaces, Thickness of Pipe Insulation, Underground Insulation

PIPE insulation performs an important function in preventing loss of heat where steam or hot water are conveyed from one part of a building to another, and in reducing the absorption of heat by cold pipes as well as preventing condensation on the outer surfaces.

BARE PIPE LOSSES

Heat losses from horizontal bare iron pipes, based on data obtained from tests conducted at the Mellon Institute, are given in Table 1. These losses are expressed in Btu per hour per linear foot of pipe per degree Fahrenheit difference in temperature between the steam or hot water in the pipe and the air surrounding the pipe. The monetary value of the loss of heat given in Table 1 may be obtained by means of Fig. 1 for various heating system efficiencies, temperature differences, and calorific values and costs of coal. To solve a problem, select the proper heat loss coefficient from Table 1 and locate this value on the upper left hand margin of the chart. Then draw lines in the order indicated by the dotted lines, the dollar value of the heat loss per 100 linear feet of pipe per 1000 hours being given on the upper right hand scale. In using this chart, the cost of coal should also include the labor for handling it, boiler room expense, etc.

Heat losses from horizontal copper pipes based on tests at the A.S.H.V.E. Laboratory, are given in Table 2¹.

In order to determine heat losses per linear foot of pipe from known losses per square foot, it is necessary to know the area in square feet per linear foot of pipe. Table 3 gives these areas for various standard pipe sizes while Table 4 gives the area in square feet for flanges and fittings for various standard pipe sizes.

Very often, even where pipes are thoroughly insulated, flanges and fittings are left bare due to the belief that the losses from these parts are not large. However, 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

¹Heat Emission from Iron and Copper Pipe, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

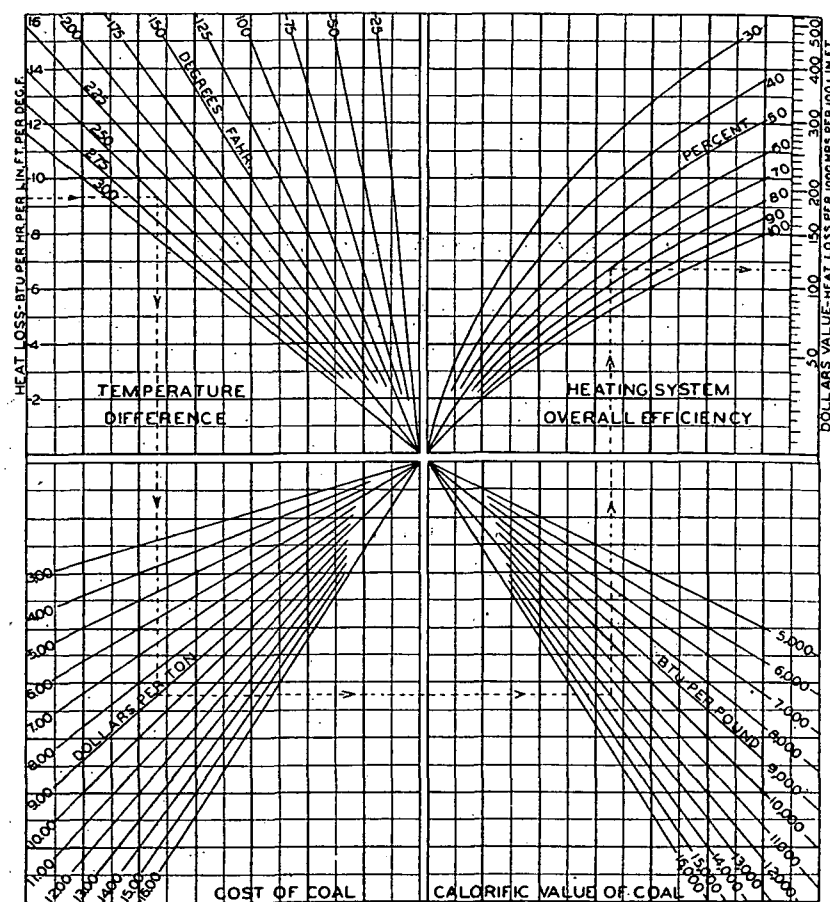


FIG. 1. CHART FOR ESTIMATING DOLLAR VALUE OF HEAT LOSS FROM BARE IRON PIPES. (SEE TABLE 1)^a

^aThis chart is based on 100 linear feet per 1000 hours. For fractions or multiples of these factors, multiply by proper percentage.

heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces.

STEAM AND HOT WATER LINES

The conductivities of various materials used for insulating steam and hot water pipes are given in Table 5. In this table the conductivities are given as functions of the mean temperatures or the mean of the inner and

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE IRON PIPES
Expressed in Btu per linear foot per degree Fahrenheit difference in temperature 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)	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
1/2	0.543	0.573	0.605	0.638	0.656	0.742	0.796
3/4	0.660	0.690	0.729	0.762	0.781	0.886	0.955
1	0.791	0.829	0.878	0.920	0.953	1.084	1.166
1 1/4	0.979	1.02	1.087	1.15	1.184	1.345	1.450
1 1/2	1.09	1.15	1.220	1.29	1.335	1.520	1.640
2	1.34	1.40	1.491	1.58	1.637	1.866	2.015
2 1/2	1.58	1.67	1.778	1.87	1.937	2.215	2.388
3	1.88	1.99	2.100	2.22	2.301	2.641	2.853
3 1/2	2.13	2.24	2.380	2.51	2.585	2.972	3.215
4	2.36	2.50	2.650	2.78	2.873	3.312	3.582
4 1/2	2.60	2.75	2.920	3.08	3.170	3.655	3.956
5	2.87	3.02	3.200	3.38	3.493	4.030	4.368
6	3.39	3.56	3.775	4.01	4.115	4.755	5.153
8	4.32	4.55	5.050	5.14	5.270	6.120	6.635
10	5.32	5.61	5.925	6.34	6.551	7.592	8.245
12	6.25	6.62	6.995	7.46	7.670	8.900	9.670

TABLE 2. HEAT LOSSES FROM HORIZONTAL BARE COPPER PIPES
Expressed in Btu per linear foot per degree Fahrenheit difference in temperature between the pipe at 210 F and surrounding still air at 70 F

NOMINAL PIPE SIZE (INCHES)	OUTSIDE DIAMETER (INCHES)	TEMPERATURE DIFFERENCE 140 F	NOMINAL PIPE SIZE (INCHES)	OUTSIDE DIAMETER (INCHES)	TEMPERATURE DIFFERENCE 140 F
1/2	0.625	0.405	2 1/2	2.625	0.986
3/4	0.875	0.478	3	3.125	1.131
1	1.125	0.550	3 1/2	3.625	1.276
1 1/4	1.375	0.623	4	4.125	1.422
1 1/2	1.625	0.695	5	5.125	1.713
2	2.125	0.841	6	6.125	2.004

TABLE 3. RADIATING 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)
1/2	0.22	2	0.622	5	1.456
3/4	0.275	2 1/2	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1 1/4	0.435	3 1/2	1.047	10	2.817
1 1/2	0.498	4	1.178	12	3.338

TABLE 4. 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
1 1/4	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1 1/2	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
2 1/2	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
3 1/2	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
4 1/2	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

^aIncluding areas of accompanying flanges bolted to the fitting.

outer surface temperatures of the insulations. This method of stating conductivities makes it possible readily to calculate the heat loss through single or compound sections. It should be emphasized that the conductivities given in Table 5 for the various insulations are the average of values obtained from a number of tests made on each type of material, also that 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 six of the types of insulation given in Table 5 for 1, 1 1/2 and 2-in.-thick materials, and for temperatures commonly encountered in engineering practice can be obtained from Tables 6 to 11.

TABLE 5. CONDUCTIVITIES (*k*) OF VARIOUS TYPES OF INSULATING MATERIALS FOR MEDIUM AND HIGH TEMPERATURE PIPES^a

TYPES OF INSULATING MATERIALS	MEAN TEMPERATURE				
	100 F	200 F	300 F	400 F	500 F
85 per cent Magnesia Type.....	0.425	0.465	0.505	0.550	0.590
Corrugated Asbestos Type.....	0.530	0.650	0.770	0.890	
(4 Plies per 1 in. thick)					
Corrugated Asbestos Type.....	0.480	0.555	0.630	0.705	
(8 Plies per 1 in. thick)					
Laminated Asbestos Type.....	0.360	0.415	0.470	0.525	0.585
(30-40 Laminations per 1 in. thick)					
Laminated Asbestos Type.....	0.545	0.605	0.665	0.725	0.785
(20 Laminations per 1 in. thick)					
Rock Wool Type.....	0.350	0.410	0.470	0.530	0.590
High Temperature Type.....	0.515	0.545	0.575	0.605	0.635
(Diatomaceous Earth and Asbestos)					
Brown Asbestos Type.....	0.600	0.640	0.675	0.715	0.750
(Felted Fibre)					

^aR. H. Heilman, *Mechanical Engineering*, Vol. 46 (1924), p. 593.

TABLE 6. COEFFICIENTS OF TRANSMISSION (*U*) FOR PIPES INSULATED WITH 85 PER CENT MAGNESIA TYPE INSULATION

These coefficients are expressed in *Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F*

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		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
1	1/2	0.744	0.754	0.764	0.774	0.779	0.802	0.814
	3/4	0.672	0.681	0.689	0.697	0.701	0.721	0.731
	1	0.613	0.621	0.629	0.637	0.641	0.659	0.670
	1 1/4	0.562	0.570	0.577	0.585	0.589	0.606	0.617
	1 1/2	0.532	0.539	0.546	0.553	0.557	0.573	0.582
	2	0.500	0.506	0.512	0.519	0.523	0.538	0.547
	2 1/2	0.475	0.481	0.487	0.493	0.497	0.512	0.520
	3	0.455	0.461	0.467	0.474	0.477	0.492	0.500
	3 1/2	0.441	0.447	0.452	0.458	0.462	0.475	0.483
	4	0.429	0.435	0.441	0.446	0.449	0.463	0.471
	4 1/2	0.420	0.425	0.431	0.437	0.440	0.453	0.460
	5	0.411	0.416	0.422	0.427	0.430	0.443	0.450
1 1/2	1/2	0.617	0.625	0.633	0.642	0.646	0.665	0.676
	3/4	0.550	0.558	0.566	0.573	0.577	0.596	0.606
	1	0.496	0.503	0.511	0.518	0.522	0.540	0.549
	1 1/4	0.453	0.459	0.465	0.472	0.475	0.490	0.498
	1 1/2	0.424	0.430	0.436	0.442	0.445	0.459	0.467
	2	0.394	0.400	0.405	0.410	0.413	0.427	0.434
	2 1/2	0.371	0.376	0.382	0.386	0.389	0.401	0.408
	3	0.352	0.357	0.362	0.367	0.370	0.380	0.387
	3 1/2	0.339	0.343	0.347	0.351	0.354	0.364	0.370
	4	0.328	0.333	0.337	0.341	0.343	0.353	0.359
	4 1/2	0.320	0.324	0.328	0.332	0.334	0.343	0.350
	5	0.312	0.316	0.320	0.324	0.326	0.336	0.342
2	1/2	0.543	0.551	0.558	0.565	0.569	0.587	0.597
	3/4	0.484	0.490	0.497	0.503	0.507	0.523	0.532
	1	0.433	0.439	0.445	0.451	0.454	0.467	0.476
	1 1/4	0.393	0.398	0.403	0.409	0.412	0.424	0.432
	1 1/2	0.365	0.370	0.376	0.381	0.384	0.397	0.402
	2	0.338	0.343	0.347	0.351	0.354	0.364	0.370
	2 1/2	0.316	0.320	0.324	0.328	0.331	0.341	0.347
	3	0.297	0.301	0.305	0.309	0.312	0.321	0.326
	3 1/2	0.284	0.288	0.292	0.295	0.297	0.306	0.311
	4	0.275	0.278	0.282	0.285	0.287	0.296	0.301
	4 1/2	0.266	0.270	0.273	0.276	0.278	0.286	0.290
	5	0.258	0.262	0.265	0.268	0.270	0.278	0.283
	6	0.250	0.254	0.257	0.260	0.262	0.270	0.274
	8	0.236	0.239	0.242	0.245	0.247	0.255	0.258
	10	0.224	0.227	0.230	0.233	0.235	0.242	0.246
	12	0.219	0.222	0.225	0.228	0.230	0.237	0.240

TABLE 7. COEFFICIENTS OF TRANSMISSION (*U*) FOR PIPES INSULATED WITH CORRUGATED ASBESTOS TYPE INSULATION (4 PLIES PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		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
1	1/2	0.890	0.919	0.949	0.978	0.995	1.065	1.106
	3/4	0.803	0.829	0.857	0.883	0.898	0.961	0.997
	1	0.731	0.756	0.780	0.804	0.818	0.876	0.909
	1 1/4	0.671	0.693	0.716	0.738	0.751	0.804	0.834
	1 1/2	0.635	0.656	0.677	0.698	0.710	0.760	0.788
	2	0.595	0.615	0.635	0.656	0.667	0.715	0.742
	2 1/2	0.567	0.586	0.605	0.624	0.635	0.680	0.705
	3	0.544	0.562	0.580	0.598	0.608	0.652	0.677
	3 1/2	0.527	0.544	0.561	0.578	0.588	0.631	0.654
	4	0.513	0.530	0.548	0.565	0.575	0.616	0.639
	4 1/2	0.502	0.518	0.535	0.551	0.561	0.601	0.624
	5	0.490	0.507	0.523	0.539	0.549	0.588	0.611
1 1/2	6	0.480	0.496	0.512	0.528	0.538	0.577	0.599
	8	0.462	0.477	0.493	0.508	0.517	0.554	0.575
	10	0.447	0.462	0.476	0.491	0.500	0.537	0.557
	12	0.441	0.456	0.470	0.485	0.493	0.529	0.550
	1/2	0.737	0.762	0.787	0.812	0.826	0.884	0.918
	3/4	0.657	0.679	0.702	0.725	0.737	0.790	0.820
	1	0.594	0.614	0.634	0.654	0.666	0.713	0.740
	1 1/4	0.542	0.559	0.577	0.596	0.606	0.649	0.673
	1 1/2	0.507	0.524	0.541	0.558	0.568	0.609	0.632
	2	0.471	0.487	0.503	0.519	0.528	0.565	0.587
	2 1/2	0.443	0.458	0.473	0.488	0.497	0.533	0.553
	3	0.421	0.435	0.449	0.463	0.472	0.506	0.525
2	3 1/2	0.403	0.417	0.430	0.443	0.451	0.483	0.502
	4	0.393	0.405	0.418	0.432	0.439	0.471	0.489
	4 1/2	0.383	0.394	0.407	0.420	0.428	0.460	0.476
	5	0.372	0.384	0.397	0.409	0.417	0.447	0.463
	6	0.362	0.374	0.387	0.399	0.406	0.436	0.452
	8	0.343	0.354	0.366	0.378	0.385	0.413	0.429
	10	0.328	0.339	0.351	0.362	0.369	0.397	0.413
	12	0.323	0.334	0.346	0.357	0.364	0.391	0.407
	1/2	0.648	0.670	0.692	0.713	0.726	0.779	0.810
	3/4	0.578	0.598	0.617	0.637	0.648	0.694	0.720
	1	0.518	0.535	0.552	0.570	0.580	0.622	0.645
	1 1/4	0.469	0.485	0.501	0.517	0.527	0.566	0.587
2	1 1/2	0.438	0.452	0.467	0.481	0.490	0.526	0.545
	2	0.404	0.417	0.430	0.444	0.452	0.483	0.502
	2 1/2	0.379	0.391	0.403	0.415	0.422	0.451	0.466
	3	0.356	0.367	0.378	0.390	0.397	0.425	0.440
	3 1/2	0.339	0.350	0.361	0.373	0.380	0.406	0.421
	4	0.328	0.339	0.350	0.360	0.367	0.392	0.406
	4 1/2	0.318	0.328	0.339	0.350	0.357	0.381	0.395
	5	0.308	0.318	0.329	0.340	0.346	0.370	0.384
	6	0.299	0.309	0.319	0.329	0.335	0.358	0.371
	8	0.282	0.291	0.301	0.310	0.315	0.336	0.349
	10	0.267	0.276	0.285	0.294	0.299	0.319	0.332
	12	0.263	0.272	0.280	0.289	0.294	0.314	0.325

TABLE 8. COEFFICIENTS OF TRANSMISSION (*U*) FOR PIPES INSULATED WITH CORRUGATED ASBESTOS TYPE INSULATION (8 PLIES PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		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
1	1/2	0.801	0.820	0.838	0.857	0.868	0.913	0.939
	3/4	0.723	0.739	0.756	0.773	0.783	0.824	0.847
	1	0.658	0.673	0.688	0.704	0.713	0.751	0.772
	1 1/4	0.606	0.619	0.633	0.647	0.655	0.688	0.707
	1 1/2	0.573	0.586	0.599	0.612	0.619	0.652	0.670
	2	0.538	0.550	0.562	0.575	0.581	0.612	0.629
	2 1/2	0.511	0.523	0.534	0.546	0.553	0.582	0.599
	3	0.489	0.501	0.512	0.524	0.531	0.558	0.575
	3 1/2	0.474	0.485	0.496	0.507	0.514	0.542	0.557
	4	0.461	0.472	0.482	0.493	0.500	0.527	0.542
	4 1/2	0.451	0.462	0.472	0.482	0.489	0.515	0.530
	5	0.442	0.452	0.462	0.473	0.479	0.505	0.520
1 1/2	6	0.432	0.442	0.452	0.463	0.468	0.493	0.508
	8	0.416	0.426	0.436	0.446	0.451	0.475	0.489
	10	0.402	0.412	0.421	0.430	0.435	0.459	0.473
	12	0.397	0.406	0.415	0.424	0.429	0.452	0.466
	1/2	0.664	0.679	0.695	0.711	0.720	0.759	0.780
	3/4	0.593	0.607	0.621	0.636	0.643	0.677	0.697
	1	0.535	0.547	0.560	0.573	0.580	0.611	0.629
	1 1/4	0.488	0.499	0.510	0.522	0.528	0.556	0.572
	1 1/2	0.457	0.467	0.478	0.490	0.496	0.522	0.537
	2	0.425	0.434	0.444	0.455	0.460	0.485	0.499
	2 1/2	0.399	0.408	0.418	0.428	0.434	0.457	0.471
	3	0.378	0.387	0.396	0.405	0.411	0.433	0.446
2	3 1/2	0.363	0.371	0.380	0.388	0.393	0.415	0.427
	4	0.353	0.361	0.369	0.378	0.383	0.403	0.415
	4 1/2	0.343	0.351	0.360	0.368	0.373	0.393	0.404
	5	0.334	0.342	0.350	0.358	0.363	0.383	0.394
	6	0.325	0.333	0.341	0.349	0.353	0.373	0.383
	8	0.309	0.316	0.324	0.332	0.336	0.355	0.365
	10	0.295	0.303	0.310	0.318	0.322	0.340	0.350
	12	0.291	0.298	0.306	0.313	0.317	0.335	0.344
	1/2	0.585	0.599	0.613	0.627	0.635	0.668	0.688
	3/4	0.520	0.533	0.545	0.558	0.565	0.595	0.612
	1	0.465	0.476	0.487	0.498	0.504	0.532	0.547
	1 1/4	0.422	0.432	0.442	0.452	0.458	0.483	0.497
2	1 1/2	0.394	0.403	0.412	0.422	0.427	0.450	0.462
	2	0.364	0.372	0.380	0.388	0.393	0.415	0.427
	2 1/2	0.339	0.347	0.355	0.363	0.367	0.387	0.398
	3	0.319	0.327	0.334	0.342	0.346	0.365	0.375
	3 1/2	0.304	0.311	0.318	0.326	0.330	0.349	0.358
	4	0.295	0.302	0.308	0.315	0.319	0.336	0.345
	4 1/2	0.285	0.292	0.299	0.306	0.310	0.327	0.336
	5	0.278	0.284	0.290	0.297	0.301	0.317	0.326
	6	0.269	0.275	0.282	0.288	0.292	0.307	0.315
	8	0.253	0.259	0.265	0.270	0.273	0.288	0.296
	10	0.240	0.245	0.251	0.257	0.260	0.275	0.282
	12	0.236	0.241	0.247	0.253	0.256	0.270	0.277

TABLE 9. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH LAMINATED ASBESTOS TYPE INSULATION (30 TO 40 LAMINATIONS PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		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
1	1/8	0.605	0.620	0.635	0.650	0.658	0.695	0.716
	3/16	0.546	0.560	0.573	0.586	0.594	0.627	0.645
	1/4	0.498	0.510	0.522	0.534	0.541	0.570	0.587
	5/16	0.457	0.468	0.480	0.491	0.497	0.525	0.540
	3/8	0.432	0.442	0.453	0.464	0.470	0.496	0.511
	1/2	0.406	0.416	0.426	0.437	0.442	0.467	0.481
	3/4	0.385	0.395	0.405	0.415	0.420	0.443	0.457
	1	0.370	0.379	0.389	0.398	0.403	0.425	0.438
	1 1/8	0.359	0.367	0.376	0.385	0.390	0.413	0.426
	1 1/4	0.349	0.358	0.366	0.375	0.380	0.402	0.414
	1 1/2	0.341	0.350	0.359	0.367	0.372	0.393	0.405
	1 3/4	0.334	0.342	0.351	0.359	0.364	0.384	0.395
1 1/2	2	0.327	0.335	0.343	0.351	0.356	0.376	0.387
	2 1/8	0.314	0.322	0.330	0.338	0.343	0.362	0.373
	2 1/4	0.304	0.312	0.320	0.328	0.332	0.350	0.361
	2 1/2	0.301	0.308	0.316	0.324	0.328	0.346	0.356
	2 3/8	0.502	0.514	0.526	0.539	0.546	0.577	0.595
	2 3/4	0.450	0.461	0.473	0.484	0.490	0.517	0.532
	3	0.405	0.415	0.426	0.436	0.442	0.466	0.480
	3 1/8	0.369	0.378	0.387	0.396	0.401	0.423	0.435
	3 1/4	0.343	0.352	0.361	0.370	0.375	0.397	0.409
	3 1/2	0.321	0.329	0.337	0.345	0.350	0.369	0.380
	3 3/8	0.301	0.309	0.317	0.324	0.330	0.348	0.358
	3 3/4	0.286	0.293	0.301	0.308	0.313	0.330	0.340
2	4	0.274	0.281	0.288	0.295	0.300	0.316	0.326
	4 1/8	0.267	0.273	0.280	0.287	0.291	0.307	0.317
	4 1/4	0.259	0.266	0.272	0.279	0.283	0.299	0.308
	4 1/2	0.253	0.260	0.266	0.272	0.276	0.291	0.300
	4 3/8	0.247	0.253	0.260	0.266	0.269	0.284	0.293
	4 3/4	0.234	0.240	0.246	0.252	0.255	0.270	0.279
	5	0.223	0.229	0.235	0.241	0.245	0.258	0.266
	5 1/8	0.221	0.227	0.232	0.238	0.241	0.255	0.263
	5 1/4	0.442	0.453	0.464	0.475	0.481	0.508	0.523
	5 1/2	0.392	0.402	0.412	0.422	0.428	0.452	0.465
	5 3/8	0.352	0.360	0.369	0.378	0.383	0.405	0.417
	5 3/4	0.319	0.327	0.335	0.343	0.348	0.367	0.379
	6	0.297	0.304	0.311	0.319	0.323	0.341	0.352
	6 1/8	0.274	0.280	0.287	0.294	0.298	0.314	0.324
	6 1/4	0.256	0.262	0.269	0.275	0.279	0.293	0.302
	6 1/2	0.243	0.249	0.254	0.260	0.264	0.277	0.285
	6 3/8	0.231	0.236	0.242	0.248	0.251	0.265	0.273
	6 3/4	0.223	0.228	0.234	0.240	0.243	0.257	0.265
	7	0.216	0.222	0.227	0.233	0.236	0.249	0.256
	7 1/8	0.210	0.215	0.220	0.225	0.228	0.241	0.248
	7 1/4	0.203	0.208	0.213	0.218	0.221	0.233	0.240
	7 1/2	0.191	0.196	0.201	0.206	0.209	0.220	0.227
	7 3/8	0.182	0.187	0.192	0.196	0.199	0.210	0.215
	7 3/4	0.178	0.183	0.187	0.192	0.195	0.205	0.210

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH LAMINATED ASBESTOS TYPE INSULATION (APPROXIMATELY 20 LAMINATIONS PER INCH THICKNESS)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		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
1	1/8	0.910	0.925	0.940	0.956	0.964	1.001	1.022
	3/16	0.823	0.836	0.850	0.863	0.871	0.902	0.921
	1/4	0.748	0.760	0.773	0.785	0.792	0.823	0.840
	5/16	0.686	0.698	0.710	0.721	0.728	0.756	0.771
	3/8	0.649	0.659	0.671	0.682	0.688	0.716	0.731
	1/2	0.610	0.620	0.630	0.640	0.647	0.671	0.685
	3/4	0.581	0.590	0.600	0.609	0.615	0.638	0.651
	1	0.558	0.567	0.576	0.585	0.591	0.613	0.626
	1 1/8	0.539	0.548	0.557	0.566	0.571	0.592	0.604
	1 1/4	0.524	0.532	0.541	0.551	0.556	0.577	0.589
	1 1/2	0.514	0.522	0.530	0.539	0.544	0.564	0.575
	1 3/4	0.503	0.511	0.519	0.528	0.533	0.553	0.565
1 1/2	2	0.492	0.500	0.509	0.517	0.522	0.542	0.553
	2 1/8	0.473	0.480	0.488	0.497	0.502	0.521	0.532
	2 1/4	0.458	0.465	0.473	0.481	0.485	0.504	0.514
	2 1/2	0.452	0.459	0.467	0.475	0.478	0.497	0.507
	2 3/8	0.755	0.767	0.780	0.793	0.800	0.831	0.848
	2 3/4	0.674	0.685	0.697	0.708	0.715	0.743	0.759
	3	0.607	0.618	0.628	0.639	0.645	0.670	0.684
	3 1/8	0.553	0.562	0.572	0.581	0.587	0.610	0.622
	3 1/4	0.517	0.527	0.536	0.545	0.550	0.572	0.584
	3 1/2	0.481	0.490	0.499	0.508	0.513	0.535	0.547
	3 3/8	0.453	0.460	0.469	0.477	0.481	0.500	0.511
	3 3/4	0.429	0.436	0.444	0.452	0.456	0.475	0.485
2	4	0.412	0.419	0.427	0.434	0.438	0.456	0.465
	4 1/8	0.400	0.407	0.415	0.422	0.426	0.443	0.453
	4 1/4	0.390	0.396	0.402	0.409	0.413	0.429	0.437
	4 1/2	0.380	0.386	0.393	0.400	0.403	0.418	0.427
	4 3/8	0.369	0.375	0.382	0.389	0.392	0.408	0.417
	4 3/4	0.351	0.358	0.364	0.370	0.374	0.388	0.397
	5	0.337	0.344	0.350	0.356	0.359	0.373	0.382
	5 1/8	0.332	0.338	0.344	0.350	0.353	0.367	0.375
	5 1/4	0.664	0.675	0.687	0.698	0.704	0.732	0.747
	5 1/2	0.591	0.601	0.611	0.621	0.627	0.652	0.665
	5 3/8	0.529	0.538	0.547	0.557	0.562	0.584	0.597
	5 3/4	0.480	0.488	0.497	0.505	0.510	0.529	0.540
	6	0.445	0.453	0.462	0.470	0.475	0.494	0.504
	6 1/8	0.412	0.420	0.427	0.434	0.438	0.455	0.464
	6 1/4	0.385	0.392	0.398	0.405	0.409	0.425	0.434
	6 1/2	0.364	0.370	0.376	0.382	0.385	0.400	0.408
	6 3/8	0.346	0.352	0.358	0.365	0.368	0.382	0.390
	6 3/4	0.336	0.342	0.348	0.354	0.357	0.371	0.378
	7	0.325	0.332	0.338	0.343	0.346	0.360	0.367
	7 1/8	0.316	0.322	0.327	0.333	0.336	0.349	0.356
	7 1/4	0.306	0.312	0.317	0.323	0.326	0.338	0.345
	7 1/2	0.288	0.293	0.298	0.303	0.306	0.317	0.324
	7 3/8	0.275	0.279	0.284	0.289	0.292	0.302	0.308
	7 3/4	0.269	0.274	0.278	0.283	0.286	0.296	0.302

TABLE 11. COEFFICIENTS OF TRANSMISSION (*U*) FOR PIPES INSULATED WITH ROCK WOOL TYPE INSULATION

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

THICKNESS OF INSULATION (INCHES)	NOMINAL PIPE SIZE (INCHES)	HOT WATER				STEAM		
		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
1	1/2	0.631	0.644	0.658	0.672	0.680	0.712	0.730
	3/4	0.569	0.581	0.593	0.606	0.613	0.642	0.659
	1	0.518	0.529	0.541	0.552	0.559	0.585	0.600
	1 1/4	0.476	0.486	0.497	0.507	0.513	0.537	0.551
	1 1/2	0.450	0.460	0.470	0.480	0.485	0.508	0.522
	2	0.422	0.431	0.441	0.450	0.456	0.478	0.490
	2 1/2	0.402	0.411	0.420	0.428	0.434	0.455	0.466
	3	0.385	0.394	0.402	0.411	0.415	0.435	0.446
	3 1/2	0.373	0.381	0.389	0.398	0.402	0.421	0.432
	4	0.363	0.371	0.379	0.387	0.392	0.411	0.422
	4 1/2	0.355	0.363	0.371	0.379	0.383	0.402	0.413
	5	0.348	0.356	0.364	0.371	0.376	0.394	0.404
1 1/2	6	0.341	0.348	0.356	0.363	0.368	0.386	0.396
	8	0.327	0.335	0.342	0.349	0.353	0.372	0.381
	10	0.317	0.324	0.331	0.338	0.343	0.360	0.369
	12	0.313	0.320	0.327	0.334	0.338	0.355	0.364
	1/2	0.523	0.534	0.545	0.556	0.563	0.590	0.606
	3/4	0.468	0.477	0.487	0.497	0.503	0.528	0.542
	1	0.421	0.430	0.440	0.449	0.455	0.477	0.490
	1 1/4	0.383	0.391	0.399	0.407	0.412	0.433	0.444
	1 1/2	0.359	0.366	0.375	0.383	0.387	0.407	0.419
	2	0.333	0.340	0.348	0.356	0.360	0.378	0.389
	2 1/2	0.314	0.320	0.327	0.335	0.339	0.355	0.365
	3	0.296	0.302	0.310	0.317	0.321	0.337	0.347
2	3 1/2	0.286	0.291	0.298	0.304	0.307	0.323	0.332
	4	0.278	0.284	0.290	0.296	0.300	0.315	0.323
	4 1/2	0.270	0.276	0.282	0.287	0.291	0.305	0.313
	5	0.263	0.269	0.275	0.280	0.284	0.298	0.305
	6	0.257	0.262	0.267	0.273	0.277	0.290	0.297
	8	0.244	0.249	0.254	0.260	0.263	0.276	0.283
	10	0.235	0.240	0.245	0.250	0.253	0.265	0.272
	12	0.230	0.234	0.239	0.245	0.247	0.260	0.267
	1/2	0.461	0.471	0.481	0.491	0.496	0.520	0.534
	3/4	0.409	0.418	0.427	0.436	0.441	0.463	0.475
	1	0.366	0.374	0.382	0.390	0.395	0.415	0.427
	1 1/4	0.333	0.340	0.347	0.355	0.359	0.377	0.387
2 1/2	1 1/2	0.310	0.316	0.323	0.330	0.334	0.351	0.360
	2	0.286	0.292	0.298	0.304	0.308	0.323	0.331
	2 1/2	0.268	0.274	0.279	0.285	0.289	0.302	0.310
	3	0.252	0.257	0.262	0.268	0.272	0.284	0.292
	3 1/2	0.241	0.246	0.251	0.257	0.260	0.272	0.280
	4	0.232	0.237	0.242	0.247	0.250	0.262	0.269
	4 1/2	0.225	0.230	0.235	0.240	0.243	0.255	0.262
	5	0.218	0.223	0.228	0.233	0.236	0.247	0.253
	6	0.213	0.217	0.221	0.226	0.228	0.239	0.245
	8	0.200	0.204	0.208	0.213	0.215	0.225	0.231
	10	0.189	0.193	0.197	0.201	0.204	0.214	0.220
	12	0.185	0.190	0.194	0.198	0.200	0.210	0.216

inclusive. The loss through other thicknesses of the materials, and for other hot water or steam temperature conditions may be obtained by interpolation. The heat loss coefficients given in Tables 6 to 11 are based on the conductivities in Table 5 and were computed from data given in Chapter 22, THE GUIDE 1931.

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at low temperatures should be insulated so as to retard the flow of heat from the outside into the low temperature area and to prevent the formation of condensation and of frost if the temperatures are low enough, as well as to prevent corrosion induced by the presence of condensed moisture on metal surfaces. Materials commonly used for insulating pipes and surfaces at low temperatures are cork, rock cork, hair felt and other felted or fibrous non-absorbent materials. Thermal conductivities of low temperature insulating materials are given in Chapter 5.

Insulating materials are available commercially to meet varying temperature gradients. For example, the thickness of insulation for ice water is approximately 1 1/2 in. if the temperature in the line is not lower than 25 F; the thickness of insulation for brine is approximately 2 1/2 in. where the temperature ranges from 0 deg to 25 F; and the thickness of insulation where the brine temperature ranges from -30 F to zero degrees is approximately 4 in.

Insulation 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 12 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 the most efficient insulation be utilized. This table is based on the use of hair felt or cork, having a conductivity of 0.30. The initial water temperature is assumed to be 10 deg above, and the surrounding air temperature 50 deg below the freezing point of water (temperature difference, 60 F).

The last column of Table 12 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 12. 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

TABLE 12. DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES

NOMINAL PIPE SIZE (INCHES)	NUMBER OF HOURS TO COOL WATER TO FREEZING POINT			WATER REQUIRED TO FLOW TO PREVENT FREEZING, POUNDS PER LINEAR FOOT OF PIPE PER HOUR		
	Thickness of Insulation in Inches					
	1	2	3	1	2	3
1/2	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
1 1/2	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	1.44
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.46	1.93

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 12, 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 12, 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.

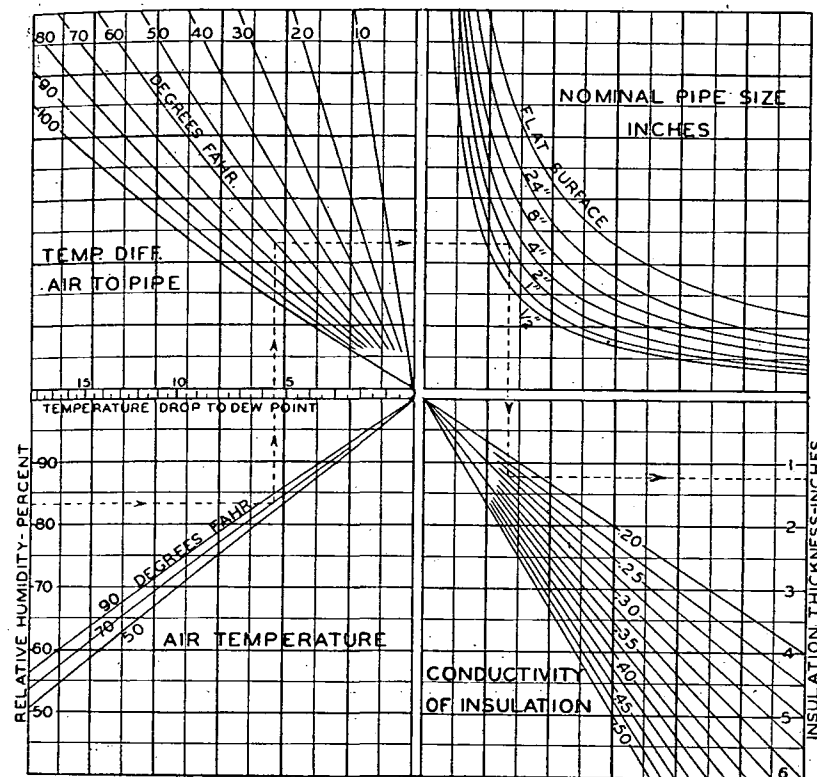


FIG. 2. THICKNESS OF PIPE INSULATION TO PREVENT SWEATING^a

^aSolve problems by drawing lines as indicated by dotted line, entering chart at lower left hand scale.

Pipe Sweating

In some cases the prevention of condensation rather than the conservation of heat is the governing factor in determining the thickness of insulation required. Fig. 2 may be used for determining the thickness of any material of known conductivity which should be used to prevent condensation on pipes and flat metallic surfaces. The surface resistances used for calculating the family of curves in Fig. 2 are based on the results of tests made on canvas-covered pipe insulation surfaces at Mellon Institute.

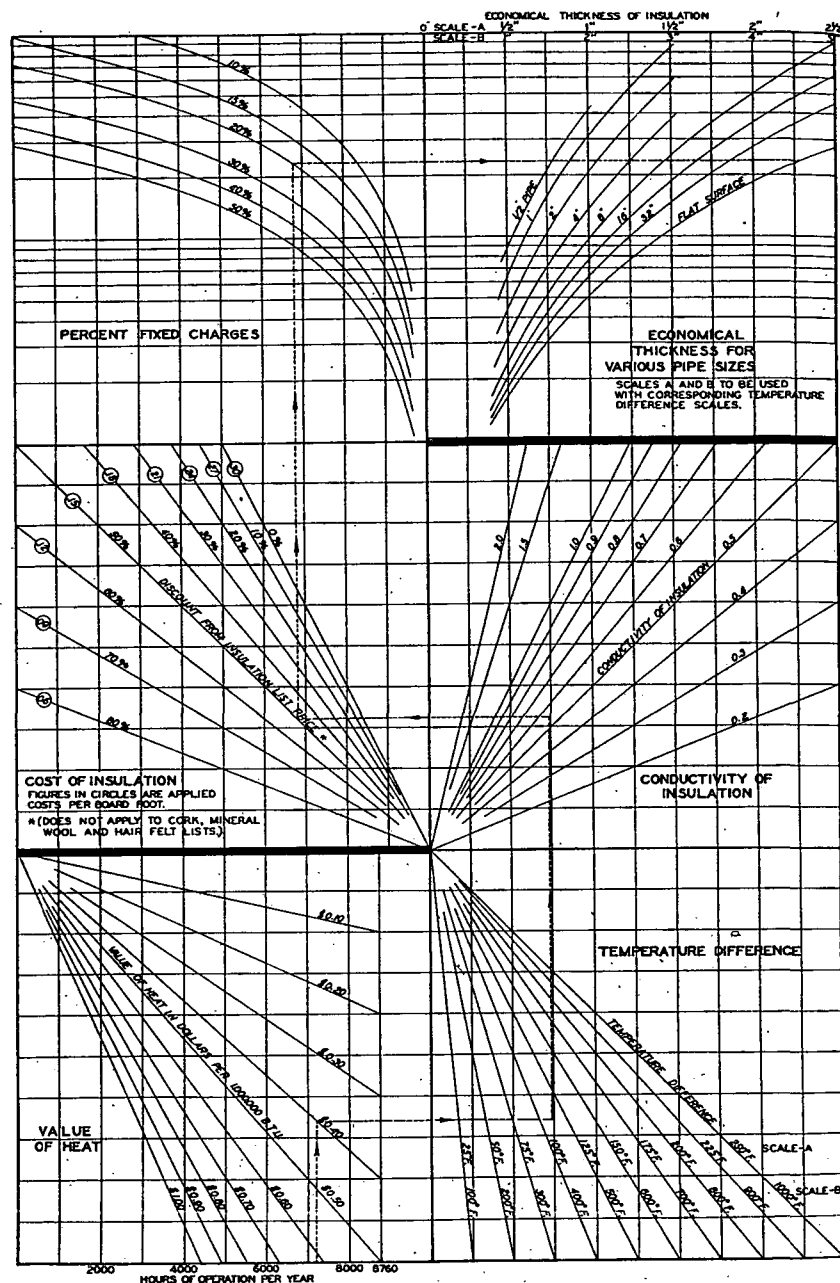


FIG. 3. CHART FOR DETERMINING ECONOMICAL THICKNESS OF INSULATION
(L. B. McMillan, *Proc. National Dist. Heating Ass'n.*, Vol. 18, p. 131.)

However, it has been found that the resistance for asphaltic and roofing surfaces is practically the same as for canvas surfaces, so that the curves given may be followed with no alteration for surfaces commonly used.

Moisture will be deposited on a surface whenever its temperature falls to that of the dew point. The maximum permissible temperature drop is indicated on Fig. 2 at the point where the guide line passes through the horizontal scale at the left center of the chart. This temperature drop represents the difference between the dry-bulb temperature and the dew-point temperature for the conditions involved. (See discussion of condensation in Chapter 7.)

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 shown for bare surfaces, because of the fact that air flowing over the surface of the insulation can increase only the rate of heat transfer from surface to air, and cannot change the internal resistance to heat flow inherent in the insulation itself. The maximum increase in loss due to air velocity ranges from about 30 per cent in the case of 1-in. thick insulation, to about 10 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 sealed as tightly as possible. Pipe insulation out-of-doors should be provided with a waterproof jacket, and other outdoor insulation should be thoroughly weatherproofed.

ECONOMICAL THICKNESS OF PIPE INSULATION

Table 13 shows the thicknesses of insulation which ordinarily are used for various temperature conditions. Where a thorough analysis of economic thickness is desired, this may be accomplished through the use of the chart, Fig. 3.

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 Insulation

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes. (See Chapter 37.) Detailed

TABLE 13. THICKNESSES OF INSULATION ORDINARILY USED INDOORS*

STEAM PRESSURES (LB GAGE) OR CONDITIONS	STEAM TEMPERATURES DEGREES FAHRENHEIT	THICKNESS OF INSULATION		
		Pipes Larger Than 4 in.	Pipes 2 in. to 4 in.	Pipes 1½ in. to 1¼ 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.

*All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ¼ in. greater than shown in this table, and covered with a waterproof jacket.

TABLE 14. THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

STEAM PRESSURES (LB GAGE) OR CONDITIONS	STEAM TEMPERATURES DEGREES FAHRENHEIT	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

data 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. 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 of accurate determination due to the many variables which have to be considered. As a result of theories developed by J. R. Allen³, together with experimental data presented by others, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table

14 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. 3. Fig. 3 is 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.

PROBLEMS IN PRACTICE

1 • What precautions must be taken in selecting insulation used for covering pipe lines carrying materials at temperatures lower than the dew point?

Materials intended for this service should be as moisture proof as possible and in addition an outer covering should be applied which is proof against diffusion of air and water vapor. If the material permits the diffusion of air, the air will reach a point in the covering where the temperature is below the dew point. The condensed water will gradually accumulate until the covering becomes saturated, which will increase the conductivity and perhaps lower the mechanical strength of the covering until it becomes worthless.

2 • Compute the total annual heat loss from 165 ft of 2-in. bare pipe in service 4000 hours per year. The pipe is carrying steam at 10 lb pressure and is exposed to an average air temperature of 70 F.

The pipe temperature is taken as the steam temperature, which is 239.4 F, obtained from Table 6, Chapter 1. The temperature difference between the pipe and air = $239.4 - 70 = 169.4$ deg. By interpolation of Table 1 between temperature differences of 157.1 F and 227.7 F, the heat loss from a 2-in. pipe at a temperature difference of 169.4 deg is found to be 1.677 Btu per hour per linear foot per degree temperature difference. The total annual heat loss from the entire = $1.677 \times 169.4 \times 165$ (linear feet) $\times 4000$ (hours) = 188,000,000 Btu.

3 • 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 Question 1. 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.

The cost of heat per 1 million Btu supplied to the system = $1,000,000 \times 11.5$ (dollars) $\div 13,000$ (Btu) $\times 2000$ (lb) $\times 0.55$ (efficiency) = \$0.804. The total cost of heat lost per year = 0.804×188 (million Btu) = \$151.15.

4 • If the steam line given in Question 1 is covered with 1-in. thick 85 per cent magnesia, determine the resulting total annual heat 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.

By interpolation of Table 6 between temperature differences of 157.1 F and 227.7 F, the coefficient of transmission for 1-in. magnesia on a 2-in. pipe is found to be 0.525 Btu per hour per square foot of pipe surface per degree temperature difference at a temperature difference of 169.4 deg. The total hourly loss per square foot of insulated pipe will then be $0.525 \times 169.4 = 89.04$ Btu. From Table 3 the area per linear foot of 2-in. pipe is

*A closely approximate solution of this problem may be quickly made by use of the estimating chart given in Fig. 1.

¹Handbook of the National District Heating Association, Second Edition, 1932.

²Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920).

found to be 0.622 sq ft. The total annual loss through the insulation = $89.04 \times 0.622 \times 165$ (linear feet) $\times 4000$ (hours) = 36,550,000 Btu. The annual bare pipe loss as determined in the solution of Question 1 was found to be 188,000,000 Btu. The saving due to insulation is then $188,000,000 - 36,550,000 = 151,350,000$ Btu per year.

From the solution of Question 2 it was found that the heat supplied to the system cost \$0.804 per million Btu; therefore, the monetary value of the saving = 0.804 (dollars) $\times 151.35$ (million Btu) = \$121.69, or 81.2 per cent of the cost when using uninsulated pipe.

5 • The manufacturer's list price for 85 per cent magnesia insulation is \$0.36 per linear foot for 1-in. (standard thick) material to cover a 2-in. pipe. Determine the period of time required for the saving found in Question 3 to pay for the cost of the insulation if it can be purchased and applied at 80 per cent of list price (20 per cent discount).

The applied cost of insulation = 165 (linear feet) $\times 0.36$ (dollars) $\times 0.80$ (net) = 47.52. Since the annual saving as found in Question 3 amounts to \$121.69, the insulation will pay for its cost in $47.52 \div 121.69 = 0.3905$ years; in other words, the cost will be repaid 2.56 times by the saving obtained in one heating season.

6 • The conductivity of magnesia insulation is 0.455 at the mean temperature which will result under the conditions of Question 3. Estimate the most economical thickness of magnesia for application on the pipe when operating under the conditions which are given in the foregoing problems and when a 20 per cent return is required on the investment for insulation.

Use chart given in Fig. 3. Begin at the left bottom margin and proceed successively as shown by the dotted line example to the following essential data which are collected from the problems previously given:

4000 hours operation per year.
\$0.804 value of heat, dollars per million Btu.
169.4 deg temperature difference.
0.455 conductivity of insulation.
20 per cent discount from list, cost of insulation.
20 per cent fixed charges, return on investment.
2-in. pipe size.

Solution of the problem by use of Fig. 3 results in a required thickness of approximately 1.05 in. The nearest commercial thickness procurable is standard thick ($1\frac{1}{2}$ in.) magnesia.

(It is of interest to note that the use of Fig. 3 will generally result in solutions which, for all practical purposes, agree closely with the specifications for thicknesses given in Table 13.)

7 • Determine the minimum thickness of wool felt insulation having a conductivity of 0.30 necessary to prevent condensation of moisture on a 4-in. pipe carrying cold water at a temperature of 40 F when the surrounding air reaches maximum conditions of 90 F with a relative humidity of 90 per cent.

The difference between the temperature of the pipe and the surrounding air is $90 - 40 = 50$ deg. For quick estimating purposes use the chart given in Fig. 2. Enter this chart at the lower left margin on the 90 per cent relative humidity line and proceed horizontally to the right to intersect the 90 deg air temperature line. Project a line up to the 50 deg temperature difference line, and then horizontally to the right to the intersection with the 4-in. pipe size line. From this point proceed down to intersect the 0.30 line which denotes the conductivity of the insulation. Directly opposite this point of intersection the correct thickness of insulation is read from the scale on the lower right margin. This chart solution denotes that wool felt 2.4-in. thick is sufficient to prevent condensation. The nearest commercial thickness procurable is $2\frac{1}{2}$ in.

For prevention of condensation as well as for protection against freezing, if the thickness determined theoretically cannot be had, it is better to apply the next greater thickness procurable rather than to use any lesser thickness because an additional factor of safety is thus obtained.

Chapter 37

DISTRICT HEATING

Underground Steam Piping, Selection of Pipe Sizes, Provision for Expansion, Capacity of Returns with Various Grades, Pipe Conduits, Pipe Tunnels, Service Connections, Steam per Square Foot of Heating Surface, Fluid Meters and Metering, Rates

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 and for the design of heating systems for buildings 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.

UNDERGROUND STEAM 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 per 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. Any unusual requirements such as those for process steam 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, in part, upon whether or not it has been passed through electrical generating units. If it has, 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 lb per square inch) are (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, and (3) simpler problems in pressure reduction at the buildings. With distribution pressures not exceeding 40 lb per square inch 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 this 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 building heating.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 lb per square inch, 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 lb per square inch is usually demanded although 30 lb per square inch is sufficient if the mangle is equipped with a large number of rolls and if a slow rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 lb per square inch.

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 the Unwin pressure drop formula. This formula, which gives pressure drops slightly larger than actual test results, is as follows:

$$P = \frac{0.0001306 W^2 L \left(1 + \frac{3.6}{d}\right)}{y d^5} \quad (1)$$

where

P = pressure drop, pounds per square inch.

W = weight of steam flowing, pounds per minute.

L = length of pipe, feet.

d = inside diameter of pipe, inches.

y = average density of steam, pounds per cubic foot.

This formula is similar to the Babcock formula given in Chapter 32.

Information on provision for expansion will be found in Chapter 34.

In general, return lines when installed follow the contour of the land, and Table 1 gives sizes of return pipes for various grades. It is evident that at points where the grade is great, smaller pipes can be installed.

PIPE CONDUITS

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 steel set in concrete.

TABLE 1. CAPACITY OF RETURNS FOR UNDERGROUND DISTRIBUTION SYSTEMS IN POUNDS OF CONDENSATE PER HOUR

SIZE OF PIPE IN.	PITCH OF PIPE PER 100 FT.					
	6"	1'	2'	3'	5'	10'
1	448	998	1890	2240	3490	5490
1 1/4	1740	2490	3990	4880	6480	9480
1 1/2	2700	4190	5740	7480	9480	14500
2	4980	7380	10700	13900	16900	24900
3	13900	22500	30900	37400	50400	74800
4	30900	44800	64800	79700	105000	154000
5	54800	79800	120000	144800	195000	294000
6	90000	138000	187000	237000	312000	449000
8	190000	277000	404000	508000	660000	938000
10	344000	498000	724000	900000	1190000	-----
12	555000	798000	1148000	1499000	1990000	-----

*Size of pipe should be increased if it carries any steam.

In laying out conduits of this type the following points should be borne in mind:

1. An expansion joint, offset, or bend should be placed between each two anchors.
2. If the distance between buildings is 150 ft or less and the steam line contains high-pressure steam, the line may be anchored in the basement of one building and allowed to expand into the basement of the second building. If the steam line contains low-pressure steam (up to 4-lb pressure), this method may be used if buildings are 250 ft or less apart.
3. If the distance between buildings is between 150 ft and 300 ft and the steam line contains high-pressure steam, the lines should be anchored midway between the buildings and allowed to expand into the basements of both buildings. If the steam line contains low-pressure steam this method may be used if buildings are between 250 ft and 600 ft apart. No manhole is required at the anchor, and a blind pit is all that is necessary.
4. For longer lines, manholes must be located according to judgment and depending upon the expansion value of the type of expansion joint or bend that is used. The minimum number of manholes will be required when an expansion bend or an anchor with double expansion joint is placed in each manhole and the pipes are anchored midway between manholes.
5. A proper hydrostatic test should be made on the assembled line before the insulation and the top of the conduit are applied. The hydrostatic pressure should be one-and-one-half times the maximum allowable pressure and it should be held for a period of at least two hours without evidence of leakage. In any case the pressure should be no less than 100 lb per square inch.

The styles and construction of conduits commonly used may be classified as follows. Some of the more common forms are illustrated in Fig. 1.

Wood Casing: The pipe is enclosed in a cylindrical casing usually having a wall 4 in. thick and built of segments which are bound together by a wire wrapped spirally around

the casing. The casing is lined with bright tin and coated with asphaltum. The pipe is supported on rollers carried in a bracket which fits into the casing. The lengths of casing are tightly fitted together with a male and female joint. This form of conduit is illustrated in Fig. 1 at A. The casing rests on a bed of crushed stone with tile drains laid below. The tile drains are of 4-in. field tile or vitrified sewer tile, laid with open joints.

Filler Type: The pipes are supported on expansion rollers properly supported from the conduit or independent masonry base. The pipes are protected by a split-tile conduit, and the entire space between the pipes and the tile is filled with an insulating filler. Thus the pipes are nested and the insulation between them and the tile effectively prevents circulation of air. The conduit is placed on a bed of gravel or crushed rock from 4 to 6 in. thick, which is extended upward so as to come about 2 in. above the parting lines of the tile. A tile underdrain is placed beneath the conduit throughout the entire length and is connected to sewers or to some other point of free discharge. At B and D in Fig. 1 are shown two forms of tile conduit of the filler type.

Circular Tile or Cast-Iron Conduit: The pipes are carried on expansion rollers supported on a frame which rests entirely on the side shoulders of the base drain foundation.

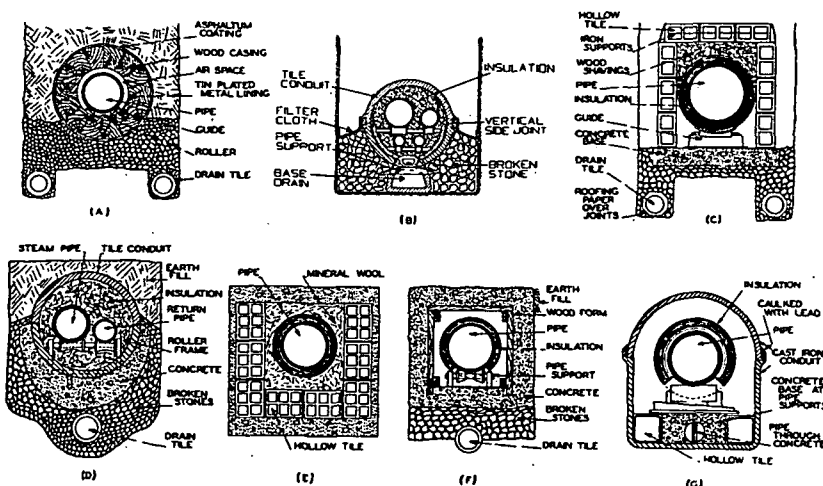


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

The pipes are protected by a sectional tile conduit, scored for splitting, or a cast-iron conduit, both being of the bell and spigot type. The conduit has a longitudinal side joint for cementing, after the upper half of conduit is in place, so shaped that the cement is keyed in place while locking the top and bottom half of the conduit together with a water-tight vertical side joint. The cast-iron conduit has special side locking clamps in addition to the vertical side joint. The entire space between the conduit and the pipes is filled with a water-proofed asbestos insulation. The conduit is supported on the base drain foundation, each section resting on two sections of the base drain, thus interlocking. The base drain is so shaped that it provides a cradle for the conduit, resting solidly on the trench bottom and providing adequate drainage area immediately under the conduit. The underdrain is connected to sewers or some other point of free discharge. For tile conduit the base drain is vitrified salt glazed tile and for cast-iron conduit it is either extra heavy tile or cast-iron. A free internal drainage area is also provided to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint in the conduit. Broken stone is filled in around the base drain and up to the vertical side joint. The broken stone is covered with an asphalted filter cloth to prevent sand from sifting through the broken stone and clogging the drainage area of the base drain. The tile conduit is made in 2-ft lengths and the cast-iron conduit in 4-ft lengths, cast in

separate top and bottom halves. Special reinforcing ribs give the cast-iron conduit ample strength with minimum weight.

Insulated Tile Type: The insulating material, diatomaceous earth, is molded to the inside of the sectional tile conduit. The space between the pipes and the insulating conduit lining may also be filled with insulation. The pipes are carried on expansion rollers supported on a frame which rests on the side shoulders of the base drain foundation. This type of conduit has the same mechanical features as those described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Cast-Iron): Each pipe is insulated in the usual way with any desired type of sectional pipe insulation over which is placed a standard waterproof jacket with cemented joints. The pipes are enclosed in a sectional tile or cast-iron conduit as described under the heading Circular Tile or Cast-Iron Conduits.

Sectional Insulation Type (Tile or Concrete Trench): A type of construction frequently used in city streets, where service connections are required at frequent intervals, the pipes are insulated as described in the preceding paragraph, and are enclosed in a box or trench made either entirely of concrete, or with concrete bottom and specially constructed tile sides and tops. The pipes are supported on roller frames secured in the concrete. At C and E, Fig. 1, are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The use of loose material in addition to the sectional insulation is, of course, optional and is only justifiable where high pressure steam is used. The conduit shown at F is of a similar type and has the advantage of being made entirely of concrete and other common materials.

Sectional Insulation Type (Bituminized Fibre Conduit): Each pipe is individually insulated and encased in a bituminized fibre conduit. The insulating material is 85 per cent carbonate of magnesia sectional pipe covering, applied in the usual manner as on overhead pipes, except that bands are omitted. After every fifth section of magnesia covering there is applied a short, hollow section of very hard asbestos material in the bottom portion of which rests a grooved-iron plate carrying ball-bearings upon which the pipe rides when expanding or contracting. This short expansion section is of the same outside diameter as the adjacent 85 per cent magnesia covering. Over the pipe covering and expansion device there are placed two layers of bituminized fibre conduit with all joints staggered, and the surface of each conduit is finished with liquid cement. Conduits are placed on a bed of crushed rock or gravel, approximately 6 in. deep, and this is extended upward to about the center line of the conduit when trench is backfilled. Underdrains leading to points of free discharge are placed in the gravel or crushed rock beds.

Special Water-Tight Designs: It is occasionally necessary to install pipes in a very wet ground, which calls for special construction. The ordinary tile or concrete conduit is not absolutely water tight even when laid with the utmost care. The conduit shown at G, Fig. 1, is of cast-iron with lead-calked joints and is water tight if properly laid. It is obviously expensive and is justified only in exceptional cases. A reasonably satisfactory construction in wet ground is the concrete or tile conduit with a waterproof jacket enclosing the pipe and its insulation, and with the interior of the conduit carefully drained to a manhole or sump having an automatic pump. It is useless to install external drain tile when the conduit is actually submerged.

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.

On account of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes not installed where provision for the heating lines is the only consideration, but only where they are required

to accommodate miscellaneous other services or provide underground passage between buildings.

SERVICE CONNECTIONS

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.

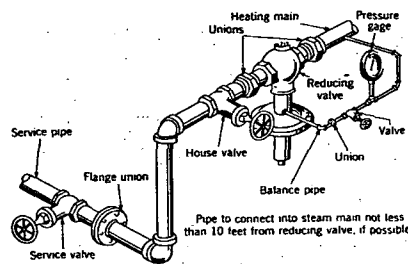


FIG. 2. CONNECTIONS FOR REDUCING VALVES OF SIZE LESS THAN 4 INCHES

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.

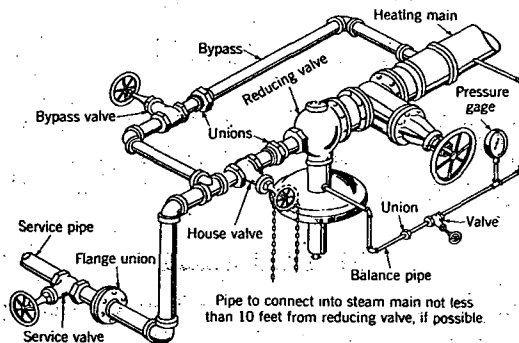


FIG. 3. CONNECTIONS FOR REDUCING VALVES OF SIZE 4 INCHES AND LARGER, AND FOR EXPANDED VALVES

Figs. 2 and 3 show typical service connections used for low pressure steam service. As shown in Fig. 2, no by-pass is used around the reducing valve on sizes less than 4 in. Fig. 3 illustrates the use of a by-pass around reducing valves 4 in. and larger. 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.

Fig. 4 shows a typical installation used for high pressure steam service. The first reducing valve, usually furnished by the utility company,

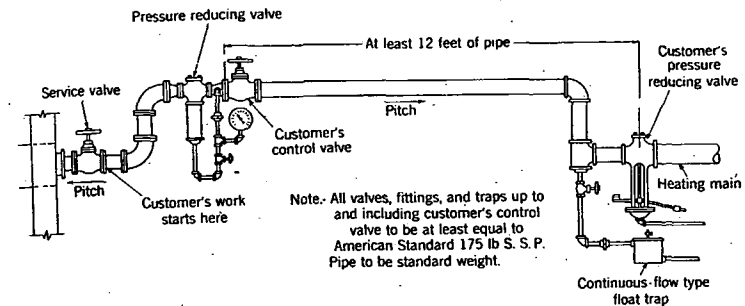


FIG. 4. STEAM SUPPLY CONNECTION WHEN USING CONDENSATION METER

effects the initial pressure reduction. The second reducing valve, usually furnished by the customer, reduces the steam pressure to that required.

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

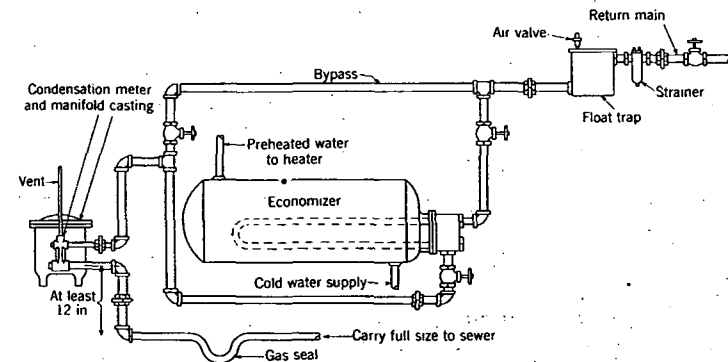


FIG. 5. RETURN PIPING FOR CONDENSATION METER

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

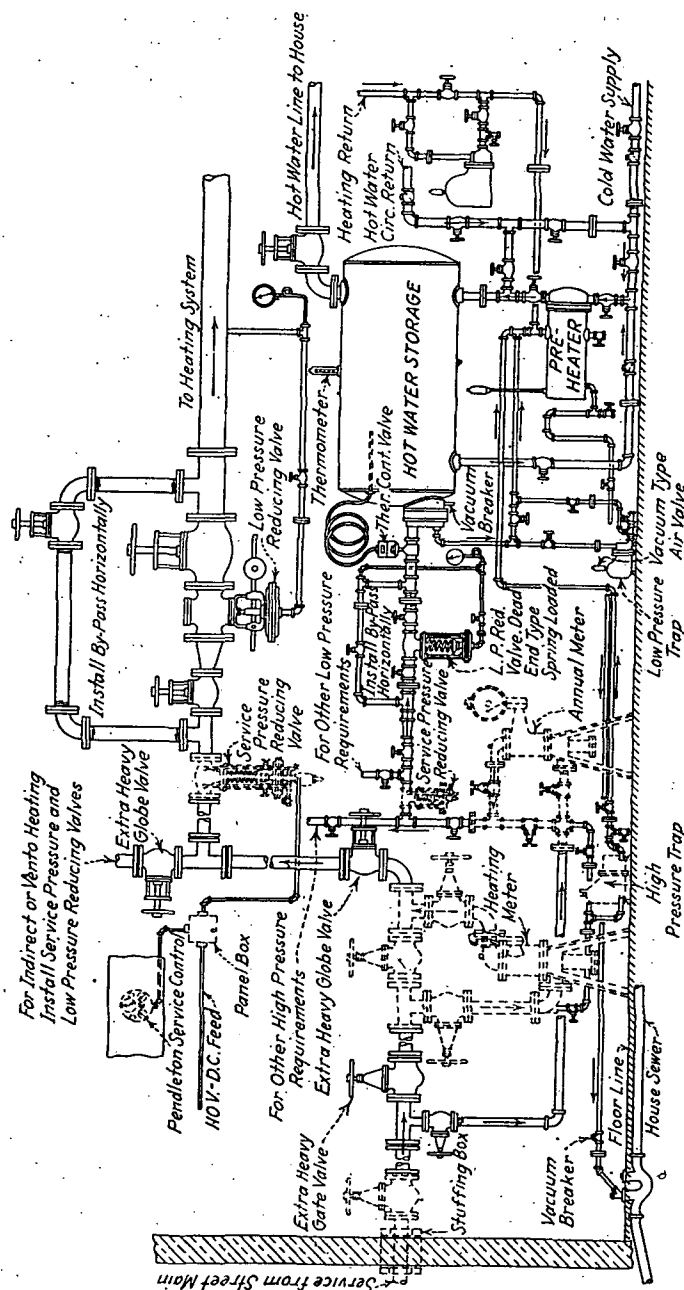


FIG. 6. TYPICAL SERVICE INSTALLATION

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 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 radiator, or as is more frequently done, by a water heating economizer (see Fig. 5) which preheats the hot water supply to the building. Fig. 6 shows a typical steam service installation for high pressure steam, complete for steam flow metering, water heating, preheating, automatic heating control, and for using steam for other purposes.

The condensation from the heating system, after leaving the trap, passes through the preheater on its way to the meter. The supply to the hot water heater passes through the preheater, absorbing heat from the condensation. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in between the preheater and the water heater proper, not at the preheater inlet, because the recirculated hot water is itself at a high temperature. The number of square feet of heating surface in the preheater should be approximately equal to one per cent of the equivalent square feet of heating surface in the building.

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 condensation if storage capacity is provided for the preheated water. Frequently a type of preheater 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.

This may be done in several ways, as by the use of thermostats of various types or by orifice systems. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds pressure, thus producing some control over the heat output. One form of control which appears to be well suited for controlling district steam service to a building is the weather compensating thermostat. It regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, and at the same time insures delivery of steam to all the radiators.

Another form of regulation, known as the time-limit control, is sometimes employed for regulating the steam supply from the central station main to the building. Such a control provides an intermittent supply of steam to the radiators either throughout the 24 hours of the day or during the day-

time hours only. The setting of a switch may provide no service, continuous service, or periodic service. For the latter, by means of several intermittent settings, steam will be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. These settings afford from 15 to 80 per cent of the maximum heating effect required on days of zero temperature. A night switch with a variety of settings may be adjusted so as to maintain throughout the night the intermittent supply called for by the day switch setting, or may be set to interrupt the operation of the day switch and entirely cut off the supply of steam to the radiation at night during certain hours which are selected by the operating engineer.

FLUID METERS

No one thing has contributed more to the advancement of district heating than the perfection of fluid meters, which may be classified as follows:

1. **Positive Meters:** The fluid passes in successive isolated quantities—either weights or volumes. These quantities are separated from the stream and isolated by alternately filling and emptying containers of known capacity.

2. **Differential Meters:** The fluid does not pass in isolated separately-counted quantities but in a continuous stream which may flow through the line without actuating the primary device of the meter. In the differential meter, the quantity of flow is not determined by simple counting, as with the positive meter, but is determined from the action of the steam on the primary element.

Additional subdivisions of these two general classifications can be made as follows:

Fluid Meters	Positive - quantity	Weighing	{ Weighers Tilting trap
		Volumetric	{ Rotary Bellows
	Differential	Quantity - Current - Turbine	
		Head (Kinetic)	{ Venturi Flow nozzle Orifice Pitot tube
		Rate of flow	{ Area (Geometric) Head area (Weir)
			{ Orifice and plug Cylinder and piston
			{ V-notch Special notch

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.

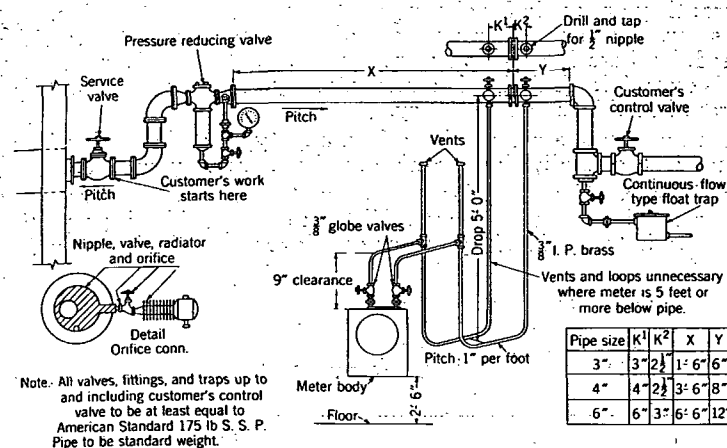


FIG. 7. ORIFICE METER STEAM SUPPLY CONNECTION

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.

Condensation Meters

The majority of the meters used by district heating companies in the sale of steam to their customers are of the condensation or flow types.

The condensation meter is a popular type for use on small and medium sized installations, where all of 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.

Two types of condensation meters are in general use: the *tilting bucket* meter and the *revolving drum* or *rotor* meter of which there are several makes on the market. Condensation meters should not be operated under

pressure; they are made for either gravity or vacuum installation. Continuous flow traps are necessary ahead of the meter if a vented receiver is not used. Where bucket traps are used, a vented receiver before the meter is essential. If desirable a receiver may be used with a continuous flow trap, but this is not necessary.

Steam flow meters are available in many types and combinations, as indicated in the subdivision covering fluid meters on page 680.

The *orifice and plug* meter is one in which the steam flow varies directly as the area of the orifice. The vertical lift of the plug, which is proportional to the flow, is transmitted by means of a lever to an indicator and to a

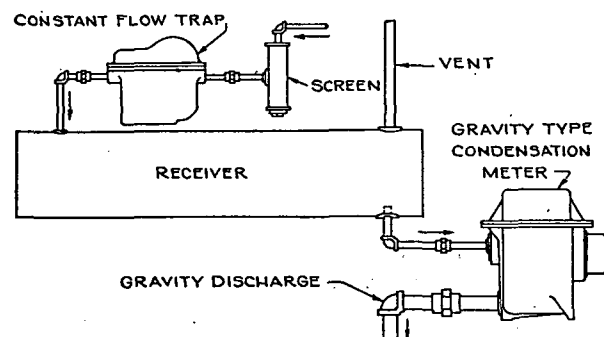


FIG. 8. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

pencil arm which records the flow on a strip chart. The total flow over a given period is obtained by measuring the area by using a planimeter on the chart and applying the meter constant.

Fig. 7 shows a typical orifice type meter connection and indicates typical requirements in the installation of this type of meter. Fig. 8 illustrates a gravity installation using a vented receiver ahead of the meter, while Fig. 9 shows a vacuum installation without a master trap.

Flow meters using an orifice, Venturi tube, flow nozzle, or Pitot tube as the primary device are made by a number of manufacturers and can be obtained in either the mechanically or electrically operated type. The electric flow meter makes it possible to locate the instruments at some distance from the primary element.

Flow meters employing the orifice, Venturi tube, flow nozzle or Pitot tube should be so selected as to keep the lower operating range of the load above 20 per cent of the capacity of the meter. This is desirable for accuracy as the differential pressure at light loads is too small to properly actuate the meter. 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.

STEAM PER SQUARE FOOT OF HEATING SURFACE

The following factors are used in New York City for the different classes of buildings listed. The factors are based on maintaining an inside tem-

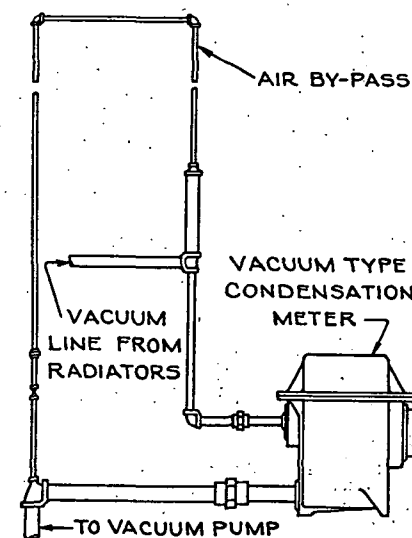


FIG. 9. VACUUM CONDENSATION METER INSTALLATION WITHOUT MASTER TRAP

perature of 70 F for certain hours, with a minimum outside temperature of 0 F and an average of 43 F for the heating season of eight months (October 1 to June 1). In this group are six types of buildings:

Manufacturing or commercial loft type where steam is used to heat the premises during the day hours to maintain 65 to 68 F from 9 a.m. to 5 p.m. No Sunday or holiday use and no night use. *Factor:* 325 lb per square foot of heating surface per season.

Office buildings using steam during daylight hours to maintain 70 F from 9 a.m. to 6 p.m. for approximately 240 days (heating season). No night use. *Factor:* 400 lb per square foot of heating surface per season.

Office buildings using steam during day hours and at night when required to 7, 8 and 9 p.m. (customary where there are stock brokers or banking offices), 240 days. *Factor:* 500 lb per square foot of heating surface per season.

Residences of the block type (not detached) where high-class heating service is required; somewhat similar to apartment buildings. *Factor:* 550 lb per square foot of heating surface per season.

Apartment houses where high-class heating service is required. (Steam off at midnight.) *Factor:* 650 lb per square foot of heating surface per season.

Hotels (commercial type) where very high-class service is required for 24 hours. *Factor:* 800 lb per square foot of heating surface per season.

By assuming one square foot of equivalent heating surface for each 100 cu ft of space heated, which seems a fair ratio in New York City, it is possible roughly to estimate the steam required per cubic foot of space, information which is often more easily obtained than the square feet of heating surface. Additional data on the heating 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. The profit need not be divided proportionately among the rate groups, but should be established from a competitive standpoint. 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 with this principle. Lastly, the rate schedule must be reasonably easy for the intelligent layman to comprehend.

Depreciation should be based on a careful estimate of the life of various elements of the property. Appropriations to reserves should be made, with generosity in good years and with discretion in less favorable years.

Glossary of Terms

Load Factor. The ratio, in per cent, of the average load to the maximum 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 per 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

A. Flat Rates.

1. Radiator surface charge. *Obsolescent*

B. Meter Rates.

1. Straight-line.
2. Step. *Obsolescent*.
3. Block.
- (a) Class rates.

C. Demand Rates.

1. Flat demand.
2. Wright.
3. Hopkinson.
4. Doherty (or Three charge).

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 M 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.

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 M lb 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 flow 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.

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.

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 M lb 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.

REFERENCES

- Pipe Line Design for Central Station Heating*, by B. T. Gifford (A.S.H.V.E. TRANSACTIONS, Vol. 17, 1911).
- Engineering and Cost Data Relative to the Installation of Steam Distributing Systems in a Large City*, by F. H. Valentine (A.S.H.V.E. TRANSACTIONS, Vol. 22, 1916).
- Transmission of Steam in a Central Heating System*, by J. H. Walker (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917).
- Efficiency of Underground Conduit*, by G. B. Nichols (A.S.H.V.E. TRANSACTIONS, Vol. 23, 1917).
- Economical Utilization of Heat from Central Plants*, by N. W. Calvert and J. E. Seiter (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924).
- Standard Connections for Condensation Meters*, (N.D.H.A. Proceedings, Vol. XII, pp. 63-76).
- Installation and Maintenance of Steam Meters*, (N.D.H.A. Proceedings, Vol. XIII, pp. 177-183).
- Inaccuracy in Flow Meter Calculations*, (N.D.H.A. Proceedings, Vol. XIII, pp. 183-193).
- Testing of Steam Meters*, (N.D.H.A. Proceedings, Vol. XIV, pp. 272-276).
- Meter Accuracy Guarantees*, (N.D.H.A. Proceedings, Vol. XIV, pp. 276-277).
- Effect of Pulsations on the Flow of Gases*, (N.D.H.A. Proceedings, Vol. XIV, pp. 277-281).
- Meter Connections*, (N.D.H.A. Proceedings, Vol. XX, pp. 126-143).
- Layout for Testing Meters*, (N.D.H.A. Proceedings, Vol. XX, pp. 391-392).
- Characteristic Meter Calibration Curves* (N.D.H.A. Proceedings, Vol. XX, pp. 444-453).
- Rates* (N.D.H.A. Handbook, 1932, Chapter 10).

PROBLEMS IN PRACTICE

1 • What is the common method of determining the size of mains in a distribution 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 maximum estimated amount of steam may be transmitted without exceeding this pressure difference. The steam's velocity is disregarded and it may reach a magnitude in excess of 35,000 fpm which is not considered high.

2 • a. What are the advantages and disadvantages of a low pressure distribution system?

b. High pressure?

a. The advantages of a low pressure distribution system include:

1. Smaller heat loss from the pipes.
2. Less trouble with traps and valves.
3. Simpler problems with pressure reducing equipment at the buildings.
4. No danger to building heating equipment from high pressure through failure of the reducing valves.

The disadvantages of a low pressure system are:

1. Larger pipe sizes.
2. Decreased field of usefulness owing to small pressure range.

b. The advantages of a high pressure system are:

1. Smaller pipe sizes.
2. Greater adaptability of the steam to various uses other than building heating.

The disadvantages of a high pressure system are:

1. Large heat loss from the pipes.
2. The high pressure traps and valves required often give more trouble than low pressure traps and valves do.
3. Extra heavy fittings are required.
4. Usually two reducing valves or some form of emergency relief is necessary to protect the building piping system.

3 • Determine the size of pipe from the following data using Unwin's formula: Length of pipe, 600 ft.

Steam to be carried, 90,000 lb per hour, dry saturated.

Initial pressure, 100 lb per square inch, gage.

Final pressure, 40 lb per square inch, gage.

Using the formula:

$$P = \frac{0.0001306 W^2 L \left(1 + \frac{3.6}{d}\right)}{y d^5}$$

The pressure drop $P = 100 - 40 = 60$ lb per square inch.

The weight of steam per minute $W = \frac{90,000}{60} = 1500$.

The length of pipe in feet $L = 600$.

The average density of steam y in pounds per cubic foot, taken from Keenan's Table:

At 100-lb gage, $y = 0.2578$

At 40-lb gage, $y = 0.1285$

Average, $y = 0.1932$

The diameter of the pipe in inches = d .

Substituting the values in the formula:

$$60 = \frac{0.0001306 \times 1500^2 \times 600 \left(1 + \frac{3.6}{d}\right)}{0.1932 \times d^5}$$

$$d = 7.35 \text{ in.}$$

Therefore, an 8-in. pipe should be used.

4 • What points should be borne in mind when laying out an underground steam conduit?

The conduit 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 the 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.

5 • What is considered the proper pressure for a hydrostatic test before completing the conduit?

In the case of any underground piping which is to be buried or otherwise made inaccessible, the assembled lines shall first be tested hydrostatically at a pressure of one and one-half times the maximum allowable service pressure and held for a period of at least two hours without evidence of leakage. In any case the hydrostatic pressure should not be less than 100 lb per square inch.

6 • What factors should be considered before determining the route of a steam line?

1. The line should be so located that it will bring in the greatest revenue (or supply the most steam) with the least cost.
2. The ultimate length and size of services and branches necessary with each possible location should be estimated, for mains should be run near to the big loads.
3. The location of the boiler room or piping center of present and future buildings to be served should be considered.
4. Where possible, make the lines straight between manholes.
5. Avoid such obstructions as other lines, sewers, ducts, curb drains, manholes, valve boxes, catch basins, fire hydrants, and poles; especially avoid electric ducts and water lines.
6. Avoid locating lines near where pile driving and foundation construction for new buildings will take place.
7. Consider construction difficulties such as traffic, hard rock, and wet earth, which increase time and labor.
8. Consider the economies of using available sidewalk vaults of buildings. Weigh the advantage of less excavation against the cost of obstruction removal.
9. Consider all operating difficulties.
10. Consider the difficulties of negotiating agreements for lines on private property where public and private rights-of-way are available.
11. Consider the effect of proposed municipal and other improvements.
12. Consider municipal regulations.

7 • State the advantages and disadvantages of tunnels over conduits.

The advantages of pipe tunnels over conduits are:

1. Accommodation for miscellaneous services other than steam.
2. Provision of an underground passage between buildings.
3. Easy installation of additional pipes and easy replacement of existing pipes with larger sizes.
4. Easy inspection and maintenance of pipes.

The disadvantages of pipe tunnels over conduits are:

1. Higher first cost.
2. Higher maintenance cost in general.

8 ● Is the steam consumption less in a building that shuts off its steam at night than in one that does not? Why?

It has been thoroughly demonstrated that the steam consumption is less in a building where the steam is shut off 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.

9 ● Is the condensate from a building supplied with purchased steam always discharged to the sewer?

No. In some cities where the customers are not spread over too wide a territory and where natural water conditions make the treatment of boiler feed water expensive, the steam company provides mains for the return of the condensate to the boilers.

10 ● What are the common methods for salvaging heat in condensate?

The most common methods are:

1. The use of a water heating economizer for preheating the hot water supply to the building.
2. The use of a cooling radiator.

11 ● What are the common means used to graduate the heat supply according to variations in outside temperature?

- a. A weather compensating thermostat regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply; at the same time it insures delivery of steam to all the radiators.
- b. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds to produce some control over the heat output.
- c. The use of an orifice system graduates heat supply.
- d. The time-limit control which may be set to provide no service, continuous service, or periodic service, is also used. For periodic service, steam may be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. This type of service is provided by several intermittent settings. A night switch will maintain the intermittent day setting, or interrupt the day operation and cut off the supply of steam at night during any desired hours.

Chapter 38

RADIANT HEATING

Physical and Physiological Considerations, British Equivalent Temperature, Control of Heat Losses, Methods of Application, Principles of Calculation, Mean Radiant Temperature, Measurement of Radiant Heating

HHEATING for comfort is generally understood to mean that heat must be supplied to control the rate of heat loss from the human body so that the physiological reactions are conducive to a feeling of comfort in the individual. While in convection heating, as described in Chapter 30, heat is transferred from a heating unit to the air and thence to the occupant, the primary object of radiant heating is to warm the occupant directly without heating the air to any extent. Thus, the difference between convection heating and radiant heating is partly physical and partly physiological.

Comfort requires that heat be removed from the body at the same rate as it is generated by the oxidation of the foodstuffs in the body tissues. The normal rate of heat production in a sedentary individual is about 400 Btu per hour¹, or, since the entire surface area of an average adult is 19.5 sq ft, about 20.5 Btu per square foot per hour. Conditions should be such as to remove heat at this rate if the surface is to be maintained at the mean normal surface temperature of the human body.

Heat is transferred from any warm dry body to cooler surroundings principally by convection and by radiation, the approximate total rate of heat loss being the sum of the two. Where the body surface is moist there is additional loss of heat through evaporation from both the body surface and the respiratory tract.

The rate of heat loss by convection depends upon the difference between the temperature of the body and that of the surrounding air, and on the rate of air motion over the body. The loss by radiation depends entirely upon the difference between the temperature of the body and the mean surface temperature of the surrounding walls and objects. This latter temperature is called the *mean radiant temperature* (MRT). Because these two types of heat loss act in a supplementary manner toward each other, a required rate of heat loss can be secured by having a relatively low air temperature and a relatively high MRT, or *vice versa*. Thus, if the air is reduced from a given temperature to a lower temperature, the amount of heat lost from the body by convection is increased, and this

¹Heat 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).

increase can be compensated for by raising the MRT. Similarly, with a higher air temperature the same total heat loss will be maintained by a correspondingly lower MRT.

The loss by evaporation depends on the air temperature, air movement, and humidity; it is increased if the humidity is reduced. For the usual conditions of heating by radiators or convectors, where the air temperature ranges from 70 F to 73 F, approximately 75 per cent of the total heat loss of 400 Btu per hour occurs by radiation and convection, and the balance, or 100 Btu per hour, occurs by evaporation. In the case of radiant heating, if the air temperature is reduced to 60 F, 84 per cent of the 400 Btu per hour, or 336 Btu per hour, is lost by radiation and convection, and 64 Btu per hour are lost by evaporation.

The mean normal surface temperature of the human body, taken over the whole area, including not only the exposed skin surface but also surfaces of the clothes and the hair, has been very extensively used as 75 F, particularly in British literature. However, results obtained by Aldrich² in rooms in which the air and wall surface temperatures were approximately 72 F gave mean values nearer to 83 F than to 75 F.

The mean body surface temperature which will maintain the optimum heat loss by radiation and convection in a uniform environment of 72 F may be calculated from fundamental equations for radiation and natural convection by substituting a comparable cylinder for the body. Heilman³ gives the following equations:

$$H_r = 0.1723 \epsilon \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right] \quad (1)$$

$$H_c = 1.235 \left(\frac{1}{D} \right)^{0.2} \times \left(\frac{1}{T_m} \right)^{0.181} \times (T_s - T_a)^{1.266} \quad (2)$$

where

- H_r = heat loss by radiation, Btu per square foot per hour.
- H_c = heat loss by convection, Btu per square foot per hour.
- T_s = absolute temperature of the body surface, degrees Fahrenheit.
- T_w = absolute temperature of the walls, degrees Fahrenheit.
- T_a = absolute temperature of the air, degrees Fahrenheit.
- $T_m = \frac{T_s + T_a}{2}$
- D = diameter of cylinder, inches.
- ϵ = the ratio of actual emission to black body emission.

If it be assumed that a normal adult has an average height of 5 ft 8 in. and an average body surface area of 19.5 sq ft, the surface of his body will have the same area as that of a cylinder 5 ft 8 in. long with a diameter of 13.15 in. The value of ϵ for skin and clothing is practically 0.95. T_a and T_w are each taken as 72 F, or 532 Absolute. The sum of H_r and H_c is taken to be 15.4 Btu per square foot per hour, which is derived as the normal rate of heat loss due to convection and radiation from a sedentary individual by dividing his total sensible heat loss by his area. Solution of

²A study of Body Radiation, by L. B. Aldrich (Smithsonian Miscellaneous Collections, Vol. 81, No. 6, December, 1928).

³Surface Heat Transmission, by R. H. Heilman (Trans. A.S.M.E., Fuels and Steam Power Section, Vol. 51, No. 22, September-December, 1929).

Equations 1 and 2, using average figures as outlined, gives a value of approximately 83 F for the normal temperature of the body surface. This agrees more closely with the values obtained by Aldrich than with the 75 F used by British investigators.

British Equivalent Temperature

The British Equivalent Temperature BET is the temperature of an environment which is effective in controlling the rate of sensible heat loss from a sizable black body in still air when the body has a maintained surface temperature of 83 F. The BET is, therefore, a function of both the air temperature and the mean radiant temperature. Its numerical value in a uniform environment (walls and air at the same temperature) is equal to the temperature of the walls and air. In a non-uniform environment (walls and air at different temperatures) the BET is equivalent to that of a uniform environment in which an 83 F surface loses sensible heat at the same rate as it does in the non-uniform environment. As originally defined, the BET was based on a body surface temperature of 75 F, but 83 F has been accepted as giving results more nearly conforming with American practice⁴. The higher the BET the less the heat loss from the body, the rate of loss in still air being approximately proportional to the difference between the BET and the mean body surface temperature.

If the BET were 83 F, there could be no sensible heat loss from a surface at that temperature, so the temperature of a normal body surface would have to rise to a point where the heat generated in the tissues could be dissipated.

When convected heat is used, the temperatures of the air and walls are nearly the same, and the optimum value of the BET from the physiological point of view is 72 F. Under these conditions the mean surface temperature of a normal body would have the optimum value of 83 F because the rate of heat loss by radiation and convection would be 15.4 Btu per square foot per hour and that by evaporation 5.1 Btu per square foot per hour, which would just balance the rate of heat production of 20.5 Btu per square foot per hour. This BET of 72 F in a uniform environment is exactly equivalent to the *effective temperature* of 66 F as defined by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS (see Chapter 3), because, in a uniform environment, a dry-bulb temperature of 72 F in still air with a relative humidity of 30 per cent gives an *effective temperature* of 66 F, which has been determined to be the optimum.

METHODS OF APPLICATION

There are two general methods of applying radiant heating, as follow:

1. *By warming the interior surfaces of the building.* Pipe coils are embedded in the concrete or plaster of the walls, ceiling or floors, the heating medium being hot water or, in some cases, steam. This has the effect of warming the entire concrete or plaster surface in which the pipes are embedded. Since the temperature of the heating medium should not exceed about 120 F on account of the possibility of cracking the plaster, the

⁴Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperatures, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock (A.S.H.V.E. JOURNAL, Heating, Piping and Air Conditioning, July, 1933).

area of the panel must be sufficient to supply the requisite quantity of heat at this low temperature. When carefully designed, this method produces comfortable and economical results.

2. *By attaching separate heated plates or panels to the interior surfaces of the structure.* These plates or panels are placed either in an insulated recess flush with the surface of the walls or ceiling or bolted on its face. They may be decorated as desired. As it is difficult to make an invisible joint between the edge of such a plate and the plaster, it is common to use a frame of plaster, wood, metal or composition around the panel. These plates may be placed either on the ceiling or the wall, or in some cases as a margin around the edge of the floor. If floor heating is required the temperature over the whole area should not exceed 70 F.

If the entire warm surface is installed at one end of the room there may be a marked difference between the BET on the two sides of a body in the room. It is usually desirable therefore that the heat be distributed at different points in the room so that no uncomfortable effects will be felt from unequal heating.

PRINCIPLES OF CALCULATION

The calculations for radiant heating are entirely different from those for convective heating. The purpose of the latter is to determine the rate of heat loss from the room by conduction, convection, and radiation when maintained in the desired condition; radiant heating involves the regulation of the rate of heat loss per square foot from the human body.

The first step in the calculations for radiant heating is to ascertain the necessary mean radiant temperature (MRT); next, the size, temperature, and disposition of the heating surfaces required in the room to produce this MRT are estimated; and after this the determination of the convective heat is made.

Mean Radiant Temperature

If the whole of the interior surface of a room were at the same temperature, this temperature would represent the MRT. Such a condition seldom exists, however, since the actual surface temperature in any heated space having surfaces exposed to the outer air varies greatly for different sides of the enclosure. It is therefore necessary to ascertain by calculation the mean of these interior surface temperatures.

The mean temperature in this sense is not the arithmetic average of the actual thermometric temperatures of the surfaces, but the temperature corresponding to the average rate of heat emission per square foot of surface. The temperature corresponding to this mean emission can be taken from Table 1. Conversely, the emission at different temperatures and also the emissivity factors can be obtained from this table. For instance, 1 sq ft of surface at 50 F will emit 104.9 Btu per square foot per hour to surroundings at absolute zero if the emissivity of the surface is 0.9.

If the area in square feet of each part of the space is multiplied by the emission value corresponding to its actual temperature, and these products are added together, the gross amount of radiant heat discharged into the room by the wall surface per hour is obtained. This quantity, divided by the total interior surface, gives the average amount of heat coming into the room from the surface of the walls per square foot of surface per hour.

Interpolating in Table 1, the total radiation from a surface at 83 F for

TABLE 1. TOTAL BLACK BODY RADIATION TO SURROUNDINGS AT ABSOLUTE ZERO*

BODY OR MEAN RADIANT TEMPERATURE Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor ϵ				BODY OR MEAN RADIANT TEMPERATURE Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor ϵ			
	ϵ 1.00	ϵ 0.95	ϵ 0.90	ϵ 0.80		ϵ 1.00	ϵ 0.95	ϵ 0.90	ϵ 0.80
30	99.3	94.3	89.4	79.4	71	136.5	129.6	122.9	109.3
35	103.5	98.3	93.2	82.8	72	137.4	130.5	123.6	109.9
40	107.6	102.4	96.8	86.1	73	138.4	131.5	124.5	110.6
45	112.1	106.5	100.9	89.7	74	139.6	132.6	125.6	111.7
46	112.9	107.3	101.6	90.4	75	141.0	133.9	126.9	112.8
47	113.9	108.2	102.5	91.1	80	146.6	139.4	132.0	117.4
48	114.8	109.1	103.4	91.9	85	152.3	144.6	137.1	121.9
49	115.6	109.9	104.1	92.4	90	157.9	149.9	142.1	126.4
50	116.5	110.6	104.9	93.2	100	169.6	161.1	152.6	135.7
51	117.5	111.6	105.8	94.0	110	181.6	172.5	163.5	145.4
52	118.4	112.5	106.5	94.7	120	194.8	185.0	175.4	155.9
53	119.4	113.4	107.4	95.5	130	210.1	199.6	189.1	168.1
54	120.2	114.2	108.2	96.2	140	223.2	212.1	201.0	178.5
55	121.1	115.1	109.0	96.9	150	237.1	225.2	213.5	189.7
56	122.1	116.0	109.9	97.7	160	251.1	238.8	226.0	201.0
57	123.1	117.0	110.9	98.5	170	270.5	257.0	243.5	216.4
58	124.0	117.8	111.6	99.2	180	288.0	273.8	259.1	230.4
59	124.9	118.6	112.4	99.9	190	306.5	291.0	275.8	245.1
60	125.8	119.5	113.4	100.7	200	325.2	309.0	292.8	260.3
61	126.6	120.3	114.0	101.4	210	348.0	330.6	313.1	278.4
62	127.7	121.4	114.9	102.2	220	371.5	353.0	334.4	297.1
63	128.6	122.2	115.8	102.9	250	437.8	415.9	394.0	350.2
64	129.6	123.1	116.7	103.7	300	575.0	546.1	517.5	460.0
65	130.5	124.0	117.5	104.4	350	740.0	703.0	666.0	592.0
66	131.6	125.0	118.4	105.4	400	942.1	895.0	847.5	753.5
67	132.5	125.9	119.3	106.0	450	1176.0	1117.0	1059.0	941.0
68	133.5	126.8	120.1	106.8	500	1464.0	1390.0	1318.0	1171.0
69	134.5	127.8	121.1	107.6	550	1791.0	1701.0	1613.0	1434.0
70	135.5	128.8	121.9	108.4	600	2405.0	2284.0	2165.0	1925.0

*These factors are calculated from the formula

$$Q = \epsilon \left(\frac{0.1723 \times T^4}{100,000,000} \right)$$

where

Q = total black body radiation, Btu per square foot per hour.

ϵ = emissivity.

T = absolute temperature, degrees Fahrenheit.

an emissivity of 0.95 is 142 Btu per square foot per hour. The difference between 142 Btu and the average amount of heat coming into the room is the amount which will be lost per square foot per hour by radiation from a body at 83 F. If a rate at which it is desired that heat be lost from the body by radiation and convection be assumed, the mean radiant emission from the walls required to give the desired result can be determined from Table 1, as can also the required air temperature for the corresponding convective effect.

The determination of the amount of radiant heating surface needed in a room requires knowledge of the climate, the type of structure, the type of heating, and the surface temperature of the walls. This problem can be solved only on an empirical basis. After some experience, however,

it is possible to estimate these variables with a considerable degree of accuracy for any climate or construction.

Assume that a mean radiant temperature of 65 F is desired. Table 1 shows that with all the walls at this temperature, and with an emissivity of 0.95, the gross heat emission is 124 Btu per square foot per hour. The total emission of radiation into the room from that surface would therefore be $A \times 124$, where A is the total inside area of the room. This is the desired emission.

If the whole area be divided into a number of different parts which are each at a uniform temperature— a_1, a_2, a_3 ,—and each is multiplied by the value of the heat emission corresponding to that temperature, and if all these products are added together, their sum will represent the total actual emission of radiation into the room at these temperatures without the aid of any hot surface.

The difference between the desired emission and the actual emission represents the additional heat which must be supplied by the hot surface. The temperature of the proposed hot surface must then be selected, and its emission per square foot at that temperature determined from Table 1. This emission is divided into the additional amount of heat needed, adjusted for the fact that the heating units will shield the walls behind them, and the quotient obtained will be the area of the required heating surface.

It is evident that this method of calculation is approximate, and depends for its accuracy on a correct estimate of the ultimate surface temperatures attained by the actual wall surfaces.

It is necessary also to calculate how much heat will be given off by the same surfaces by convection, and thereby to determine whether this amount of convected heat will warm entering ventilating air to the temperature maintained. If it will not, additional convection surfaces must be introduced to make up the deficiency.

MEASUREMENT OF RADIANT HEATING

Convection heating, having as its object the raising of the air temperature to a specified degree, must be measured by thermometric methods which indicate essentially the air temperature, and not the rate of heat loss from the human body. Radiant heating, having as its object the control of the rate of heat loss from the human body, can be measured only by methods which basically are calorimetric, that is, which measure directly the rate of heat loss from an object maintained at the temperature of the body, irrespective of air temperature.

The apparatus for this purpose consists essentially of a hollow sphere, or cylinder, containing a fluid which can be maintained accurately at 83 F (the accepted mean surface temperature of the human body), with an accurate means of measuring the rate of heat supply required to maintain the temperature at that exact point. The latter measurement can be made with sufficient accuracy by electrical methods. Although a BET of 72 F is desirable, the mean radiant and air temperatures may both vary, provided the heat loss by radiation and convection from a surface at 83 F is maintained at the rate of 15.4 Btu per square foot per hour,

which corresponds to $\frac{15.4}{3.415} = 4.5$ watts per square foot of exposed surface.

This instrument, the *eupatheoscope*, can readily be adapted as a thermostat by electrical control to shut off or turn on heat when the critical temperature of 83 F in the vessel is increased or decreased. A modification of the instrument is called the *eupatheostat*.

Another instrument for maintaining comfort conditions is at present available only in a model adapted to British practice as it is designed for a temperature of 75 F. It consists of a blackened copper sphere of approximately 6 in. diameter in which is housed a cylindrical sump containing a volatile liquid. In operation, a small electric heating coil drawing about 5 watts creates in the sphere a vapor pressure which is constant as long as the heat losses from the sphere are standard. If the temperature of the air or the MRT becomes too high for comfort, a greater pressure is created, owing to a smaller loss of heat from the sphere. This increase of pressure acts on a diaphragm and shuts off the supply of heat to the room.

For testing work, the *globe thermometer* is a very useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere from 6 in. to 9 in. in diameter, usually made of thin copper and painted black. The temperature thus recorded is termed the *radiation-convection temperature*.

EXAMPLE

Example 1. The surface areas, temperatures, and emissions for a room having a volume of 5760 cu ft are given in Table 2. The figures for temperatures are fairly representative of American practice with well-built walls, and are based on an emissivity of 0.95 which approximates that of most paints and building materials.

TABLE 2. SURFACE AREAS, TEMPERATURES, AND EMISSIONS FOR A ROOM OF 5760 CU FT

	AREA Sq Ft.	ASSUMED SURFACE TEMPERATURE (DEG FAHR)	HEAT EMISSION (BTU PER SQ FT PER HOUR)	TOTAL HEAT EMISSION FROM AREA (BTU PER HOUR)
External Wall.....	297	50	110.6	32,850
Glass.....	279	45	106.5	29,710
Inner Wall.....	480	55	115.1	55,250
Ceiling.....	480	55	115.1	55,250
Floor.....	480	55	115.1	55,250
Total.....	2016			228,310

The mean radiant temperature of the room is $\frac{228,310}{2016} = 113.2$ Btu per square foot per hour which, as seen from Table 1, corresponds to an MRT of 53 F for an average emissivity of 0.95.

For an average individual having a body surface of 19.5 sq ft, under conditions of comfort with a body surface temperature of 83 F, the heat given off by radiation may be determined by means of Equation 1 as 217 Btu per hour, or 11.1 Btu per square foot per hour. This corresponds to an environmental emission of $142 - 11.1 = 130.9$ Btu per square foot per hour, and, according to Table 1, to an MRT of 72 F.

If this body be placed in the room described, it will lose heat at the rate of 19.5 (142 - 113.2) = 562 Btu per hour. This loss is 345 Btu per hour, or 17.7 Btu per square foot per hour, more than the rate of heat loss for comfort, which is only 19.5 (142 - 130.9) = 217 Btu per hour.

In order to determine the amount of radiating surface necessary to maintain the MRT at 72 F, assume the surface temperature of the hot plates to be installed to be 200 F, which is approximately the temperature they would have if heated by steam.

The 2016 sq ft total area of the surfaces of the room multiplied by 130.9, which is the emission in Btu per square foot per hour necessary to maintain a body surface temperature of 83 F, gives a total desired emission of 263,890 Btu per hour. It is necessary to supply enough radiant heating surface to increase the total actual mean radiant heat emission by the room from 228,310, as shown in Table 2, to the 263,890 Btu desired. The additional heat needed is the difference between these figures, or 35,580 Btu. Since, from Table 1, the emission per square foot at 200 F is 309 Btu, the required radiant heating surface needed is $\frac{35,580}{309} = 115$ square feet. The effect of this surface suitably

placed would be to raise immediately the mean radiant temperature to the required degree and to maintain it at that value as long as the surfaces remained at the values assumed.

In the solution of this particular example, the radiation loss from the human body was selected as 217 Btu per hour, which is that taking place under optimum comfort conditions, with a body surface temperature of 83 F in a uniform environment at 72 F. The mean radiant temperature necessarily was 72 F. If the optimum BET of 72 deg Fahr is desired, an air temperature of 72 F also must be maintained. If it is desired to maintain a lower air temperature than this, a mean radiant temperature greater than 72 F must be selected and the radiation loss from the individual must be recalculated from Equation 1.

The calculation may be simplified by preparing tables showing, at the usual temperatures, the area of hot surface required to bring each square foot of actual wall surface at various temperatures up to a general standard of from 60 F to 70 F. It would then be necessary only to multiply the respective areas by the appropriate factors, and to add the results, to obtain the required total.

REFERENCES

- Room Warming by Radiation, by A. H. Barker (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).
- Panel Warming, by L. J. Fowler (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).
- Calculations for Radiant Heating, by T. Napier Adlam (*Heating and Ventilating*, October, 1931).
- Principles of Calculation of Low Temperature Radiant Heating, by A. H. Barker (*Proceedings of The Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).
- Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperatures, by A. C. Willard, A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).
- What will be the Future Development of Heating and Air Conditioning, by W. H. Carrier (*Heating, Piping and Air Conditioning*, January, 1933).
- Method of Installing the Panel Heating System in the British Embassy Building (*Heating, Piping and Air Conditioning*, July, 1934).
- Panel Heating, by C. M. Oates (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).
- Notes on Electric Warming with Special Reference to Low Temperature Panel Systems, by R. Grierson (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 28, 1929).
- Radiant Heat, by A. F. Dufton (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 30, 1931).
- Radiant Heat, by A. F. Dufton (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 31, 1932).
- Notes on the Theory of Radiant Heating, by C. G. Heys Hallett (*Proceedings of Institution of Heating and Ventilating Engineers*, London, Vol. 29, 1930).

PROBLEMS IN PRACTICE

1 • Differentiate between radiant and convection heating.

The primary function of radiant heating is to warm the individual by direct heat rays and without heating the surrounding air, while in convection heating it is generally the function of the heating medium to transfer the heat to the air and thence to the occupant of the room.

2 • What is the mean normal surface temperature of the human body as determined for the United States?

83 F.

3 • Explain the difference between radiant heating and ordinary methods of heating.

Radiant heating provides conditions to control the heat loss from the human body according to the physiological requirement of the body; whereas, ordinary methods of heating warm the air in a building to a predetermined temperature compatible with a feeling of comfort.

4 • Describe how to calculate the required amount of radiant heating surface.

a. Obtain the mean heat emission in Btu per square foot per hour for room surfaces X , using values given in Table 1, and surface temperatures as shown in second column of Table 2.

b. Deduct X from 142 (142 being the emission per square foot given off by the human body at 83 F surface temperature) = Y in Btu per square foot per hour.

c. From $(142 - X)$ deduct 11.1 (11.1 being the average radiation which the human body should lose per square foot for comfort conditions) = $(142 - X - 11.1) = Z$.

d. Multiply total interior surface of room by Z and divide by the emission per square foot from radiant heater, giving the surface S of radiant heater in square feet.

5 • Give a simple formula to calculate radiant heating surface required, and explain.

$$S = \frac{(142 - X - 11.1) A}{B}$$

where

S = surface of radiant heater, square feet.

142 = Heat emission, Btu per square foot per hour which the human body would give off at 83 F, with surroundings at absolute zero.

X = mean heat emission, Btu per square foot per hour from surfaces of room.

11.1 = heat emission, Btu per square foot per hour from human body.

A = total surface, square feet of walls, ceilings, windows, etc., in room.

B = heat emission per square foot from radiant heater surface.

6 • Is the heat generated in the body affected by action? If so, does it vary greatly?

Yes. With hard work or energetic exercise, the total heat generated in the body may be five to six times that generated when it is at rest.

7 • When and why does the human body feel cold?

The body feels cold not only when it loses heat at a greater rate than it can generate it, but also when heat is abstracted from the body disproportionately. The human body does not require any heat from without because it generates more heat than is sufficient to maintain the correct temperature; therefore, it is only necessary to provide conditions that will maintain the correct ratio of losses.

- 8 ● a. Where did radiant heating derive its name?
b. What is actually meant by radiant heating?

- a. The term radiant heaters was introduced about 25 years ago to designate flat heating surfaces made to give off practically all their heat by radiant ether waves instead of relying on convected warm air.
- b. The term radiant heating now applies to methods of heating where, instead of heating the air to a predetermined temperature, flat heating surfaces are so placed in a room that the average virtual temperature of all wall, ceiling, floor, and glass surfaces exposed to the body is just sufficient to prevent the body's losing too much heat by radiation. The air temperature can be much cooler with radiant heating because radiation losses from the body are compensated.

- 9 ● What kind of heating surfaces are in general use?

The heating units may have flat iron surfaces heated with steam and placed under windows, or hot water pipes may be embedded in the floor, walls, or ceilings. Electrical radiant heaters are made by embedding resistance elements in porcelain or electric conductors woven into a thick paper which can be fastened to the walls or ceilings.

- 10 ● What kind of heat rays are commonly generated in radiant heating? Give examples.

All heat rays are generally assumed to be the same as light rays; they travel at the speed of light, but they are invisible and longer. The rays used in heating are 0.00005 to 0.0001 in. long compared with visible red rays of about 0.000027 in.

- 11 ● What natural evidence have we that air temperature alone is no criterion of comfort and that radiant heat affects the body more quickly?

When standing in the sunshine on a cool spring day, a person feels perfectly comfortable, but when a cloud passes over the sun, he instantly feels much cooler as the shadow reaches him. A shielded thermometer recording the temperature of the air shows no reduction in air temperature in so short a period, so that the person actually feels a sensation of cold which an ordinary thermometer cannot register. This shows that light and heat rays are shut off simultaneously and travel at the same speed; it also proves that radiant rays affect the comfort of the body quicker than air temperature does.

Chapter 39

ELECTRICAL HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Steam Heating, Electric Hot Water Heating, Electric Hot Water Heating for Domestic Supply, Cooling and Reverse Cycle Heating by Electric Refrigeration, Auxiliary Electric Heating, Control, Calculating Capacities, Power Problems, Insulation, Electric Heating Data

ELECTRIC heating is steadily assuming a more important place in heating, ventilating and air conditioning installations, accelerated in many territories by the load building efforts of the utilities which usually include reduced rates to encourage such installations. Electrical heating has a logical place in the heating industry because of its features of flexibility, cleanliness, safety, convenience and ease of control. Electrical heating practice has many basic principles in common with fuel heating, but there are also important differences. When heat units are delivered to each room by wire, no combustion process is necessary, either at a central plant or at the individual room units. The maximum output of an electric heater is a fixed constant, unaffected by the temperature of the surrounding air and it follows that the maximum total load on an electrical heating system is the total wattage of connected electric heaters, regardless of weather conditions. The real obstacle to the more general adoption of electric heating for buildings is the cost of the electricity itself. Because the heat units produced electrically are more costly, their conservation is of more relative economic importance than with fuel heating, so that sponsors of electric heating give greater attention to temperature-insulated building construction.

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, because 100 per cent of the energy applied in the resistor is always transformed into heat. In electric heating practice the engineer need not be concerned about efficiencies of heat production, but rather about efficiencies of heat utilization. It is the engineer's problem 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 terms used in fuel heating are given in Chapter 44. The following terms apply particularly to electric heating:

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

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.

HEATING ELEMENTS

Commercial electric heating elements are divided into open type elements, enclosed type elements, and cloth fabrics. *Open type elements* have resistors exposed to view. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. Open type elements are used extensively for operation at high temperatures when radiant heat is desired. They are also frequently used at low temperatures for convection and fan circulation heating, especially in large installations.

Enclosed type elements have metallic resistors embedded in a refractory insulating material, and encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat-dissipating area. Enclosed 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 convection and fan circulation 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.

ELECTRIC HEATERS

Electric heaters may be divided into three groups, conduction, radiant and convection.

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 pipe 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 incandescent 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

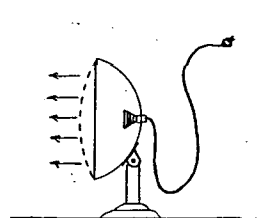


FIG. 1. PORTABLE RADIANT ELECTRIC HEATER

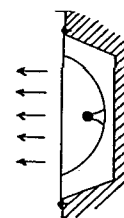


FIG. 2. RADIANT ELECTRIC HEATER RECESSED IN WALL

upon the body. They are not satisfactory for general 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. The location of radiant heaters is important. They should never face a window because some rays would pass through the glass and be lost. Figs. 1 and 2 show common types of portable and wall-mounted radiant heaters.

Gravity convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used

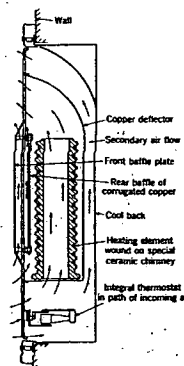


FIG. 3. CONVECTION ELECTRIC HEATER RECESSED IN WALL

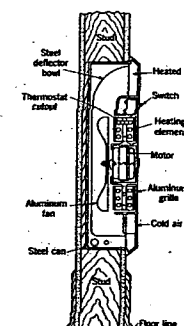


FIG. 4. FAN CONVECTION ELECTRIC HEATER RECESSED IN WALL

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 (Figs. 3 and 4). 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.

Fan convection electric heaters, which include a built-in fan unit, circulate room air over the heating elements at 300 to 400 fpm. Heaters of this type are manufactured in three designs, portable, wall hung and wall insert. The fan draws air from the floor zone through a lower grille, forces it over the heater elements and out through an upper grille into the breathing zone. The improved heating effectiveness of this type electric heater has done much to promote electric heating for residences.

UNIT HEATERS

Fan unit electric heaters, similar to steam unit heaters, having a propeller type fan mounted behind the heating element, are made in many styles and can be located and used much the same as steam unit heaters

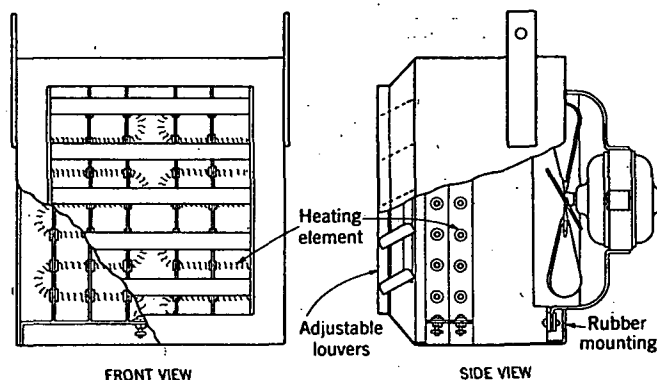


FIG. 5. FAN UNIT ELECTRIC HEATER

(Figs. 5 and 6). These electric unit heaters are used in industrial plants, sub-stations, power houses, pumping stations, etc., where large initial use of electricity for power purposes earns a low rate for electric heating. Portable unit heaters (Fig. 7) of this type 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 served 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.

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 as a change in air volume flowing over steam coils does not greatly affect the temperature 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, being a constant source 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 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

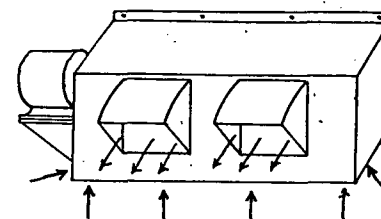


FIG. 6. LARGE INDUSTRIAL TYPE FAN UNIT ELECTRIC HEATER

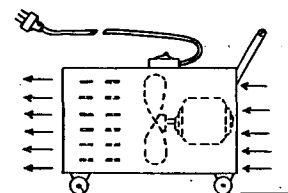


FIG. 7. LARGE INDUSTRIAL TYPE PORTABLE FAN UNIT ELECTRIC HEATER

unless some compensating action is performed by control. Automatic modulation to vary the electrical heat input and synchronize it properly with the air flow has been successfully applied to central fan systems.

ELECTRIC STEAM HEATING

Electric steam heating differs from fuel heating only in the use of *electric boilers* to generate steam. Electric steam boilers 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 on either direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood that the heating elements will burn out, they are made removable for cleaning off deposits of scale which restrict the heat flow. Large electric boilers are usually of the type employing water as the resistor. Only alternating current can

be used, as direct current would cause electrolytic deterioration. Large boilers of this kind have electrodes immersed in the water where heat is generated directly. Electric steam boilers are useful in industrial plants which require limited amounts of steam for local processes, and 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.

ELECTRIC 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. However, 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 may bring electric water heating systems into general use. This type of system, known as the off-peak hot water storage heating system, offers striking possibilities in the Southern and Pacific Coast States where the degree-day heating load is low and the Utilities have developed or are now developing large blocks of hydro-electric power which can be marketed at 1 cent per kw-hr or less.

In this system of heating, the primary storage tank is simply a large, well-insulated, pressure type steel tank, equipped with immersion electric heating elements connected to line with automatic time switches, which also have automatic limit controls for temperature and pressure. The heating system installed in the home or other type building may be of the individual radiator type or fan-served indirect type, with provisions for the heating and humidification phases of an air conditioning system.

The following types of off-peak heating systems are used:

1. Hot water with gravity circulation.
2. Hot water with forced circulation.
3. Warm air utilizing forced circulation.
4. Unit heaters utilizing hot water as the heating medium.
5. Warm air utilizing stored heat in connection with waste heat.

The essential parts of all types of off-peak storage systems are:

1. Heavily insulated storage tank.
2. Immersion heating elements.
3. Automatic charging control.
4. Heat distributing system.
5. Automatic temperature control.

The insulated storage tank, the heating elements and the charging control are practically identical for all types of off-peak systems. Only the heat distributing system and the room temperature control differ to any extent.

The size of the storage tank is based upon the heat losses of the building, the heating schedule, the hours of heating from storage, and allowance for heating up. In new buildings and where openings permit, factory fabricated tanks are used. Otherwise, the tanks are welded at the final location.

In *hot water systems*, the storage tank supplies hot water to standard heating units (radiators or convectors) installed in the building. The water in the storage tank is heated to approximately 250 F. In the system using gravity circulation, the temperature of the hot water supplied to the heating units is varied by the action of a mixing valve under control of the room thermostat. In larger installations, the circulation is obtained by means of a pump under thermostatic control. In this case, the temperature of the water supplied to the heating units is regulated by mixing the proper amount of return water with the 250 F water from the storage tank through a suitable by-pass.

The *warm air system* makes use of extended heating surface and a plenum chamber assembly (*i.e.*, a central fan system) which is usually installed in the basement alongside the storage tank. The warm air supply ducts are run in the usual manner from the top of the plenum chamber to the registers in the rooms. The return air is brought back through cold air ducts to the inlet side of the fan or blower. Temperature regulation is obtained by thermostatic control of the blower and the valve admitting hot water to the heating units. Air cleaning and humidifying apparatus is readily incorporated in the system.

Unit heaters are used in connection with off-peak storage heating systems for industrial buildings. For sub-stations and certain types of industrial buildings, the electric storage heater is used to supply hot water to unit heaters placed in various parts of the building. Certain sections of the building, such as offices and rooms which are partitioned off from the larger areas, may be heated by ducts leading from indirect heating units supplied with hot water in parallel with the unit heaters.

Warm air supplied from off-peak storage is sometimes used in conjunction with a system designed to utilize waste heat. In this application, heating units are installed in the waste heat ducts, which supply waste heat from the losses of synchronous condensers and other apparatus. When the waste heat does not provide sufficient heating, the air temperature is boosted by supplying the proper amount of hot water from the storage tank to the heating units. Controls are provided, which not only make the entire operation automatic, but insure maximum recovery from the waste heat source.

ELECTRIC WATER HEATING FOR DOMESTIC SUPPLY

Electric water heaters of the automatic storage type for domestic hot water supply are simple and reliable, and in many sections of the country very low electric rates have been established by the electric utilities to secure this load. In some districts, 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 out, without too frequent demands on the regular heating element which takes the high domestic lighting service rate. Some utilities now offer a schedule which, beyond a stipulated minimum, lowers the rate for all service if an electric water heater is installed.

INDUSTRIAL ELECTRICAL HEATING

Electric heating elements have been successfully developed for innumerable industrial furnaces such as annealing, brazing, carburizing, enameling, forging, ceramic firing, hardening, metal melting, nitriding and process heating. Industrial ovens where precise control of high temperatures is necessary can be very successfully operated with electric resistance elements where temperatures as high as 700 F are required. Electric heaters for heating oil to high temperatures for secondary circulation in process work are used as a substitute for superheated steam. Special oil can be electrically heated as high as 600 F and pumped at a pressure just sufficient to cause flow. When used in heating coils or jacketed vessels, this gives a safe and convenient automatic system for moderate-sized installations.

COOLING AND REVERSED CYCLE HEATING BY ELECTRICAL REFRIGERATION¹

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.

For a detailed description of this cycle see Chapter 2 on Refrigeration.

¹The 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).

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).

Edison Building Heated and Cooled by Electricity, by H. L. Doolittle (*Power*, Vol. 74, p. 384-351, September 8, 1931).

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.

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).

AUXILIARY ELECTRIC HEATING

In conjunction with heating systems of other types, an auxiliary electrical heating arrangement is a convenient means of caring for mild days in the spring and fall which require little heat to make a house or building comfortable. Likewise, such electrical 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.

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 cared for electrically.

CONTROL

Because the efficiency of electric heat production is the same for large or small units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Wherever radiant heaters are used, thermostats can be used which operate through changes in air temperature but such thermostats can not integrate the combination effect of the radiant and convected heat. More accurate control can be had by the eupatheoscope, described in the Chapter 38 on Radiant Heating. For all convection and fan circulation heaters thermostatic control is useful. Heaters up to 5000 watts on 230 volt current can be thermostatically controlled by direct acting wall thermostats. For larger size heating elements the thermostat should operate a relay which in turn interrupts or closes the circuit to the heating element.

CALCULATING CAPACITIES

The methods of calculating heat losses outlined in Chapters 6, 7, and 8 may be used for electric heating exactly as for fuel heating. The total heat requirements in Btu per hour may then be converted into the electrical rating of an equivalent heating system by using the equation:

$$\frac{\text{Total Btu per hour}}{3415} = \text{kw rating of required electric heating} \quad (1)$$

For comparison with steam radiation:

$$\frac{3415 \text{ Btu (one kw hr)}}{240} = 14.2 \text{ sq ft of steam radiation}$$

While many empirical rules based on cubic contents, floor areas, etc., are used in steam heating practice, they should never be used to determine size of equipment for an electrical heating installation.

For electrical heating actual heat losses should be figured accurately because the utility company rates and minimum charge items are commonly based on the size of equipment installed, so that cost of operation might be penalized by rule-of-thumb sizing of the heating equipment.

POWER PROBLEMS

The first point to determine is the cost of the power which is available for electric heating. Unlike fuels, there is no uniform cost for electric power because of the unequal cost of distribution to 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 power plants at uniform loads; hence, even the time of use may affect the cost of power.

Homes are almost universally supplied with lighting current of 115 volts, which cannot be used economically for any but the smallest heaters. Usually the service lines will not permit more than plug-in devices. The underwriters permit heaters of 1250 watts to be used from approved baseboard receptacles. Where homes have 230 volt service for cooking and water heating, and rates are favorable, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 230, 460 or 575 volts. All polyphase heaters should be balanced between phases.

INSULATION

The value of building construction which incorporates built-in insulation to reduce the outward heat loss in winter and the inward heat gain in summer has been placed in the spot light by the increasing adoption of complete air conditioning. With electric heating, adequate insulation is very important. Two groups of 16 homes each, in Southern California, are compared in the following actual figures taken from operating records of not less than two years, and in some cases up to eight years.

ITEM	INSULATED HOMES	UNINSULATED HOMES
Average Size of Home: Number of Rooms.....	8.7	8.7
Number of Bathrooms.....	2.6	2.3
Average kw Connected Load per Home:		
Range.....	9.6	9.3
Water Heater.....	6.3	5.9
Heating.....	34.0	32.7
Total kw.....	49.9	47.9
Annual kwhr for Heat per Home.....	4562	8223
Annual kwhr per kw Heat Connected.....	134	252

Cost of Heating

If the consumer obtains heating at 2 cents per kwhr his annual cost of heating an insulated home would be \$91.24, as compared with \$164.46 for heating an uninsulated home. Thus, insulation in the average home creates an annual saving of \$73.22, representing 45 per cent of the annual heating cost if the house is not insulated.

Cost of Insulation

When the average cost of insulating a home is from \$25.00 to \$50.00 per room, or from \$200.00 to \$400.00 for a home of the size used herein,

it is therefore apparent that savings in heating costs will pay for the cost of insulation in from 3 to 6 years. In case the insulation warrants a reduction in connected load, the subsequent reduction in installation cost of heating equipment can be applied against insulating costs which would materially reduce total cost.

ELECTRIC HEATING DATA

Electric heater capacity is rated in kilowatts (kw). Electric energy is measured in kilowatt-hours (kwhr). Cost of operation = kw rating × hours used × cost per kwhr.

One boiler horsepower (bhp) = 33,471.9 Btu per hour.

One kilowatt-hour (kwhr) = 3,415 Btu.

One boiler horsepower = $\frac{33,471.9}{3,415} = 9.80$ kwhr.

One boiler horsepower will evaporate 34.5 lb water per hour from and at 212 F.

One kilowatt-hour = $\frac{34.5}{9.80} = 3.52$ lb of water per hour at 212 F.

Additional conversion factors are given in Chapter 44.

PROBLEMS IN PRACTICE**1 • Under what conditions are electric heaters most feasible?**

- In climates such as in the South and in California, where they are used economically for many heating purposes.
- As auxiliary to central steam or hot water heating plants.
 - Fall and spring.
 - During peak loads.
 - During shut-down periods.
- In factories, offices, etc., where they have a large minimum load charge for electrical power due to large size of connected motors that they cannot use up but must pay for regardless of non-use.

2 • On what basis should electric heating costs be compared to heating with fuels?

- Use.
- Safety.
- Rates.
- Locality.
- Initial investment:
 - Interest.
 - Depreciation.
- Ease of serving electric heaters.
- Ease of control.

3 • Approximately how low must the rates be to permit the use of electricity for heating purposes?

Probably the energy must sell for 2 cents or less per kwhr. At 2 cents the cost would be \$5.86 per 1000 Mbh. (See Chapter 29 for comparison with other fuels.) This looks high, but the seasonal energy consumption would not be as large with electricity as with other fuels, for reasons stated in Question 1.

4 ● In fan heating systems, what is an important difference between a steam heated coil and an electrically heated coil?

A coil supplied with steam at constant pressure will remain at constant temperature regardless of the amount of air passing over it. The temperature of the electric coil supplied with a constant amount of energy will rise if the air quantity is decreased and fall if the air quantity is increased.

Chapter 40

AIR CONDITIONING FOR INDUSTRIAL PROCESSES

Moisture Content and Regain, Typical Industries Requiring Air Conditioning, Classification of Problems, Control of Regain, Control of Rate of Chemical Reaction, Control of Rate of Bio-chemical Reaction, Control of Rate of Crystallization, Conditioning and Drying, Atmospheric Conditions Required, General Requirements, Air Conditioning of Libraries, Greenhouse Heating

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 most materials of vegetable or animal origin, and to a lesser extent minerals in certain forms, take moisture from or give it up to the surrounding air.

In industries where the physical properties of the product affect value, the question 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 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

TABLE 1. 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 Fibres	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	Fibre	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	
Food-stuffs	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 Fibre	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 ₂ SO ₄	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason	

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 constant 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 necessarily imply that the material as a whole has been completely dried out and has re-absorbed moisture. In the case 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. In measuring moisture it is necessary to dry out a sample so that the loss in weight may be used as a basis for calculating the regain of the whole lot.

The moisture content of an 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 different percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying out varies with the nature of the material, its thickness and density.

Table 1 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 deg 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.

TYPICAL INDUSTRIES REQUIRING AIR CONDITIONING

A few of the industries in which air conditioning plays an important part and the major uses in these industries are as follows:

Automobile. Drying of siccative coatings, manufacture of steel, manufacture of artificial leather, drying of rubber, manufacture of tire fabrics, cementing of inner tubes, conditioning of wooden spokes and other wooden parts, conditioning and manufacturing of all electrical windings in connection with the electrical apparatus, storage battery plates and the rubber containers.

Bakery. Flour storage, yeast and ingredient storage, mixers, fermentation rooms, make-up room, proof boxes, loaf cooling, wrapping (including paraffin paper), cake mixing and cake icing.

Brewery. Fermentation and starting rooms.

Chemical. Powders (including explosives and baking powder), drying of salts of all kinds, hygroscopic compounds and drugs, glues and gelatines.

Clay Products. Bricks, pottery and ceramics.

Confectionery. Chocolates, bon bons, hard candy, gum drops, marshmallows, caramels, chewing gum and starch and various sugars.

Drugs and Pharmaceuticals. Drugs and pharmaceuticals might also be included under chemicals, but definitely to be added to this group are capsules, hygroscopic colloidal crystals, serums and toxins.

Electrical Goods. Toll cable manufacture, telephone exchanges, winding rooms, lamp manufacturer and filament departments.

Films and Film Laboratories. Drying cabinets, printing rooms, perforating rooms, projection assembly rooms, moving picture studios, celluloid and color photography.

Foods. Bread and cake, cereals, macaroni, meats (cold storage markets), yeast, enzymic products, fruits, including apples and bananas, both for preserving and ripening.

Furs. Fur storage.

Incubators. Human babies, chickens and similar hatching.

Laboratories. All kinds.

Leather. Drying and processing of hides, skins and manufacture of bags, shoes and findings.

Linoleum. Drying, printing, oil cloth, and linseed oil buildings.

Matches. Storage of raw materials, machine drying and packing.

Minerals. Gold beater rooms, gold and silver leaf manufacturing, metal enameling, and mottled ware, particularly all cutting on iron.

Paper and Paper Products. Moisture absorption in manufacture, cutting, folding, binding and furnishing bags, including gluing, parchment paper, cellophane containers, paste board containers, paste board bottles and egg containers.

Pearls. Artificial pearls.

Printing, Lithography and Rotogravure. Playing cards, process work, storage, offset work, binding, rollers and ink.

Soap. Crystallizing under the cold process.

Textiles. Cotton: drying, spinning and weaving. Rayon: chemical house, spinning, drying, twisting, reeling, winding, inspection and storage. Silk: storage, twisting and reeling, spinning, weaving, knitting, tin and lead weighing and regain rooms (hosiery and underwear).

Tobacco. Cigarettes: storage, mixing, blending, paper and machine manufacture. Cigars: storage, curing, cleaning, wrapping and packing.

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 book. A few of the salient points of the general subject are covered in this chapter.

CLASSIFICATION OF PROBLEMS

The problems of industrial air conditioning fall into four general classes:

1. Control of Regain.
2. Control of Rate of Chemical Reactions.
3. Control of Rate of Biochemical Reactions.
4. Control of Rate of Crystallization.

CONTROL OF REGAIN

In the first class the textile plant offers a good example. 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 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 process in a cotton mill, the fibers should be in a condition easily to be brought parallel to each other.

A relative humidity of 50 to 55 per cent gives the best result for these preliminary processes. 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. As pointed out in a paper presented before the *Cotton Manufacturers Association* in 1926, a higher relative humidity is necessary to offset the extra work performed on the fiber as the spindle speed was increased. Today many cotton mills carry 70 per cent relative humidity in the spinning rooms. 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 57 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 absorbing moisture, release sensible heat equivalent to the latent heat of the moisture taken up by the material. This may account for a large percentage of the total load.

CONTROL OF RATE OF CHEMICAL REACTION

Typical examples of the second classification, the control of the rate of chemical reactions, occur in the manufacture of rayon. The pulp sheets are conditioned, cut to size, and passed through a mercerizing process. It is essential that this be under close control of both temperature and relative humidity. Temperature controls the rate of reaction directly, while the relative humidity maintains a constant rate of evaporation from the surface of the solution and gives a solution of known strength throughout the mercerizing period.

Another well known example of this class is the *drying* of varnish which is an oxydizing process dependent upon temperature. High relative humidities have a retarding action on the rate of oxydization at the surface and allow the gases to escape as the chemical oxydizers *cure* the varnish film from the bottom. This produces a surface free from bubbles and a film homogeneous throughout.

Temperatures for *drying* varnish vary with the type. A relative humidity of 65 per cent is beneficial. In the field of biochemical control, industrial air conditioning has been applied to so many different and well known products that it is difficult to select an outstanding example.

CONTROL OF RATE OF BIOCHEMICAL REACTIONS

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 with a relative humidity of 65 per cent to hold the surface of the dough open, so that the CO_2 gases formed by the fermentation may pass out and produce a loaf of bread, when baked, of even, fine texture without large voids.

Another example of a similar nature 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 come 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 must be changed and the skin cured and colored. Then the fruit is cooled to maintain as slow 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 become 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 F to 59 F have been found to be best suited for this process. A high relative humidity, 88 to 90 per cent is necessary to hold shrinkage to a minimum and, at the same time, develop the rind so that it will be sufficiently tough to stand 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 industrial 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 temperature and relative humidity 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 kettles 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 temperature and relative humidity. If the cooling and drying is too slow, the coating will be rough and semi-translucent, and the appearance unsightly; if too fast, the coating will chip thru to the interior. Only by balancing temperature, relative humidity, and volume of air to the sugar solution can the proper rate be obtained and a perfect coating assured.

The foregoing is presented as typical of a few of the problems met with in industrial air conditioning. They are far from complete but, with the

help of a few of the natural laws, may act as a stimulus to the imagination and assist in solving others.

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. Either conditioning or drying are frequently made continuous processes in which the material is conveyed through an elongated compartment by suitable means.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity during processing depends upon the product and the nature of the process. As far as the behavior of the material itself and its desired final condition are concerned, each material and process represents a different problem. The best relative humidity may range up to 100 per cent. Similarly the most desirable temperature may range between wide limits for different materials and treatments. Extremes in either relative humidity or temperature require relatively expensive equipment for maintaining these conditions and controlling them automatically. Also, in departments where people are working, their health, comfort, and productive efficiency must be considered. A compromise often is desirable.

It is generally considered that relative humidities below 40 per cent are on the dry side, conducive to low regains, a brittle condition of fibrous materials, prevalence of static electricity, and a tendency toward dryness of the skin and membranes of human beings. At the other end of the scale, humidities above 80 per cent are relatively damp, conducive to high regains, extreme softness, and pliability.

Table 2 lists desirable temperatures and humidities for industrial processing. In using this table, care must be taken in qualifying the process. In preparing many materials, conditions are not maintained constantly, but different temperatures and humidities are held for varying lengths of time.

GENERAL REQUIREMENTS

In general, 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 reaction; of warming or cooling to any desired degree, and of giving or permitting ample air supply at all times. Refrigeration may or may not be required, depending upon natural conditions, the required relative humidity and the maximum

TABLE 2. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
AUTOMOBILE	Assembly line	65	40
BAKING	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	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	80 to 90	80 to 95
	Storage of flour	70 to 80	60
	Storage of yeast	28 to 40	60 to 75
BIOLOGICAL PRODUCTS	Vaccines	below 32	
	Antitoxins	38 to 42	
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	35
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	45
	Storage of grains	60	30 to 45
DRUG	Storage of powders and tablets	70 to 80	30 to 35
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	0 to 10	50
	Storage of sugar	80	35
FUR	Drying of furs	110	
	Storage of furs	28 to 40	25 to 40

TABLE 2. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING
(Continued)

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
INCUBATORS	Chicken	99 to 102	55 to 75
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	
LIBRARY	Book storage (see discussion in this chapter)	65 to 70	38 to 50
LINOLEUM	Printing	80	40
MATCH	Manufacturing	72 to 74	
	Storage of matches	60	50
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	25 to 50
	Storage of paper	60 to 80	35 to 45
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)	60 to 75	20 to 60
	Storage of rollers	60 to 80	35 to 45
RUBBER	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
SOAP	Drying	110	70
TEXTILE	Cotton— carding	75 to 80	50
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	60 to 70
	weaving	68 to 75	70 to 80
	Rayon— spinning	70	85
	twisting	70	65
	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
TOBACCO	Cigar and cigarette making	70 to 75	55 to 65
	Softening	90	85
	Stemming or stripping	75 to 85	70

permissible temperature. Washing, purifying and recirculating of the air may be desirable. Good distribution is essential to the control of air motion and for the prevention of undesirable conditions. Accurate, sensitive and reliable automatic control of humidity or temperature, or both, is vital in most cases.

Ordinarily, 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 used for driving 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 a normal degree of 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.

In the general application of industrial air conditioning, the conditions to be maintained are governed almost entirely by the requirements of the product. If any consideration is to be given to the comfort and, therefore, the efficiency of the occupants, it is secondary. In a great many cases, the requirements of the product must necessarily govern, for the physical properties of the material are more important for maximum production than the efficiency of the worker. There are, however, many cases where the worker can and should be given equal consideration and better overall results may be obtained by a proper compromise.

AIR CONDITIONING OF LIBRARIES¹

Temperature has little effect on the preservation of books. A temperature over 100 F, combined with low relative humidity, may cause the book materials to become brittle, while a temperature much below freezing may cause permanent deterioration of the glue in the binding. The relative humidity should be maintained between 40 and 70 per cent, although these limits need not hold for short periods of time. If the relative humidity gets much below 40 per cent, first the glue and then the paper will tend to become brittle which will not cause any permanent damage unless the book is used while in this condition, as a subsequent increase in humidity will bring the materials back to their normal condition. If the relative humidity gets above 80 per cent, the growth of mildew may be expected.

One of the principal agents of destruction and deterioration of paper

¹See U. S. Bureau of Standards Bulletin No. 128 entitled A Survey of Storage Conditions in Libraries, by Kimberly and Hicks.

and books in libraries is sulphur dioxide gas in the air. If air containing sulphur dioxide is allowed to come in contact with cellulose, the principal constituent of paper, sulphuric acid is formed on the surface. This acid is not volatile at ordinary temperatures and therefore accumulates throughout the life of the paper. The destructive effect of the acid on the paper is independent of the relative humidity of the surrounding air. Low alkaline concentration spray water may be used in an air washer to neutralize the acid condition. Such an air washer must be especially constructed to resist corrosion.

GREENHOUSES

Table 3 lists customary dry-bulb temperature ranges for different types of plants and flowers raised in greenhouses.

TABLE 3. CUSTOMARY TEMPERATURES FOR DIFFERENT TYPES OF GREENHOUSES^a

TYPE OF HOUSE	TEMPERATURE RANGE DEG F	TYPE OF HOUSE	TEMPERATURE RANGE DEG F
Carnation.....	55 to 65	Orchid, cool.....	50 to 60
Conservatory (general collection).....	65 to 70	Palm, warm.....	65 to 75
Cool.....	50 to 60	Palm, cool.....	55 to 65
Cucumber.....	65 to 70	Propagating.....	55 to 60
Fern, Common.....	60 to 65	Propagating, Bottom heat.....	70 to 75
Fern, Tropical.....	65 to 75	Rose.....	65 to 70
Forcing.....	60 to 70	Sweet pea.....	50 to 60
General purpose.....	60 to 70	Tomato.....	65 to 70
Lettuce.....	55 to 65	Tropical.....	65 to 75
Orchid, warm.....	65 to 75	Violet.....	45 to 55

^aLower values may be considered approximate night temperatures and higher values approximate day temperatures.

PROBLEMS IN PRACTICE

1 ● Why is air conditioning required for some industrial processes?

To control the physical properties of the materials being processed.

Example. In the manufacture of chewing gum, it is rolled into slabs and scored. The scored slab must then be broken at the score marks to form the sticks. If the slab is too warm, breaking is impossible, if the slab is too cold or too dry, it becomes brittle and shatters, thereby producing much material to be reworked.

2 ● Why is it necessary to control the physical properties of the material being processed?

To permit permanent adjustment of machinery.

Example. In the manufacture of cigarettes, the amount of tobacco fed upon the paper tape is determined by pressure against springs. When the tobacco is over-moist and, therefore, over-soft, a great excess will go into the finished cigarette; when the tobacco is too dry and, therefore, harsh, too little goes into the finished cigarette.

3 ● A condition of 75 F dry-bulb temperature and 55 per cent relative humidity is being maintained in a cigarette manufacturing department. What will be the regain and moisture content of the tobacco?

The regain, from Table 1 = 17.75 per cent.

The moisture content = $\frac{17.75 \times 100}{100 + 17.75} = 15.1$ per cent.

4 • A 1-lb sample taken from a 100-lb batch of material is found to have a bone dry weight of 0.89 lb. This material is to be processed under atmospheric conditions which should produce a regain of 15 per cent. Compute the finished weight for each original 100-lb batch.

Let W equal the number of pounds of moisture in a finished batch.

$$\frac{W}{89} = \text{regain} = 15 \text{ per cent} = \frac{15}{100}$$

$$W = 13.35$$

$$89 + 13.35 = 102.35 \text{ lb finished weight.}$$

5 • A bundle of sea island cotton is found to have a bone dry weight of 9.26 lb. What is the proper relative humidity at 75 F to produce a weight of 10 lb at equilibrium?

$$\begin{aligned} \text{Desired conditioned weight} &= 10.00 \text{ lb} \\ \text{Bone dry weight} &= 9.26 \text{ lb} \end{aligned}$$

$$\text{Weight of moisture required} = 0.74 \text{ lb}$$

$$\text{Regain} = \frac{0.74}{9.26} \times 100 = 7.9 \text{ per cent.}$$

From Table 1, the proper relative humidity required is 60 per cent.

6 • Compute the bone dry weight of 1000 lb of manila rope which has been stored for a considerable period of time in a conditioned room at 75 F dry-bulb temperature and 50 per cent relative humidity.

Assuming that this material has come to equilibrium under the atmospheric conditions given, Table 1 shows a regain of 8.5 per cent.

Let W equal the total weight of moisture in pounds.

1000 - W = bone dry weight in pounds.

$$\frac{W}{1000 - W} = \text{regain} = 8.5 \text{ per cent} = \frac{8.5}{100}$$

$$W = 78.3 \text{ lb moisture}$$

$$1000 - 78.3 = 921.7 \text{ lb bone dry weight.}$$

7 • An egg evaporating plant wishes to dry 2000 lb of egg whites (85 per cent water) to crystalline form each 24 hours. The maximum permissible air delivery temperature in the dryer is 140 F. What air volume will be required, assuming that outside air is at 95 F dry-bulb and 78 F wet-bulb and that air leaves the dryer 70 per cent saturated?

Moisture to be removed = $2000 \times 0.85 = 1700$ lb. Using psychrometric chart and starting at the intersection of the vertical 95 F dry-bulb temperature line and the 45 per cent humidity line, move horizontally to the right to the intersection with the 140 F vertical temperature line at 10 per cent relative humidity; then move along the constant heat (or wet-bulb line) to its intersection with the 70 per cent relative humidity curve and read 94 F dry-bulb, which will be the temperature of the air leaving the dryer.

$$\begin{aligned} \text{Moisture per cubic foot at 94 F and 70 per cent relative humidity} &= 11.8 \text{ grains} \\ \text{Moisture per cubic foot at 95 F and 78 F wet-bulb} &= 8.0 \text{ grains} \end{aligned}$$

$$\text{Moisture added per cubic foot of air handled} = 3.8 \text{ grains}$$

$$\frac{1700 \times 7000}{24 \times 60 \times 3.8} = 2170 \text{ cfm.}$$

No allowance is made for heat lost in the transmission to and from the dryer or for the heat required to raise the product from its entering temperature to that maintained in the dryer. This would necessitate a trial and error solution common to all drying problems.

Chapter 41

DRYING

Definition, Methods of Drying, Types of Dryer Construction, Mechanism of the Drying, Control of the Drying Operation, Dryer Design, Dryer Arrangement and Construction, Experimental Technique

DEFINITION OF TERMS

THE term drying, in its broader sense, refers to the removal of water or other volatile liquid from a gaseous liquid, or solid material. Except in the case of solids, the term is not ordinarily used unless the water or other liquid removed is present in a relatively small amount. In a more restricted sense, drying is the removal of water by vaporization from a non-volatile liquid or solid.

Where the solid material 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 filtration, settling, pressing, centrifuging, or other mechanical means. 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 in general much less than by vaporization.

METHODS OF DRYING

Classification of Methods According to Heat Supply

Drying methods may be classified basically in accordance with the means by which heat is supplied to the material to be dried. Thus, in some cases, as in *air current dryers*, the heat of vaporization is transferred to the material from the atmosphere surrounding it. In *natural air dryers*, this heat is merely the sensible heat of the outdoor atmosphere; while in so-called *artificial air dryers* heat is added to the air by means of radiators or the addition of a heated gas such as superheated steam or products of combustion, the air being almost entirely displaced by such gases in some drying processes. Where products of combustion can be used, a particularly high overall efficiency of the dryer is possible.

In other types of dryers, known as *contact dryers*, heat is supplied by direct contact between the material and a heated surface such as a pan or shelf, such surfaces being heated by products of combustion, steam, hot water, heated oil, or electrical resistance. In both air dryers and contact dryers, additional heat may be supplied by direct radiation from hot surfaces which are *in sight* of the material. In the *sun drying* of raisins and other fruits, this radiant heat comes from the sun, while in artificial dryers it is derived from heated surfaces such as steam coils.

Radiant Heat

Radiant heat can at best supply only a portion of the total heat of vaporization. About one-third to one-half of the heat given off by a radiating surface passes through the air to the object which is to be dried without materially raising the temperature of the intervening air. The remaining portion of the heat from the radiator warms the air by convection and produces warm air currents which in turn transfer their heat to the material. A given amount of heat can be supplied to the material by means of heated air currents with only one-third to one-tenth as much radiating surface as would be required if this same quantity of heat were to be transferred by direct radiation. Direct radiation is, therefore, only an incidental source of heat in most dryers, except in the drying of certain kinds of material where the advantages of radiant heat will offset the higher first cost of the equipment. In vacuum dryers radiant heat is particularly useful in the drying of some bulky solids, owing to the difficulty of transferring heat to the material by means of rarified air currents at extremely low pressures.

Continuous and Intermittent Operation

Dryers may be operated continuously or intermittently. In the continuous type, the material is fed in at one end of the dryer and discharged at the other end when dry. The movement may be either a steady flow, or may consist of the periodic or progressive removal of a minor portion of the charge at one end of the dryer and the admission of a like portion at the other end, at more or less regular intervals. Dryers of the intermittent type, often called charge or batch dryers, are filled at the start with the material to be dried and are emptied again when the drying is completed, after which the process is repeated. The material in charge dryers is usually stationary but is sometimes shifted in position periodically or kept in continuous motion within the dryer for greater uniformity and speed of drying.

With constant rates of flow of air and material, continuous dryers are usually simple to control and can be fitted advantageously into the train of manufacturing operations with minimum loss of time and requirements for storage and handling. Charge dryers, on the other hand, permit the use of complicated schedules of temperature and humidity which would be difficult to attain with continuous operation; and they have the important advantages of simplicity and low first cost, making them particularly adapted to small installations and experimental work.

Temperature and humidity conditions in a charge dryer should be substantially uniform throughout the dryer, in order to produce uniform drying throughout the charge and prevent injury to the material. These conditions may remain uniform throughout the drying cycle or may be changed periodically as the drying progresses. In continuous or progressive dryers, the drying conditions usually vary from one end of the dryer to the other, either to suit the varying requirements of the different stages of drying or because of different rates of evaporation at different positions in the dryer.

With some materials, such as clay and other plastic material, the usual requirements are for the entering material to encounter the air at the

lowest temperature and highest humidity. As the material dries, it gradually moves forward to higher temperatures and lower humidities and encounters the highest temperature and lowest humidity just before leaving the dryer. This lends itself very nicely to a tunnel dryer with the air movement opposite to the travel of the material as the air cools and picks up moisture, producing the exact effect desired.

Other material, like lumber, requires a high temperature and high humidity at the beginning, and lower temperature and humidity at the end. Such a condition requires very special handling of the air in a continuous dryer, and it is here that a batch dryer is of advantage. The material stands still and the entire dryer can be kept at any desired temperature and humidity, which can be changed at will at any interval of time. In batch dryers it is almost always necessary to recirculate a considerable proportion of the air to conserve the heat and to prevent the air from being too dry. A continuous dryer, on the other hand, often does not require recirculation as the air has been in contact with the material for a longer time, and the increasing humidity and lower temperature encounters material in various stages of drying. This question of recirculation is more fully dealt with later under *Control of the Drying Operation*.

Low and High Temperature Drying

The division line between what is known as low temperature drying and high temperature drying is, roughly speaking the boiling point of water under ordinary atmospheric conditions. In high temperature drying, the temperatures range from 212 to 1000 F or more. Under these conditions, all of the water in the material will vaporize quickly and the humidity of the air becomes of minor importance. High temperature drying, is usually faster and more economical of heat than low temperature drying, but often cannot be used because many materials, particularly foods and other organic substances, are injured or destroyed by temperatures above the boiling point. For such materials low temperature drying is required, the drying being produced by supplying enough heat to the material, and surrounding it with a sufficiently low humidity so as to produce a higher vapor pressure within the material at its surface than is present in the air surrounding it. As long as this difference in vapor pressure is maintained, the drying will proceed until completed.

In drying with temperatures above the boiling point, the choice of means of heating the air becomes more difficult. Exhaust steam at low pressures becomes useless. High pressure steam may be used in coils; either in separate banks near the fans, or as direct radiation. By this means, air may be heated to about 340 F. This does not mean, however, that the dryer will be as high as that. Above these temperatures, hot oil may be circulated in the coils. Occasionally electric heat is used, and in some cases superheated steam has been admitted to dryers requiring high temperatures. The most widely accepted method, however, is to employ oil fired or gas fired air heaters which are now standard equipment in many industries, even when lower temperatures are required. This avoids putting drying loads (which often fluctuate widely) on the boiler plant. They also have the advantage of being independent for isolated locations.

These heaters operate on two broad principles. The *Direct Heaters* mix the products of combustion with the air which is to be blown into the dryer. The amount of mixture is determined by the temperature desired in the mixture. The *Indirect Heater* passes the products of combustion through an interchanger which avoids mixing these gases with the air. With either type heater, multiple type burners are used in order to obtain automatic temperature control without sacrificing good burner performance. The type of fuel used whether gas or oil depends on the locality and the important item of cost. The gas may be natural, manufactured as a by-product of coke, or derived from oil. High heater efficiencies are secured with the proper selection of the type of heater and careful arrangement of the heater in the drying circuit.

Vacuum Drying

It is sometimes advantageous to conduct the drying in a partial vacuum, in order to obtain rapid vaporization at comparatively low temperatures. This method is especially useful for materials which are sensitive to heat, such as valuable food products which must be dried rapidly at low temperatures in order to preserve flavors and prevent harmful organic changes. A vacuum as high as 28 in. of mercury is sometimes used where a very low boiling point is desired, but the usual range is from 12 to 20 in.

In vacuum drying, heat is supplied to the material either by contact with the heated walls or shelves of the container, which is the usual method, or by direct radiation, either from the walls or shelves or from special radiators. Some materials containing only 2 to 3 per cent moisture are first heated before being put in the vacuum chamber. When the vacuum is established, the heat already in the material vaporizes the moisture and becomes latent, thus drying and cooling the material at the same time.

The air within the vacuum container is so rarified as to reduce materially its value as an agent for transferring heat from walls or radiators to the material. The volume of air which must be circulated in a vacuum chamber in order to transfer a given amount of heat is increased several times as compared with air at atmospheric pressures, and the cost of creating a circulation sufficient to transfer any large portion of the heat required for rapid evaporation, is apt to be prohibitive. The evaporated moisture is removed by means of condensers in the vacuum line.

It should be clearly understood that a low temperature of product may not necessarily be secured in a vacuum dryer. If the material is very thin, is dried by radiant heat, and has a large amount of free water, then the temperature of evaporation for the constant rate period will correspond closely with the temperature of water vapor equal to the vacuum maintained. After the critical moisture content is reached, the temperature of the material is increased and approaches the temperature of the radiating or contact surface. With direct contact dryers, and especially with thick materials, the temperature is liable to be increased considerably over that corresponding to the vacuum. In these cases, much of the expected advantages of vacuum drying will be lost. (See later paragraphs on mechanism of drying, and dryer design.)

Adiabatic and Constant Temperature Drying

Air dryers are called *adiabatic* when the only source of heat for drying purposes is the heated air entering the dryer from the outside. When part or all of the heat required is applied by steam coils or other forms of radiation within the dryer itself, it is known as a *constant temperature dryer*. In adiabatic dryers, the temperature of the air decreases, as the drying proceeds, in direct proportion to the amount of heat consumed or made latent by evaporation. If this continued without new heated air being introduced, drying would slow down and finally stop.

The constant temperature dryer does not depend on the air to carry the heat and only enough air need be handled to remove the water evaporated. The great danger in the constant temperature dryer is that local zones in the dryer will vary widely in temperature, and hence, in humidity, while in the adiabatic dryer temperatures and humidities should be quite uniform in a properly designed dryer. In the adiabatic dryer there is ample opportunity to use the zone system whereby the temperature and humidity may be different in various zones. This is often desirable in a continuous dryer where the required condition at the different stages is other than that which would result if the air were allowed to vary in temperature and humidity due to the evaporation which has taken place in the preceding zone.

TYPES OF DRYER CONSTRUCTION

Drum Dryers are used for drying thick liquids or pastes. A steam heated cylinder, with its lower side immersed in a bath of the material, is revolved slowly. The film of material that forms on the hot surface of the drum dries rapidly and the resultant solid is scraped off by a fixed knife edge.

Cylinder or Can Dryers are designed for the continuous removal of moisture from material in the form of thin rolled sheets such as textiles, paper, and pulp products. Heat is transferred to the material by direct contact with a series of steam heated cylinders. In case the material shrinks on drying, as does paper, the drives must be arranged so that the speed of the drums is gradually less toward the dry end of the series. These dryers can be much improved as to speed and capacity by the application of jets of heated air to absorb the moisture as it leaves the material. This absorption of the vapor also prevents condensation on the ceiling and aids greatly in ventilating the dryer room.

Pan or Agitator Dryers are used for sludges, powders, or material in lump form and consist of a cylindrical metal shell. The material to be dried is placed in the bottom of the shell and is constantly shifted in position by means of agitators, either the shell or the agitators or both being in motion. Warm air, gas, or products of combustion, are passed over the material and additional heat is usually supplied from a steam heated jacket around the container, or by steam supplied to hollow stirring arms. Continuous agitation of the charge serves not only to expose new surfaces to contact with the air, but also to reduce the average distance from the interior to the surface of the solid and so to speed up the drying where the rate of liquid diffusion to the surface is the controlling factor.

Rotary Dryers are suited particularly to the continuous drying of granular, crystalline, or lumpy material which does not tend to ball or stick together. They consist of a revolving cylindrical shell into which the fresh material is fed at one end and from which the dry material is removed at the other end. Lifting plates usually extend from end to end of the inside of the cylinder, parallel to its axis, projecting radially toward the center. Warm air or products of combustion are blown through the shell, either parallel or counter current to the direction of movement of the material. The rotation of the cylinder continually elevates the material and drops or throws it through the current of hot gas or air, the inclination of the shell moving the charge forward at any desired speed. To furnish additional heat to the material by contact, the cylinder is often provided with a steam jacket or with steam coils inside the cylinder. In drying material which is not injured by high temperatures, the outside of the shell may be heated by products of combustion and these gases then led through the dryer in contact with the material.

Spray Dryers

Spray Dryers are used for the drying of thick liquids, including any product in solution, suspension or emulsion, such as milk, rubber latex, chemicals and soap where the desired product is a powder or fine uniform particles. The liquid is usually first concentrated to from 35 to 50 per cent solid content and is then atomized in the form of a fine spray at the top of a lofty chamber, falling through a current of heated air or furnace gases, the dried powder accumulating at the bottom of the chamber. The air can be heated to a considerably higher temperature than would be used for the same material in a contact dryer because the critical temperature in a spray dryer is the wet bulb temperature instead of the dry-bulb temperature. The drying is practically instantaneous. The spraying may be done by atomizing nozzles worked either by the pressure of the liquid or by compressed air, or the liquid may be dropped in a fine stream onto a revolving disc rotating at high speed. That portion of the powder which is carried away with the air currents may be recovered in anyone of several types of dust collectors such as cyclones, screens, dust bags or electric precipitators. The air or gas may be recirculated in part, after removing surplus moisture.

Loft, Compartment, Cabinet or Room Dryers consist of room-like enclosures or boxes in which the material to be dried is placed on trays, trucks, racks, or moving conveyors and is heated by warm air currents which also serve to carry away the evaporated moisture. Drying of this sort is sometimes called *air-processing*. The air is warmed, either by direct mixture with furnace gases, or by contact with surfaces heated by direct flame, flue gases, steam, hot water, or electricity. The source of heat may be *direct*—located within the dryer, usually in the form of wall coils, floor coils, or distributed coils—or *indirect*, the air being heated by radiators outside the dryer and circulated through the dryer by fans and distributing ducts. Baffles are often used to assure the passage of the heated air directly over the surface of the material and to prevent short-circuiting from inlet to outlet. Where the material is placed on trays, shelves or racks, it is advisable to use open-wire screen bottoms for small material or leave spaces between larger articles when possible, to allow

the air to travel not only over the material but also upward or downward through it.

Tunnel Dryers consist of a comparatively long, narrow box-like enclosure, through which the solid to be dried is moved either continuously or periodically at short intervals, being fed in at one end in a wet condition and removed from the other end in a dry condition. The material may be carried on some type of mechanical conveyor or on wheeled cars or trucks moving on a track. The drying is done by means of heated air which moves through the dryer from one end to the other, warming the material and removing the evaporated moisture. Some products dry best if the air and material are made to move in the same direction, but in most cases it is found best to have the air move in a direction opposite to the travel of the material, that is, counter-currently. Where wheeled cars are used for heavy, bulky products, the rails in the tunnel are often sloped in a ratio of 60 to 1, or 100 to 1, to obtain the benefit of gravity in moving the material.

One end of the tunnel is usually hotter than the other, to suit the varying requirements of the different stages of drying. In the natural draft tunnel, heat is provided at the hotter end by steam coils or other means, and the evaporated moisture is removed at the opposite end by means of ventilating chimneys. In the forced draft, or blower type of tunnel, the air is circulated through the tunnel by fans, being heated either by direct radiation within the tunnel, or indirect radiation in the fan duct, or a combination of both. Evaporated moisture may be removed either by ventilating chimneys or duct openings, or more rarely, by condensers in the duct system.

Electric Induction Dryers

Some new developments are occurring in the use of electricity for drying or baking metal parts. Among these is a system whereby metal parts are passed through a high frequency field, thereby inducing currents in these parts. Heat is then generated within the parts themselves, baking *from the inside out*.

The advantage of this system is a more efficient use of electricity, since oven temperatures are lower than work temperatures, keeping radiation and ventilation losses down to a minimum. This system can be applied only to production lines turning out uniform objects. The apparatus required, in addition to the conventional oven, is a high frequency motor generator set and a system of coils within the oven to obtain the proper electric field.

MECHANISM OF THE DRYING

Stages of Moisture Diffusion

A thorough knowledge of the mechanism of the diffusion of moisture from the interior of a solid material to the surface and into the air is necessary to an understanding of the different types of commercial dryers, and is essential where the design of the dryer is to be based upon small scale laboratory tests. Assuming uniform velocity and distribution of air at a constant temperature and humidity, over the surface of the solid, the drying cycle will be divided ordinarily into two distinct stages. The first stage, known as the *constant rate period*, is found in the drying of any

very wet solid as long as moisture is brought to the surface so rapidly that the surface remains thoroughly wet and evaporation can proceed at a constant rate, precisely as from a free water surface. The second stage, known as the *falling rate period*, is reached when the moisture in the material has been dried down to a point called the critical moisture content, below which the diffusion of moisture from the interior to the surface is no longer adequate to keep the surface thoroughly wetted. From this point on, the rate of drying will decrease until the operation is completed.

Constant Rate Period

During the constant rate period, the surface is kept saturated with water by diffusion from the interior and the operation is called *saturated surface drying*. The rate of drying is limited by the rate of diffusion of water vapor, through the surface air film surrounding the solid, out into the main body of the air. When the heat necessary for vaporization is supplied only by thermal conduction through the surface air film, the temperature of the solid is the wet-bulb temperature of the air. Heat gained by radiation or conduction from adjoining dry surfaces raises the temperature of the surface above the wet-bulb temperature. The vapor pressure of the water at the surface, therefore, increases and consequently the driving force causing diffusion through the surface air film becomes greater, causing a corresponding increase in the drying rate.

By placing the wet material in sight of hot surfaces during this constant rate period, the rate of drying may be increased several fold without overheating the material itself. Similarly, material placed on hot shelves, as well as material dried in shallow pans or on heated cylinders, will dry faster because of the heat received by conduction from the surfaces with which it is in contact. A controlling factor during this stage is the velocity of the air, affecting as it does the thickness of the surface air film through which the water vapor must diffuse. The higher the velocity, the faster will the drying proceed, the rate of drying varying approximately as the 0.6 power of the air velocity.

Falling Rate Period

After the critical moisture content of the material has been reached, two distinctly different forms of diffusion and evaporation are encountered. In both cases, the drying rate falls off rapidly with reduction of the moisture content. In the first type, known as *unsaturated surface drying*, the rate of evaporation per unit surface area is substantially constant and nearly independent of the thickness of the sheet or layer of the material. However, where the solid gains its heat only from the air around it and obtains none by radiation or conduction from adjacent surfaces, the transfer of heat from the air to the solid is, under constant drying conditions, found to be independent of the moisture content and of any variation in the drying rate due thereto. Furthermore, the drying rate is increased by greater air velocity and by lessened humidity of the drying air. These facts are explained by assuming that evaporation takes place on the surface of the solid, but that the concentration of moisture there is insufficient to saturate the whole surface, so that the surface behaves as though only a part of it were wet.

In the second type of diffusion, known as *sub-surface drying*, the rate of evaporation per unit surface area of the sheet of material is inversely proportional to the thickness and is uninfluenced either by the velocity or the humidity of the surrounding air. The heat transfer from the air to the stock falls off rapidly as the moisture content and the drying rate decrease. This indicates that evaporation is taking place, not at the surface of the solid, but beneath the surface, so that water as vapor must diffuse not only through the gas film around the solid, but also through that layer of the solid itself between the zone of evaporation and the surface. Consequently, this type of evaporation is known as sub-surface drying, the

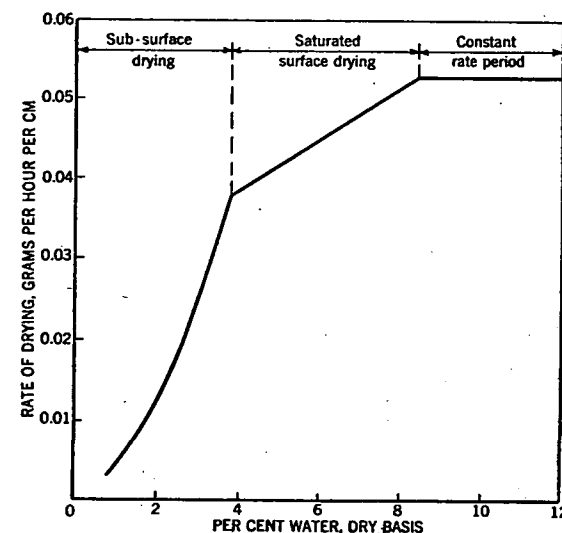


FIG. 1. RATE OF DRYING OF WHITING SLAB

limiting factor in the rate of drying being the speed of diffusion of the liquid water from the interior of the solid to the zone of evaporation.

The Drying Cycle

In the drying of a typical wet solid under constant conditions, saturated surface evaporation usually starts at a constant rate. This will continue until the critical moisture content of the stock is reached, after which there will follow a period of unsaturated surface drying, at a falling rate. This is finally superseded by sub-surface drying which continues at a decreasing rate until the operation is completed. The drying of a slab of whiting gives a typical illustration of this cycle, as shown in Fig. 1.

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 stage 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 solely in practice, to unsaturated surface drying.

Capillary and Hygroscopic Moisture

Most solid materials to be dried are of a powdery, granular, cellular, or fibrous nature and contain water in two characteristic forms which have a direct relation to the rate and character of the drying operation. The first form is known as *free* or capillary moisture and comprises the water which is contained in the capillary spaces between the particles or fibres of the material. The second form is called *adsorbed* or hygroscopic moisture and is intimately associated with the physical nature of the material, having a direct effect upon such physical properties as size, strength, electrical conduction, heat conduction, etc. Removal of the capillary water has little or no effect upon the material except to reduce its weight, while removal of the hygroscopic water causes definite changes in physical properties and characteristics.

Fiber Saturation Point

The total amount of adsorbed or hygroscopic moisture that a given amount of material can contain is definitely limited. This limit is known as the *fiber saturation point*¹ and corresponds roughly to the critical moisture content. Beyond this point, any additional moisture must be in free or capillary form and the amount of such free water that the material can hold will depend upon the relative volume of capillary spaces that may be present. The fiber saturation point is of particular importance in the drying of thick, bulky materials having a more or less colloidal structure such as lumber and clay products, in which the removal of hygroscopic moisture is accompanied by shrinkage, stiffening, hardening, loss of plasticity, and other physical changes. Successful drying requires that these changes be controlled within safe limits in order to avoid injury to the material.

Removal of Free Water

In most commercial products which require a drying operation, the free water contained will flow readily from the interior to the surface by capillary action, as in a wick. Textiles, paper, fiber board, clay products, and the sapwood of most species of lumber represent material of this type, as well as finely divided materials which come in granular or powder form. In such substances the free water is removed usually without difficulty in the successive stages of saturated surface drying and unsaturated surface drying, followed by the removal of the hygroscopic moisture in the form of sub-surface drying.

In certain other materials of thick and more or less rigid form, notably the heartwood of some kind of lumber, the free water contained in the capillary spaces appears to be, in a sense, *bottled up* and cannot be made to flow from the interior to the surface in appreciable amounts. To remove the free water from such substances necessitates sub-surface vaporization within the material at the boundary of the free water zone, followed by diffusion of the vapor through the surrounding zones either as vapor or as adsorbed hygroscopic moisture. In either case, the zone surrounding the free water must be dried below its fiber saturation point in order to lower the vapor pressure to a point below saturation and thus permit this diffusion to take place. This means that shrinkage and other physical changes, such as stiffening or hardening, must occur in this outer zone before the interior portions have lost their free water and are ready for such changes. Consequently severe and harmful tensile and compressive stresses are apt to be set up in the drying of this type of material, due to unequal and non-synchronous shrinkage in the different zones, resulting in such injuries as surface cracking, honey-combing, warping, and *case hardening*, or the setting of the surface portion in a stiffened expanded condition.

In obtaining maximum speed of drying of materials in which there is no capillary flow of free water to the surface, dependence is placed in most cases upon the effect produced by heated air or other gas surrounding the material. In such cases the speed of drying can be increased in only two ways, first, by lowering the moisture content at the surface through the use of a lower humidity, or second, by increasing the temperature. The risk of harmful shrinkage limits the first, and the risk of exceeding the critical temperature of the material limits the second. In many cases, however, increased speed of drying can be secured by increasing the temperature of the air, and at the same time, increasing its humidity. In this way the amount of heat passing into the material is increased, while the increased humidity prevents the surface drying too fast and case hardening. An extreme example of this method is turning steam into a dryer in which green lumber is being dried.

The use of too high a temperature in removing the free water from fibrous or cellular materials of this kind, increases the plasticity of the structure and, at the same time, is apt to create a liquid tension within the cells themselves. This force, which may amount to many atmospheres under certain conditions, tends to draw the walls of the cells together, thus producing a *collapse* of these cells. This phenomenon is the cause of much injury in the drying of certain kinds of lumber and is probably present, to a greater or less extent, in the drying of all refractory material of a cellular nature through which the free water cannot flow by capillarity.

Another result of trying to force the drying of these refractory materials at too fast a rate, is the increasing resistance to the passage of heat from the surface to the interior and the passage of vapor outward, as the dryness of the surface increases. With many materials of this type, if the surface is allowed to dry below a certain point, it becomes increasingly difficult, not only for additional heat to penetrate to the interior, but also for vapor to work its way outward through the dry outer zones to the

¹U. S. Forest Service Bul. 70, p. 82, *Effect of Moisture on Strength*, Tiemann, 1907.

surface. Under these conditions, artificially increasing the humidity of the surrounding air is often a valuable help.

Internal Moisture Gradient

In drying by surface evaporation, an internal concentration gradient is set up between the moisture content at the center of the material and the moisture content close to the surface. The steepness of this gradient is dependent upon the rate of evaporation at the surface and the rate of transfusion of moisture from the center outward. In general, the steeper the gradient, the faster the drying. Since raising the temperature increases the rate of transfusion, the rate of drying is thereby increased without increasing the steepness of the gradient. For this reason the maximum temperatures which can be used without injury to the material will usually be found to produce the maximum rate of speed consistent with safety.

In keeping the moisture gradient within safe limits, during those stages of drying when surface shrinkage is taking place, the drying should be retarded by preventing the surface from drying below a certain critical moisture content which must be determined experimentally for different materials. This control of surface drying is accomplished by regulation of the relative humidity of the air in the dryer in conjunction with the regulation of the temperature. The most inexpensive means of securing the desired humidity is to recirculate a sufficient portion of the air. This not only utilizes the moisture which has previously been evaporated from the goods, but also conserves the heat contained in the air.

Equilibrium Moisture Content

With every material having hygroscopic properties, a point of equilibrium is reached between the vapor pressure of the moisture in the air and the vapor pressure of the moisture in the surface of the material. Every combination of temperature and humidity produces a corresponding equilibrium moisture content in the material. It is, therefore, possible to so regulate the conditions in the dryer at any stage of the operation that moisture in the material cannot drop below a certain equilibrium point, as may be desired for best results.

In the drying of most hygroscopic materials, the drying is continued until a somewhat lower moisture content is reached than equilibrium under conditions in use, to allow for enough regain to balance the moisture remaining in the interior and surface to a uniform condition. It is usually undesirable to carry the drying much beyond the equilibrium point of the material in use, as over drying is liable to produce undesirable physical changes in the material, and such over drying also unnecessarily prolongs the time and increases the cost of the operation.

CONTROL OF THE DRYING OPERATION

Circulation

It will have been seen that in any successful dryer heat has to be applied and the water evaporated has to be removed. A circulation of air or gas inside the dryer is essential for three reasons. First, all the heat, except such as may be supplied by conduction and radiation, must be

supplied by the air circulation. Secondly, the air must carry away the water vapor resulting from the drying. Third, duty of the air circulation is to have sufficient velocity to sweep away the film, or layer of nearly saturated air that is apt to remain in close contact with the evaporating surface, and whose high vapor pressure retards further evaporation. During the saturated and unsaturated surface drying stages, the drying rate will vary directly with the velocity of circulation over the material. When the stage of sub-surface drying is reached the velocity of circulation at the surface becomes of less importance, except that it must maintain uniform temperatures about the material, in order to produce uniform drying.

Circulation is expressed either by the rate of air flow over the material per minute, or by the number of air changes in the dryer per minute or per hour. The former is the more logical method. The air change expression is really a relic of the days when all fresh air was used, while an efficient dryer often uses only a portion of fresh air and the bulk of the air is recirculated.

As will have been seen, the heat required for evaporation is extracted from the air. If a small quantity of air is circulated through the dryer, it will be evident that a great drop in the temperature of the air occurs. This causes a great variation of temperature in contact with the material. In addition to this drop in temperature, a small quantity of air means the absorption of a large amount of vapor. The combination of these two factors will result in widely varying drying effects in different parts of the dryer. The larger the amount of air circulated, the less will be the drop in temperature, the less will be the amount of vapor absorbed per cubic foot of air, and hence, the closer will be the drying effects of the air in various parts of the dryer. If this large quantity of air, however, is all fresh air and the entire amount exhausted, the air thrown away will be far from having reached the proper saturation and a great waste of heat will occur. It may be sufficient to exhaust only a very small percentage of the amount of air that would be best from a circulating standpoint. Therefore, to meet both conditions, it is usually best to determine the volume required for circulation, then to only exhaust the portion that is sufficient to carry away the moisture and to recirculate the balance, introducing fresh air only in an amount equal to the amount exhausted. The higher the temperature of the dryer and the higher the humidities carried, the greater becomes the amount of air recirculated.

In some dryers the recirculation is obtained by the use of propeller type fans inside the dryer for recirculating over and over while the proper amount of fresh air is discharged into the dryer by a separate fan located outside. In other dryers, the fans for circulating the air are located outside the dryer and so arranged that a large portion of the air is recirculated and the rest introduced as fresh air into the same fan. In other dryers, a rapid circulation is passed through one section of the dryer; the entire amount is withdrawn, treated as to temperature and humidity and then introduced into the next section of the dryer. This results in a very slight drop in the temperature of the air in any part of each section, and by adding heat at each stage, the temperature and humidity can be controlled as desired throughout the dryer.

Next to circulating the proper amount of air, the most important con-

sideration is the method of introducing the air to the dryer. This is often called the distribution system. If there are dead areas, the air in that section will cool down rapidly, saturation will be reached and drying will be much slower in that section. If drying is to be uniform, then the circulation in every section of the dryer must be uniform. In addition, care must be taken not to have the air travel too far, causing a temperature drop too great and causing just as unequal drying conditions as if there were not enough circulation. The velocities must also be uniform over the goods at all parts or the drying will again become uneven.

Many of the older types of dryers were, and still are, of the type known as gravity or natural draft dryers. In these dryers, the natural laws of gravity are utilized to create a circulation. By regulating dampers and using chimneys or stacks, the moisture laden air is permitted to escape and an equal amount of fresh air admitted. The system is simple, and in some cases, may be adequate. The circulation cannot be increased beyond the amount of fresh air used, and hence, the circulation is inadequate to carry the proper heat uniformly through the dryer. This usually means that the drying is slow and often varies as the weather fluctuates. In a gravity dryer the circulation is usually slow, and such dryers are practically limited to materials which require baking or similar heat treatment rather than removal of much moisture. Even in a baking process, however, a rapid circulation results in much more even application of the heat so that most modern dryers have mechanical circulation by fans.

Many materials, such as gelatines, photographic films, etc., require drying at temperatures lower than the outside temperature. In such cases, the air must be cooled and dehumidified by refrigeration. It is then most economical not to use any fresh air in summer, but to recirculate all the air and remove the excess moisture by means of cold sprays of water or cooling coils. In high temperature dryers, on the other hand, the required temperature may be higher than steam temperatures and these may be secured either by entirely heating the air by oil or gas fired heaters, or such heaters may be utilized as a booster to the steam heated air. As such dryers need only a very small amount of fresh air, it may be best to recirculate a large volume and introduce only a very small quantity of fresh air heated to a high temperature by oil or gas heaters. These make possible temperatures up to 1000 F.

It is obvious that the air circulating through a dryer cannot be at the saturation point or drying would stop. The amount of saturation that can be carried in a dryer varies with the design of the dryer, rarely being over 50 per cent in a batch type dryer while in a tunnel dryer it may be 75 per cent or even higher.

Heat and Humidity Control

Regardless of whether the source of heat is steam or oil or gas fired heaters, there should be absolute control over the temperature and humidity. There is very little excuse today for hand operated valve control. If results do not have to be very accurate, a periodic setting of a thermostat and hand controlled dampers may be sufficient, but, in any case, there should be a recording instrument on every modern dryer. Only in this way can there be a check of the vigilance of the operator. The controls may be made more elaborate controlling automatically the

heat source, the humidity source, the amount of air recirculated and the amount of air exhausted. Although these are usually set to produce the desired temperature and humidity, in some cases controls have been developed which mechanically change their settings in accordance with a predetermined schedule. In some cases samples of the material are suspended on sensitized scales in the dryer opposite peep holes and the dryer is operated until the samples have reached the proper weight.

Operating the Dryer

No matter how well the dryer may be designed, a considerable part of the success of a drying operation depends upon the operator. He must watch the material and study the results in a constant effort toward improvement. Good dryer operation includes regular sampling and testing of material, watching the conditions within the dryer, and keeping systematic and complete records of all essential data, such as the movement of material, the length of the drying period, the temperatures and humidities at critical points in the dryer, and the extent to which the equipment is kept fully occupied. Standard schedules of drying time and drying conditions for material of different kinds are also useful in estimating production and regulating the operation of the dryer.

Moisture Content Calculations

Moisture content is usually expressed as a percentage of the original combined weight of the material and its contained water. Thus, if the moisture content were 20 per cent, the dry material would weigh 80 per cent of the total weight. In drying some types of material, notably lumber, it is customary to use the *bone dry* or water free weight of the material as the divisor, the weight of the water being calculated as a percentage of this dry weight. Regain, or absorption of hygroscopic moisture after drying, is almost always expressed by the latter method, as a percentage of the water free weight of the material.

The usual procedure, in determining the moisture content of a representative sample of the material, is to weigh this sample before it has had a chance to lose weight by evaporation, then dry it to a water free basis in an oven heated to a temperature slightly above the boiling point (about 215 F) until it has reached constant weight, then take a second weight record, and subtract the second weight from the first to determine the weight of the water. This figure can then be multiplied by 100 and divided by either the first weight or the second weight to obtain the percentage based on the original or final weight of the sample. Ovens used for this purpose are usually heated by steam or electricity and thermostatically controlled to prevent overheating.

Time of Drying

In Table 1, the average temperature range and drying time for a number of representative materials is given. The figures do not represent the maximum speeds attainable, nor necessarily the optimum temperature conditions, but are fairly typical of general commercial practice. In materials in which no capillary movement of free water can take place, the speed is limited by the rate of transfusion through the material, which is again determined by the critical temperatures and maximum gradients

TABLE 1. DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS

KIND AND THICKNESS OF MATERIAL	TEMPERATURE DEG F	DRYING TIME
Bedding.....	150-190	
Cereals.....	110-150	
Cocoanut.....	145-155	4-6 Hours
Coffee.....	160-180	24 Hours
Cores, Oil Sand, for molding..... $\frac{1}{2}$ in.—1 in. thick	300	30 Minutes
Black sand with Goulac Binder about $\left\{ \begin{array}{l} 3 \text{ in. thick} \\ 8 \text{ in. thick} \\ 16 \text{ in. thick} \end{array} \right.$	480 480 700	2½ Hours 4½ Hours 10 Hours
Feathers.....	150-180	
Films, Photographic.....	90	
Fruits and Vegetables.....	140	2-6 Hours
Furs.....	110	
Glue.....	70-90	2-4 Hours
Glue Size on Furniture.....	130	4 Hours
Gut.....	150	
Gypsum Wall Board— $\frac{3}{8}$ in. thick..... $\left\{ \begin{array}{l} \text{Start Wet} \\ \text{Finish} \end{array} \right.$	350 275	60 Minutes
Gypsum Blocks.....	350-190	8-16 Hours
Hair Goods.....	150-190	1 Hour
Hats, Felt.....	140-180	
Hops.....	120-180	
Hides, Thin Leather.....	90	2-4 Hours
Ink, Printing.....	70-300	
Knitted Fabrics.....	140-180	
Leather, Thick Sole.....	90	4-6 Hours
Lumber, Green Hardwood.....	100-180	3-180 Days
Lumber, Green Softwood.....	160-220	2-14 Days
Macaroni.....	90-110	
Matches.....	140-180	
Milk and other Liquid Foods (Spray Dried).....	250-300	Instantaneous
Molds, Green Sand, C. I. Flasks (one surface only exposed)		
8 in. thick.....	600	6 Hours
13 in. thick.....	700	13 Hours
Nuts.....	75-140	24 Hours
Paper, Machine Dried.....	180	
Paper, Air Dried.....	90-200	
Rubber.....	80-90	6-12 Hours
Sand, loose, 1 in. deep.....	300	10-15 Minutes
Shade Cloth.....	240	1-2 Hours
Soap.....	125	12 Hours
Starch.....	180-200	1-4 Hours
Stock Feed, Mixed.....	180-220	20-30 Minutes
Sugar.....	150-200	20-30 Minutes
Tannin and other chemical (Spray Dried).....	250-300	Instantaneous
Terra Cotta (air-drying in conditioning room).....	150-220	12-96 Hours
Wall Board, pasted ply board.....	300	15-20 Minutes
Wall Board, fiber-insulating, Roller Type Dryer.....	300-385	2½-3 Hours
Wall Board, fiber-insulating, Truck Type Dryer.....	300-385	24-48 Hours

which can be used without injuring the material. In this case, theoretical analysis, according to Lewis, shows that in the early stages the time of drying is proportional to the square of the thickness and to the square of the total moisture lost, and the time moisture curve is a parabola. In the later stages, after all free water has been evaporated, the curve becomes logarithmic. The form of this drying curve will, in general, resemble that shown in Fig. 2. On the other hand, in drying material in which free water can flow by capillary action at a sufficient rate to keep pace with surface evaporation, the time is proportional to the first power of the thickness and the rate will be governed by the drying conditions of the air and the area of surface exposed.

DRYER DESIGN

Methods of Calculations

A psychrometric chart applying to atmospheric pressure and plotted in units convenient for use in dryer calculations, is given in Fig. 3. The amount of moisture in the air, H is given in pounds of water per pound of dry air. With this choice of units, changes in H represent directly the moisture picked up by the air from the material being dried. The inclined

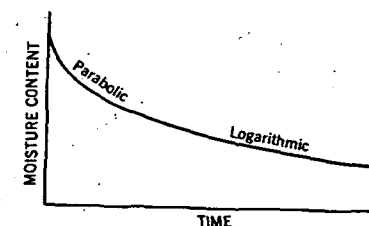


FIG. 2. TYPICAL TIME-MOISTURE CURVE FOR MATERIALS IN WHICH NO CAPILLARY FLOW OF FREE MOISTURE OCCURS

straight lines are adiabatic cooling lines for air in contact with water at its wet-bulb temperature, and also represent the cooling of air in the adiabatic drying of any stock, the sensible heat of which is negligible in comparison with the heat of vaporization of the water in it.

The following nomenclature and explanation of terms will be used in the discussion of drying calculations:

H = humidity 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.

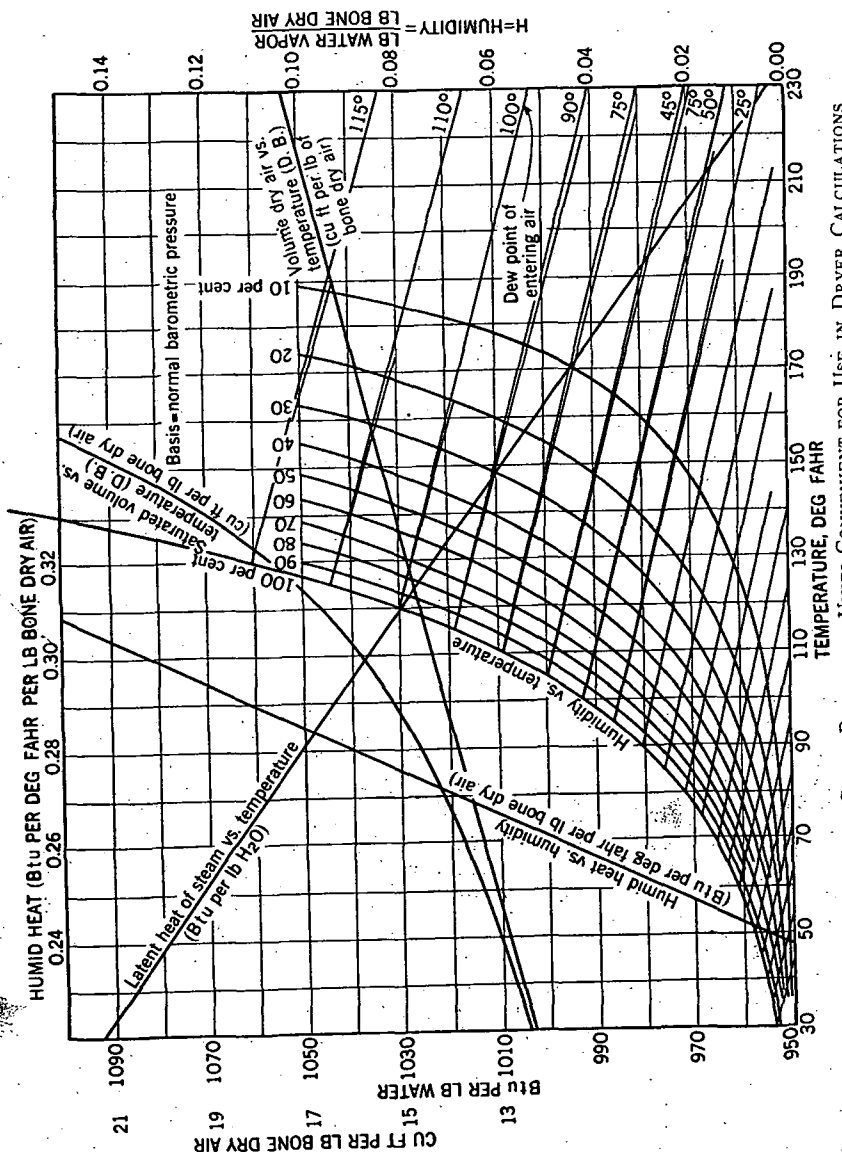


FIG. 3. PSYCHROMETRIC CHART PLOTTED IN UNITS CONVENIENT FOR USE IN DRYER CALCULATIONS

CHAPTER 41—DRYING

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 deg.

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:

$$G(H_2 - H_1) = S(w_1 - w_2) \quad (1)$$

In discontinuous dryers, e.g., compartment dryers, the drying operation is given by the equation:

$$G(H_2 - H_1) = S' \frac{dw}{d\theta} \quad (1a)$$

In the continuous dryer, the heat consumption per unit time is:

$$\frac{Q}{\theta} = Gs_1(t_2 - t_1) + G(r_2 + t_2 - t'_2)(H_2 - H_1) + S(t'_2 - t'_1)(s' + w_1) + B \quad (2)$$

Equation (2) 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:

$$S' (t''_1 - t''_1) (s' - w_1)$$

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, rarely introduces serious error. The evaluation of equation (1a) may be approximated in a similar manner.

The first term of the right hand member of equation (2) 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 is imperative.

The 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 is found experimentally to maintain 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 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. 3, $H_2 = 0.105$. 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.

Inspection of equation (1) shows that (H) is linear in w . Hence, one can construct on Fig. 4, 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. 3 shows that its wet-bulb temperature is 129 F. This is plotted at the right hand side of Fig. 4. 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 (1) or by reading directly from the diagram, the value being 0.0392. Fig. 3 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. 4.

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 (1)) and from Fig. 3, $t_w = 95$ F. Hence the wet-bulb depression, $t - t_w = 150 - 95 = 55$ F.

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. 4 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 (1). Thus, per 100 lb of dry stock, it is necessary to supply 633 lb of dry air. Furthermore, since from Fig. 3 it is seen that the volume of 50 per cent saturated air at 70 F, is 13.55 cu ft per lb, 8580 cu ft of room air must be supplied per 100 lb dry stock.

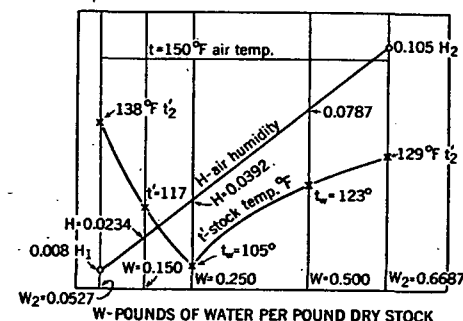


FIG. 4. CHART SHOWING TEMPERATURE-HUMIDITY RELATIONSHIP IN A DRYER

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 11,400 cu ft per 100 lb of dry stock. Finally, the heat necessary to supply to the dryer; as a whole, or to any section of it, may be computed from equation (2).

Estimating Method

Estimating methods are often used, especially where approximate figures are desired and time does not permit of more elaborate figures. Such methods are really the result of practical experience and are of value only in proportion to the experience of the user. There are, however, certain of these short cuts which are of value.

The temperature will drop approximately $8\frac{1}{2}$ F per grain of water evaporated per cubic foot of air (measured at 70 F) or approximately 0.62 F per lb 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 $1\frac{1}{2}$ lb to a more usual figure of from $2\frac{1}{2}$ to 3 lb of steam per pound of water evaporated.

DRYER ARRANGEMENT AND CONSTRUCTION

Outside handling and storage space usually is an important consideration in dryer design. Continuous dryers have an advantage in this respect since they provide storage for a considerable proportion of the material in process, while for charge dryers enough outside space must be provided to handle at least one full charge awaiting the dryer and one full charge upon removal. The more rapid the drying operation, the more efficient must be the handling facilities, in order to hold operating costs to a minimum and to keep the dryer operating at maximum capacity.

An important consideration in construction is to provide tight, well insulated doors which can easily be opened and closed, with minimum labor and loss of time. Fire resistance of doors is also important, in case of outside exposure to fire risks. Insurance rates must be considered in their relation to fire protective arrangements such as automatic sprinklers, fire doors and walls, steam jets for use in smothering fires that may start inside the dryer, etc. Ceilings of dryers should be as flat as possible and the space between the material and the walls and ceiling should be held to a minimum. A factor often overlooked is that of suitable provision for ventilation of the space around the dryer openings so that warm moist air which may escape from the dryer when the doors are opened, may be carried away before it has a chance to condense on the ceiling or windows of the buildings in which the dryer is located, or into which it opens. Convenient arrangements for the operator should also be provided where tests can be made, records kept, and clothes changed, in case it is necessary for the operator to work inside the dryer.

EXPERIMENTAL TECHNIQUE

One vitally important phase of dryer design has been ignored in the preceding discussion, namely, *the time required to dry the stock*, or, what is equivalent to the same thing, *the size of the dryer*, and the character and extent of the *contact surface* between the stock and the air. In general, this cannot be determined except on the basis of experimental data on the specific material to be dried. The source of such data may either be the known performance of actual commercial drying installations handling the same stock or direct experimental determinations in the laboratory. Data of this type for many materials is given in Table 1.

Where it is necessary to determine drying conditions and rate in the laboratory, it is vitally important to properly control the experimental conditions. Where possible, the material upon which the experiments are made should have the same shape and size as that to be treated commercially. Furthermore, the conditions of exposure to the drying air, the temperature and humidity of that air, and its velocity and distribution over the material should be identical with those used in the full scale operation. Where the commercial operation is by batch, it is relatively easy to duplicate commercial conditions in the laboratory. However, where continuous operation is intended, it is usually difficult to build a continuous experimental dryer. In such a case, a preliminary diagram of the type of Fig. 4 should be constructed and the drying conditions of the batch experimental operation controlled in the laboratory to conform to

the humidity moisture content relationship of the ultimate continuous operation. In this way, dependable data on the drying rate can be obtained in the laboratory.

As understanding of the mechanisms of drying and of the drying characteristics of the material to be dried, is of the utmost importance in designing successful and economical dryers, especially in the interpretation and extrapolation of plant and laboratory test data. Air velocity has an important influence on the rate of drying in the constant rate period and in the first zone of the falling rate period, but in the second zone of the falling rate period, the rate of diffusion of water to the surface controls the drying, and hence increasing the air velocity past the surface can have little effect on the rate of drying.

REFERENCES

- 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).
- Drying in Industrial Plants, by J. O. Ross.
- 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).
- Commercial Dehydration, by J. E. Whitley (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920, p. 551).
- 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 Dryers, by J. E. Bolling (A.S.H.V.E. JOURNAL, October, 1921, p. 715).
- Modern Drying Machinery, by H. B. Grenshaw, London, 1926.
- The Kiln Drying of Lumber, by A. Koehler and R. Thelen, New York, 1926.
- Drying, by W. H. Carrier (*Marks' Mechanical Engineers Handbook*, 2nd edition, 1924).
- Drying, *Kent's Mechanical Engineers Handbook*, 10th edition, 1923.
- Calculations for Drying Design, by Grosvenor (*Transactions, A. I. Chem. Eng.*, 1908, p. 184).
- The Rate of Drying Solid Materials, by J. Lewis (*Ind. Eng. Chem.*, 1921, p. 427).
- Principles of Chemical Engineering, by Walker, Lewis, McAdams, 1923, (Chapters 12 to 16 on Evaporation, Humidity and Drying).
- The Kiln Drying of Lumber, by H. D. Tiemann (*Lippincott*, 1920).
- Drying by Means of Air and Steam, by E. Hausbrand (*D. Van Nostrand & Co.*, 1901).
- Principles of Drying Lumber and Humidity Diagram, by H. D. Tiemann (*Forest Service Bul.* 104, 1912).
- Symposium on Drying. Articles by W. K. Lewis, W. H. Carrier, A. E. Stacey and Fleming, R. G. Metz, G. B. Ridley, C. O. Lavett, D. J. Van Marle (*Jour. Ind. Eng. Chem.*).
- The Drying of Solids, by T. K. Sherwood (*Bul. Mass. Inst. Tech.*, Nos. 237, 247 and 258).
- Drying, by Perry (*Chemical Engr. Handbook*).
- Adiabatic Drying of Hygroscopic Solids, by A. M. McCready, and W. L. McCale (*Trans. Am. Inst. Chem. Engr.*, 1933).

PROBLEMS IN PRACTICE

1 • What is meant by an adiabatic dryer?

A dryer in which all of the heat required for evaporation is furnished only by the heat in the circulating air.

2 • What other broad classes of dryers are used commercially beside adiabatic dryers?

Dryers where the heat is furnished by contact with a heated surface, by radiant heat, by electrical induction or by a combination of any of these sources with heated air.

3 • What makes a commercial adiabatic dryer differ from a theoretical one?

The word *adiabatic* means no heat lost to the outside and that the sensible heat lost by the air is equal to the latent heat of the water evaporated. In an actual dryer, the solid containing the water, and the water itself must be heated to the temperature of evapora-

tion, before evaporation can begin. Radiation losses from the dryer enclosure is the other factor causing deviation from the theoretical adiabatic process.

4 • What is a Zone Dryer?

This term refers to a continuous dryer where the drying medium is divided into two or more sections, in order to have better control of the temperature and humidity gradients through the dryer.

5 • What is meant by high and low temperature drying?

High temperature drying refers to an evaporator process where the temperature used is above the boiling point of water; i.e., 212 F. Low temperature drying utilizes temperatures under 212 F, the range in which humidity is an important consideration.

6 • How may air for high temperature drying be obtained?

By direct oil or gas fired heaters, by indirect oil or gas fired heaters, by hot circulated oil, by high pressure steam, or by electric heaters.

7 • What is meant by the critical point or critical moisture content?

The point in the drying of a hygroscopic substance where all of the free moisture has been evaporated and after which the water is all in the combined or bound state. It is the dividing point between the constant rate drying period and the falling rate. Shrinkage of the material, if subject to shrinkage, will begin to occur at this point.

8 • What is equilibrium moisture content?

The moisture content of a hygroscopic material in the presence of air at a given temperature and humidity which will remain constant no matter how long the material is exposed. The vapor pressure of the bound water in the material is equal to the vapor pressure of the surrounding air at this point.

9 • If a material enters a dryer containing 70 per cent water and 30 per cent solids, and leaves the dryer with 10 per cent water and 90 per cent solids, (a) what is the evaporation per pound of dried product? (b) What is the evaporation per pound of bone dry material?

a. $\frac{90}{30} - 1 = 2$ lb water per pound dried product.

b. Water entering = $\frac{70}{30} = 2.33$ per cent on bone dry basis.

Water leaving = $\frac{10}{90} = 11$ per cent on bone dry basis.

Water evaporated 222 per cent on bone dry basis.

Evaporation = 2.22 lb water per pound bone dry material.

10 • What items must be included in a calculation of the dryer heat requirements?

- Heating water to be evaporated from the entering temperature to the temperature of evaporation.
- Evaporating water to be removed.
- Superheating evaporated water from the temperature of evaporation to the exit temperature of the air.
- Heating material from entering to leaving temperatures.
- Heating residual water from the entering to the leaving temperatures.
- Heating conveyor or other supporting materials.
- Radiation losses through the enclosure.
- Sensible heat in the exit air.

Chapter 42

MOTORS AND THEIR CONTROL

Direct Current Motors, Alternating Current Motors for Single Phase and Polyphase, Special Applications, Classification of Motors, Manual Control, Automatic Control, Pilot Controls, Direct Current Motor Control, Squirrel Cage Motor Control, Multispeed Motor Control, Slip Ring Motor Control, Single Phase Motor Control

THE commonest form of prime mover is the electric motor, which is available in a large number of types for the various services to which it may be applied. The equipment for starting, controlling and protecting these motors varies with the type and with the functions it is desired to attain. Motors used for heating, ventilating and air conditioning applications may be divided into two general classifications as follows:

- For use with direct current.
- For use with alternating current.

DIRECT CURRENT MOTORS

There are three types of direct current motors available:

- Shunt Wound.
- Compound Wound.
- Series Wound.

Shunt Wound motors are suitable for application to fans, centrifugal pumps, or similar equipment, where the amount of starting torque required is relatively small, for the majority of applications are used in the field of heating, ventilating and air conditioning. They may be used on reciprocating pumps and compressors, if started under unloaded conditions.

Compound Wound motors are required for application to compressors, stokers, reciprocating pumps when started under loaded conditions, and also when applied to similar equipment where high starting torque is required. Whenever frequent starting makes high starting and accelerating torque desirable, or where sudden changes of load are encountered, compound wound motors are used.

Series Wound motors find only limited application in a few special cases and are available in only a limited range of sizes.

Direct current motors are available with speed characteristics of four types:

1. Constant speed.
2. Adjustable speed.
3. Adjustable varying speed.
4. Varying speed.

Constant Speed motors may be shunt wound or compound wound. Shunt wound motors have a nearly flat speed-load characteristic, with a regulation of 12 per cent for 5 hp and smaller and 10 per cent for $7\frac{1}{2}$ hp and larger, based on full load speed.

Compound wound motors have a speed regulation over the range from full load to no load of not more than 25 per cent, based on full load speed.

Adjustable Speed motors are usually shunt wound since it is impractical to maintain the proper relation between the shunt and series fields of compound wound motors when wide variations of the field strength are required to obtain the speed adjustment.

Adjustment of the speed of shunt wound motors is obtained by field control, with the minimum or base speed at full field strength and higher speeds at reduced field strength (obtained by adding resistance in the field circuit). The speed regulation from no load to full load will not exceed 22 per cent for 2 to 5 hp; nor 15 per cent for $7\frac{1}{2}$ hp and larger. Below 2 hp, the regulation may exceed 22 per cent. If closer speed regulation is required, specifically wound motors must be obtained.

Practically constant horsepower output is obtained at all speeds up to a ratio of 2 to 1. For higher speed ratios, the horsepower rating at the minimum speed is less than at the maximum speed, this difference varying with the speed ratio. High efficiency is maintained over the entire speed range. Most listed constant speed motors are suitable for operation up to a speed ratio of 2 to 1 by the use of proper control equipment.

Adjustable Varying Speed motors may be either shunt or compound wound and speed adjustment is obtained by adding resistance in series with the armature. The speed thus obtained is always below the rated speed. Any standard shunt or compound wound constant speed motor may be used in conjunction with the proper armature resistor. The usual range of speed reduction is 50 per cent. The speed obtained for any setting of the resistor depends on the load of the motor and will vary with this load.

The speed regulation at high speed is comparable to a constant speed motor, but becomes poorer as the speed is decreased.

When operating at reduced speed, an increased torque requirement which the motor could easily handle at rated speed is easily sufficient to stall the motor; for example, a motor operating at two-thirds speed would be stalled by a torque about 50 per cent in excess of the normal requirement.

The efficiency of the motor is reduced as the speed is reduced, since the loss in the resistor is greater at lower speeds. Speed reduction by armature control is usually selected where:

1. A wide speed range is not required.
2. Close speed regulation is not necessary.

3. Operating time at reduced speed is short.
4. Operating load at reduced speed is small so that the reduced efficiency can be ignored.

Varying Speed motors are series wound and the speed varies with the load on the motor. They should be used where:

1. The load is practically constant or increases with speed.
2. The motor can easily be controlled by hand.

They should not be used where there is a possibility of operation without load or at a reduced load, as the speed of the motor may become dangerously high.

For shunt wound motors with full field strength, the starting torque varies almost directly with the starting current, which is dependent on the resistance in the armature circuit. With varying positions of the starting rheostat, it is possible to obtain a wide range of starting torque, within the limits of starting current permitted by the power company.

A compound wound motor requires somewhat less current for the same starting torque. The maximum torque is limited by commutation.

ALTERNATING CURRENT MOTORS

Alternating current motors may be divided into two main groups, namely, (1) those operating on single phase current, and (2) those operating on polyphase current.

1. Single phase motors are available in three common types:
 - a. Repulsion start, induction run.
 - b. Capacitor type.
 - c. Split phase.
2. Polyphase (2 or 3 phase) motors are available in four common types:
 - a. Squirrel cage induction motor.
 - b. Automatic start induction motor.
 - c. Slip ring, wound rotor induction motor.
 - d. Synchronous motor.

Where the public utility supplying the current determines that a particular installation should be served with polyphase current, it is generally understood that the major portion of the motors will be for polyphase current, although it is commonly acceptable for the smaller motors to be single phase. This will in general, limit the use of single phase current to the smaller motor ratings and the polyphase to the larger motors.

Single Phase Motors

Repulsion start, induction run motors are suitable for use with fans, centrifugal pumps, reciprocating pumps and compressors (started loaded or unloaded) since these motors have relatively high starting torques and low starting currents. This type of motor has a constant speed characteristic. The motor has a commutator and brushes to obtain starting and an integral automatic governor for changing to an induction motor when nearly up to speed.

The *capacitor type* motor is available only in the small and fractional horsepower ratings. It is manufactured in three types:

1. Capacitor Start, limited to $\frac{1}{4}$ hp and smaller, using the condenser in starting and disconnecting the condenser by means of an automatic governor when operating at full speed. The starting torque is low; used for direct connected service only.

2. Capacitor Start, Capacitor Run having a condenser in the circuit while starting and running at speed. An auto transformer supplies high voltage while starting and normal voltage when the motor is operating at full speed. The change is accomplished by an automatic governor in the motor.

These motors have starting and operating characteristics suitable for the operation of the various types of air conditioning equipment within the range of sizes available. This type of motor has a constant speed characteristic.

The choice between the capacitor motor and repulsion, induction motor is largely a matter of personal opinion.

Adjustable speed motors in this type are available where low starting torque is satisfactory.

These motors do not have a governor but act as capacitor start and capacitor run motors with a fixed condenser in circuit at all times in addition to an auto transformer arranged with several taps, a few of which are selected to give the desired variation in speed.

3. Capacitor type motors with medium starting torque are also available in the construction described previously.

Split phase motors are available in sizes $\frac{1}{4}$ hp and smaller and should be used where high starting torque is not required (such as small fans, pumps and oil burners). The starting current is higher than for either the repulsion, induction, or capacitor motors. A special type, with enclosed end brackets and a thrust bearing, is available up to $\frac{1}{2}$ hp for use with propeller fans mounted on the motor shaft. This type of motor has a constant speed characteristic.

Polyphase Motors

Squirrel cage induction motors are available in three types and a full range of sizes:

1. The normal torque, normal starting current squirrel cage motor has close speed regulation, high efficiency, high power factor, high pull-out torque, high starting torque and is suitable for general purpose applications. This motor has a large current inrush and a low starting current power factor. It operates with these characteristics only when started directly across the line on full voltage. When central stations require current limiting starting equipment on such motors, the starting torque is less. Current limiting hand operated compensators are standard equipment and have 50, 65 and 80 per cent taps, the 65 per cent tap being regularly connected when the compensator leaves the factory.

2. The normal torque, low starting current squirrel cage motor has approximately the same torque as the normal current motor, but the starting current is approximately 20 per cent less than the normal torque motor on full voltage and well within the recommended *EET* limits on 30 hp sizes and smaller.

This motor lends itself to automatic or remote control because no current limiting starting equipment is necessary for 30 hp and smaller sizes. A magnetic starter with low voltage and thermal relay overload protection gives the most satisfactory service.

These switches may be controlled by push button, thermostat or pressure relay to meet the requirements of any particular installation.

3. The high torque, low current squirrel cage motor has a starting torque approximately 25 to 50 per cent greater than the normal torque motor on full voltage with starting current approximately 10 per cent less than the normal torque motor started on

full voltage, but within the required limits on 30 hp sizes and smaller. These motors are also started directly across the line on full voltage through a magnetic starter or other approved starting device.

These three types of motors are also available in two, three, or four speed designs with variable torque, variable horse power or constant torque characteristics. Two speed motors may be either single, or two winding; three speed motors are single, two, or three winding; and four speed motors are two, three, or four winding. When a motor is wound with a winding for each speed, better operating characteristics are obtained because no sacrifice is made for the other speed and operating characteristics approaching single winding motors may be expected.

Frequently, multispeed motors lend flexibility to an installation that cannot be obtained in any other way.

Multispeed motors are started, through magnetic starting equipment with overload and low voltage protection, directly across-the-line with compelling relays to insure starting on low speed regardless of the ultimate running speed. Starting on low speed limits the starting current to the starting current of the low speed winding and consequently lowers the maximum demand.

Often where the central station requires current limiting starting equipment for the normal torque, normal starting current motor, it is advisable to use the normal torque low starting current multispeed motor.

It is apparent from these motor characteristics that a squirrel cage motor may be selected for operating any air conditioning and allied equipment.

Automatic start induction motors are constructed with two windings on the rotor, one of which is a high resistance, squirrel cage winding used in starting and gives a high starting torque. A centrifugal mechanism within the motor switches to the second low resistance winding when the motor comes up to speed, thus obtaining running characteristics equal to the normal torque, normal current squirrel cage motor. The power factor of the starting current is high.

Split ring, wound rotor motors are built for two classes of service, constant speed and adjustable variable speed. The motors are identical in each case and use the same primary control, the only difference being in the secondary control.

Slip ring motors for constant speed service are used where high starting torque with low starting current is required for bringing heavy loads up to speed. The resistance is in the secondary or rotor circuit, only when starting, and is short circuited when the motor is up to speed.

For adjustable varying speed service, part or all of the secondary controller resistance is in the circuit whenever the motor is operating below full speed. The speed obtained with a given resistance in the secondary circuit is dependent on, and changes with, the load on the motor. The horsepower developed by the motor is approximately proportional to the speed, whereas the power required by the motor is practically the same at reduced speed as at full speed, hence the efficiency at reduced speeds is much lower than at full speed.

Synchronous motors are ordinarily used only where there is a necessity for, or advantage in, obtaining power factor correction. It is advisable to consider each application as a special case.

TABLE 1. CLASSIFICATION OF MOTORS

CURRENT	TYPE	SPEED CHARAC- TERISTICS	STARTING TORQUE	STARTING CURRENT	HP RANGE	TYPE OF APPLICATION SEE FOOTNOTES
	<i>Constant Speed Drives</i>					
DIRECT	1. Shunt	Constant	Medium	Medium	All	(a) Fans and (c) Centrifugal Pumps
	2. Compound	Constant or Variable	High	Medium	All	(b) (c) (e) Recip- rocating Pumps and frequent or hand starting
	3. Series	Variable	High	Medium	Small	(d) Fans direct connected
POLY- PHASE	4. Squirrel Cage General Purpose	Constant	Medium	High 6-8 Times	All	(a) Fans and (c) Centrifugal Pumps
	5. Squirrel Cage Medium Torque	Constant	Medium	Medium 5-6 Times	Medium Small	(a) Fans and Centrifugal Pumps
	6. Squirrel Cage High Torque	Constant	High	Medium 5-6 Times	Medium Small	(b) Reciprocating Pumps (e) and Compressors started loaded
	7. Slip Ring Wound Rotor	Constant	High	Low 1-3 Times	All	(a) and Hoists (b) Reciprocating Pumps (c) and Frequent (e) or Hand Starting
	8. Synchronous High Speed	Constant	Medium	Medium 5-7 Times	Medium Large	(a) Fans and Centrifugal Pumps
	9. Synchronous Low Speed	Constant	Low	Low 3-4 Times	Medium Large	(a) Reciprocating Compressors Starting Unloaded
SINGLE PHASE	10. Repulsion Induction	Constant	High	Medium	Medium- Small	(a) Fans (b) Pumps and Compressors
	11. Capacitor	Constant	Low	Medium All	Small and Fractional	(d) Fans—direct
	12. Capacitor	Constant	Medium	6-8 Times		(a) Fans—belt
	13. Capacitor	Constant	High			(b) Compressors and Pumps

TABLE 1. CLASSIFICATION OF MOTORS—(Continued)

CURRENT	TYPE	SPEED CHARAC- TERISTICS	STARTING TORQUE	STARTING CURRENT	HP RANGE	TYPE OF APPLICATION SEE FOOTNOTES
	<i>Adjustable Speed Drives</i>					
DIRECT	14. Shunt Field Adjustment	Constant	Medium	Medium	All	(a) Fans and (e) Centrifugal Pumps
	15. Shunt Armature Resistor	Variable	Medium	Medium	All	(a) Fans and (e) Centrifugal Pumps
POLY- PHASE	16. Squirrel Cage Regrouped Poles	Constant Multi- speed	Medium or High	High	All	(a) Fans and (b) Pumps
	17. Slip Ring Secondary Resistance	Variable	High	Low	All	(a) Fans and (b) Centrifugal Pumps
SINGLE PHASE	18. Capacitor Transformer Adjustment	Variable	Low	Low	Fractional	(d) Fans

^aApplications:
^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 hand starting (large WR^2) requires a higher starting and accelerating torque.
^d. Fans direct connected.
^e. Stoker drives.

SPECIAL APPLICATIONS

A few applications of motors may require special constructions such as splash proof, explosion proof, fully enclosed, and self-ventilated to meet hazardous or special duty conditions. These requirements are frequently encountered in certain industrial applications, in which cases it is necessary to select the motors from the viewpoint of service conditions, as well as the required operating characteristics to meet the demands of the machines being driven.

The general classification of motors used for heating, ventilation and air conditioning is shown in Table 1.

CONTROL EQUIPMENT FOR MOTORS

In selecting control for alternating and direct current motors it is necessary to determine whether the installation is to be operated by manual or automatic control. The available controls and the function of each group of apparatus may be outlined as follows:

1. Manual Control:

a. To establish current.

- (1) Snap switch.
- (2) Knife switch.
- (3) Manually operated contactor.
- (4) Drum switch.

- b. Establish current and add overload protective device.
 - (1) Snap switch with overload element.
 - (2) Knife switch with fuse or thermal cutout.
 - (3) Manual contactor with overload protective device; also reduced voltage starting compensator.
 - (4) Drum switch with overload protection.
 - c. Establish current and add overload and low voltage protective devices.
 - (1) Not used.
 - (2) Not used.
 - (3) Manual contactor or reduced voltage compensator with overload and low voltage release.
 - (4) Drum switch equipped with latch coil to give low voltage release.
2. Automatic Control:
- a. To start on full voltage.
 - (1) Without overload device.
 - (2) With overload device.
 - (3) With combination overload device and knife switch.
 - b. Reduced voltage starting.
 - (1) Primary resistance type starter.
 - (2) Auto compensator type.
 - (3) Reactance type.

PILOT CONTROLS

In selecting pilot control devices to operate in conjunction with either manual or automatic motor control, it is necessary that they be classified as follows:

1. *Two Wire Control.* Most thermostats, float switches, and pressure regulators, provide two wire control which gives low voltage release. A three position pilot switch can be used in connection with this method and thus provide manual control. With a low voltage (12 or 20 volt) control circuit it is desirable to use a low voltage thermostat. When this type of thermostat is used it will be found that a saving in the wiring cost results. When using the low voltage thermostat on a control circuit a relay and transformer panel should be used instead of the low voltage coil on the starter.

2. *Three Wire Control.* Momentary contact start and stop push button stations are usually furnished as standard accessories with automatic starters, which gives low voltage protection. This control cannot be used in combination with two wire pilot devices.

In selecting manual control for an alternating or a direct current motor, the common practice is to locate the control near the motor. When the control is installed at the motor, an operator must be present to start and stop or change the speed of the motor by operating the control mechanism. Frequently manual control is 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 multispeed motors, is only used as a speed setting device with the starting and stopping functions operated automatically through thermostats, and pressure switches.

Because of the increasing complexity of air conditioning systems, heating, ventilating and air conditioning equipment is being operated on

automatic control with less dependence 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. In the majority of air conditioning installations, requiring motors 1 hp and larger, two or three phase alternating current is usually supplied.

DIRECT CURRENT MOTOR CONTROLS

Air conditioning installations using direct current power are now only used where alternating current is not available. Direct current motors are always started through starters, which are devices using a resistance to be put in series with the armature circuit during starting only, the resistance being gradually cut out as the motor comes up to speed. The starting current is held within safe limits by the use of the resistance.

The speed of a direct current motor may be regulated by the following methods:

1. Speed regulation by field control—by using a device with resistance to be put in series with the field winding. After the motor has been started to be used to increase the speed of the motor above full field speed.

2. Speed regulation by armature control—by using devices with resistance to be put in series with the armature circuit to be used to reduce the speed of the motor below full field or normal speed.

3. Combinations of field and armature control, so that the starting, field control, or armature control may be combined in a single unit.

Field control is usually preferred, depending on the size of the installation. For example, if a direct current motor were required with speed regulation between 1200 and 600 rpm, a choice of supplying a 1200 rpm motor with armature control or a 600 rpm motor with field control, both giving the same speed variation would be possible. While the 1200 rpm motor with armature control is lower in first cost than the 600 rpm motor with field control, the cost of operating the 600 rpm motor with field control is less and will save the difference in first cost over a period of time depending on the size of installation. A wide speed variation can be easily obtained in a direct current motor by using a combination of field and armature control.

SQUIRREL CAGE MOTOR CONTROL

To meet the requirements of various drives of an air conditioning system, three types of squirrel cage, two or three phase motors may be used:

- 1. Normal torque, normal starting current.
- 2. Normal torque, low starting current.
- 3. High torque, low starting current.

Because of the large current inrush of the normal torque, normal starting current motor, central stations usually require current limiting

starting equipment on such motors above 5 hp. To meet the starting current requirements, manual or automatic current limiting compensators are used. These compensators are equipped with 50, 65 and 80 per cent voltage taps, the 65 per cent tap being regularly furnished when the compensator leaves the factory. Motors 5 hp and smaller have starting currents within the requirements of central stations and manual or magnetic, full voltage control may be used.

The normal torque, low starting current motor has a starting current which is approximately 20 per cent less than the normal current motor on full voltage and well within the required current limits on 30 hp sizes and smaller. This motor, therefore, lends itself to across-the-line control because no current limiting equipment is necessary. In selecting motors for fans, pumps, or blowers, it should be noted that while the cost of the normal starting torque, low starting current motor is higher, the cost of full voltage control is lower, so that the total cost of low starting current motors with across-the-line control is lower.

A magnetic starter with low voltage and thermal overload protection gives the most satisfactory service. These switches may be controlled by remote push button stations, thermostats, or pressure switches to meet the requirements of any particular installation.

The high torque, low starting current motor has a starting current approximately 10 per cent less than the normal torque, low starting current motor when started on full voltage. These motors, most commonly used on compressor drive, can be started directly across-the-line with manual or magnetic starters.

MULTISPEED MOTOR CONTROL

To make an installation more flexible, multispeed motors are available with two, three or four speed designs, with variable torque, constant torque or constant horsepower characteristics. Multispeed may be started by means of manual or magnetic starting equipment.

When using automatic magnetic control with two, three, and four speed separate winding or consequent pole motors, control is 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, which is known as selective speed control. It is commonly used in the smaller theatre installations where the fan and motor is located back stage and the speed control is located in the lobby.

Magnetic multispeed motor controllers may also be provided with a compelling relay which makes it necessary that the operator press the first speed button before regulating the motor to the desired speed. This assures the operator that the motor is always started at low speed before the motor is adjusted to one of the higher speeds. Starting on low speed limits the starting current to the starting current of the low speed winding, and therefore, permits the use of motors in sizes larger than ordinarily permitted by central stations for full voltage starting.

Timing relays, which provide for automatic acceleration, may be used for control. With the automatic acceleration feature, it is only necessary

to press the button for the desired speed. The motor will always start in low speed and automatically step up to the desired speed.

Where the change of speeds does not occur at regular intervals, and where it is only necessary to change from one speed to another to take care of seasonal requirements, a manual drum speed selector may be used. This drum is used to select the proper motor speed while an automatic starter is used to start and stop the motor.

The smaller size speed selector drums rated 10 hp at 200 volts and smaller may also be used as a motor starter to make and break the current, as well as, serving as a speed selector device. Reversible or non-reversible drums may be supplied depending on the requirements of the installation.

In the large size drums, a separate contactor must be provided to make and break the current. The contactor may be any approved starter. Overload and low voltage protection may be accomplished by using a magnetic starter. No push button station is required, the handle switch on the drum having the same characteristics as a three wire push button station.

In selecting two speed motors for fan, pump, blower, or compressor drive it will be found that the two winding motors are more expensive than the single winding. The control for two speed, two winding motors is more economical and the combined price of the motor and contactor is only slightly higher. Because of the better performance of the two speed motor and the factor of safety in having two independent motor windings, the increased cost is considered worth the difference.

SLIP RING MOTOR CONTROL

When close speed regulation and low starting current is required slip ring or wound rotor motors are used. Slip ring motors are built for two classes of service, constant speed and adjustable varying speed. The motors for the two classes of service are identical, the only difference being in the secondary control used with the motors. Control for both primary and secondary of a slip ring motor is required.

The primary control for a constant or adjustable speed is the same type as used with squirrel cage motors. Manual or magnetic starters, across-the-line type, may be used depending on the installation.

The starting current and starting torque of a slip ring motor are almost entirely dependent on the amount of resistance in the secondary control and in the manner in which the secondary control is operated. The *National Electric Manufacturers Association* has adopted service classifications which allow a selection of resistors permitting a starting current on the first contact of resistance varying from approximately 25 per cent of full load current to approximately 200 per cent of full load current or more, and permitting the resistor to remain in the secondary circuit of the motor for a period varying from not more than 15 seconds during an interval of operation from 4 minutes to continuous.

Speed regulation of a slip ring motor is obtained by inserting resistance in the secondary circuit and usually provides for a 50 per cent speed reduction when the motor takes its full rated current at normal speed.

As resistors are supplied for both fan duty and constant torque duty, care should be taken in selecting the proper resistors.

Slip ring motors when used with centrifugal pumps and fans should have fan duty resistors. Because of the low current inrush of the fan and pump load a starting resistor *NEMA* classification No. 15 may be used. For speed regulation resistor, classification No. 93 should be selected. On a compressor drive using an unloader, a constant torque resistor classification No. 15 should be used. If the compressor is started under load, *NEMA* classification No. 56 or 76 are used. For constant torque speed regulation, resistor No. 95 is used.

SINGLE PHASE MOTOR CONTROL

Where three phase current is not available or where single phase operation is preferred, then single phase repulsion induction, capacitor type or multispeed single phase motors may be used. Since the starting currents of all single phase motors are designed to operate within the recommended starting current rules of the *EEI*, across-the-line type starters are available as follows:

1. Enclosed two pole manually operated motor starters with thermal overload protection.
2. Enclosed two pole automatic motor starter operated by a push button, thermostat or similar device, with thermal overload relay and low voltage protection.
3. Where the requirements of the central station supplying the power are such as to necessitate a reduced starting current, a manual or magnetic resistance type starter with low voltage protection is commonly used.

In selecting across-the-line control for single phase capacitor type motors it is usually very desirable to use three pole across-the-line starters. Control for multispeed, single phase capacitor motors may be selected from tables on three phase rating when consideration is given to the increased current and the necessary switching of connections.

PROBLEMS IN PRACTICE

1 ● When motors are being considered as prime movers, what are some of the basic considerations that determine the final selection of the correct unit?

- a. The kind of current available for driving the necessary motors is a primary consideration. There are two groups of motors available for driving the equipment on any job, which are the direct current type or alternating current type. The proper group selection depends entirely on the current available.
- b. It is also necessary to decide whether constant speed or variable speed operation is desired.
- c. Consideration must also be given to the type of service required.
 1. Whether variable torque or constant torque motors will be required.
 2. Whether a high starting torque is required or whether a relatively small starting torque is required.

2 ● When using direct current motors: a. What three types are available as regards their windings; b. What four types are available with reference to their speed characteristics?

- a. Shunt wound, compound wound, and series wound.
- b. Constant speed, adjustable speed, adjustable varying speed, and varying speed.

3 ● With direct current motors as prime movers what type would you use: a. For driving a fan; b. For driving a compressor?

- a. A fan requires a relatively small starting torque, therefore, a shunt wound motor would be ideal for this type service.
- b. A compressor has a constant torque, therefore, a compound wound motor would be the proper selection for this duty.

4 ● What is one of the important factors that should be taken into account when a series wound direct current motor is being considered?

With a series wound motor the speed varies with the load, therefore this type should never be used where there is a possibility of the motor operating without being loaded. The resultant high speed may prove to be dangerous.

5 ● With the use of alternating current motors what two groups are generally considered?

Motors using single and polyphase power supply.

6 ● Under the alternating current group of motors what common types are available: a. For single phase duty; b. For polyphase duty?

- a. Repulsion start, induction run, capacitor type, and split phase.
- b. Squirrel cage induction, automatic start induction, slip ring wound rotor induction, and synchronous type.

7 ● What is the most commonly used of the polyphase motors?

The squirrel cage induction motor is the type most generally used for ordinary application.

8 ● With the use of squirrel cage motors what speed characteristic is available and what construction is used to make these more flexible?

The squirrel cage motor is a constant speed unit and cannot be used for variable speed operation. To make this type of motor more flexible, multispeed units are used quite frequently. These units may be single winding for the two speed unit or for different number of windings depending upon the number and combination of speeds required. For two speed single winding units the second speed is always one half of top speed.

9 ● Differentiate between synchronous speed and full load speed of a motor.

Synchronous speed is the theoretical or no load speed. With the induction motor there is a certain amount of slip depending upon the load. As a rule, at full load, the speed is approximately 96 per cent of synchronous speed, however, motor manufacturers generally list full load speeds on their motor name plates.

The synchronous type of motor has a full load speed which is the same as the synchronous.

10 ● What are the general requirements generally recommended by the power company with reference to connecting polyphase motors to the power line?

For motors five horsepower and smaller, normal torque, normal starting current type of units can be connected directly to the line.

For motors thirty horsepower and smaller, normal torque, low starting current type of units can be used with across-the-line type of control.

Above these sizes it is necessary to furnish current limiting starting compensators.

It is always advisable to check with local power companies as there are no standards for connecting of loads on the power line and they are likely to vary with different power companies.

11 ● In controlling direct current motors what two methods are used, what speed ranges are obtained, and what is the relative efficiency of each method?

In controlling direct current motors, resistance is placed in either the armature circuit or the field circuit. For armature control, the speed is reduced with the increase of re-

sistance. With the field control, the speed is increased with the addition of resistance in the field circuit.

For most listed direct current motors, it is possible to obtain operation up to a speed ratio of two to one with field control equipment. This type of control is used in connection with shunt wound motors for best results.

For speed adjustment by resistance in series with the armature circuit, a reduction of 50 per cent in speed can generally be obtained. This control can be used with either shunt or compound wound motors.

The field control method of changing speeds on direct current motors is the most efficient. Due to the large current in the armature circuit, this method results in a high loss when the speed is reduced any appreciable amount. It is well to remember that with field control only constant horsepower output is obtained, therefore, care should be taken that the motor at normal speed is large enough to care for any increase in load as a result of speeding up the unit.

12 • What reduction in speed is possible and how is it obtained when alternating current slip ring motors are used?

Speed variation in slip ring motors is obtained by inserting resistance in the secondary circuit. This generally allows for a 50 per cent speed reduction when it is fully loaded at normal speed.

Chapter 43

TEST METHODS AND INSTRUMENTS

Pressure Measurement, Temperature Measurement, Air Movement, Humidity Measurement, Carbon Dioxide Determination, Dust Determination, Flue Gas Analysis, Measurement of Smoke Density, Heat Transmission, Eupatheoscope Problems in Practice

SEVERAL types of measuring apparatus are available for accurately determining the thermal capacity and air movement of gaseous vapors and homogenous materials. This chapter gives a brief description of the principal instruments used in connection with the proper control and testing of heating and air conditioning installations.

PRESSURE MEASUREMENT

Atmospheric pressure is usually measured by a *mercurial barometer* which, in its simplest form, consists of a glass tube about 3 ft long, closed at the upper end, filled with mercury and inverted in a shallow bath of mercury. The pressure of the atmosphere on the exposed top of the mercury in the cistern supports a column of mercury in the tube to a height of about 30 in. Readings are taken of the height of the column between the levels of mercury in the tube and in the cistern. Atmospheric pressure is the same as the pressure exerted by this supported column of mercury, and, in pounds per square inch, is equal to its height in inches times 0.491, which is the weight in pounds of 1 cu in. of mercury. At latitude 45 deg and sea level, and at a temperature of 32 F, the atmosphere will support a column of mercury 29.921 in. in height. The pressure of 14.7 lb per square inch, derived by multiplying 29.921 by 0.491, is called *standard* or *normal barometric pressure*. Since the height of the barometer depends on the density of the mercury as well as on the pressure of the atmosphere, and since the density is dependent on the temperature, mercurial barometer readings should always be corrected for temperature. An *aneroid barometer* contains no liquid; it is portable but less accurate than the mercurial barometer. Atmospheric pressure in bending the thin corrugated top of a partially exhausted metallic box, or in distorting a thin-walled bent tube of metal, is made to move a pointer.

Pressures above or below atmospheric are usually measured by means of gages which indicate the difference between the pressure being measured and atmospheric pressure at the same time and place. A gage which indicates pressures higher than atmospheric is known as a *pressure gage*, and a gage which indicates pressures lower than atmospheric is known as a *vacuum gage*. The most common type of these gages contains a flexible hollow metal tube of oval cross section, known as a *Bourdon tube*. When subjected to unequal inside and outside pressures, this tube tends to

straighten out, and a pointer motivated by this straightening indicates the pressure difference on a suitably graduated scale.

High vacuum readings such as are encountered in condenser and steam jet refrigeration practice are commonly obtained by the use of mercury column vacuum gages. When the readings obtained with the mercurial barometer and those with the mercury vacuum gage have both been corrected to 32 F, the difference in the two readings will give the absolute vacuum in inches of mercury. The following equation may be used to make corrections for temperature:

$$h = h_1 [1 - 0.000101 (t_1 - t)] \quad (1)$$

where

h = height of mercury column corrected to temperature t .

h_1 = actual height of mercury column.

t_1 = actual temperature of mercury column.

t = temperature to which column is to be corrected.

A gage which indicates pressures slightly above or below atmospheric is known as a *draft gage*. It is essentially a U tube containing either water, kerosene, alcohol, or mercury, with one leg exposed to the air and the other connected to a point where the pressure is to be determined. When the pressure being read is equal to atmospheric, the level of the liquid in the legs will be the same, indicating a zero gage pressure. When a pressure is applied to one leg, one side will fall and the other will rise an equal amount. The difference in height between the two liquid levels indicates the pressure expressed in inches of liquid used in the gage.

Various forms of high sensitivity draft gages¹ frequently called micromanometers are available for the measurement of small pressure differentials and may be sensitive to pressures as small as 0.001 in. of water. These gages are often useful where measurements are to be made on pressure differentials less than 0.1 in. of water, although their total range may extend as high as 5 to 10 in. of water.

TEMPERATURE MEASUREMENT

In engineering work, *mercurial thermometers* are largely employed to measure the intensity of heat. These depend on the uniform expansion of mercury to indicate changes in temperature. An amount of mercury held in a sealed tube with a bulb at one end will rise to one definite level when immersed in melting ice, and to another definite level when immersed in boiling water. These two points are marked, and the space between them is divided into a number of equal portions, each of which is called a degree. In the Fahrenheit scale, there are 180 deg thus obtained, while the centigrade scale has 100 and the Réaumur has 80. Like divisions are marked off on the column above and below these two determined points in order that a greater range of temperature may be read.

*Thermocouples*² may be used to measure any range of temperatures up to 2,900 F. When two dissimilar metals are joined at two points and a

temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends on the composition of the wires and the difference in temperature between the junctions. A potentiometer or sensitive galvanometer of high resistance connected to the thermocouple will give a deflection which is a function of the temperature difference between the hot and cold junctions. Thermocouples connected in series are called *thermopiles*. Thermocouples for the measurement of high temperatures are calibrated with the aid of the known melting points of pure metals.

Resistance thermometers are suitable for temperature measurements up to 1800 F. These thermometers depend for their operation on the change of resistance with temperature of a platinum, nickel, or copper wire coil, and they are calibrated in the same way as thermocouples.

Pyrometers of various types may be used for temperatures above 500 F. The *mercurial pyrometer* is a thermometer with an inert gas, such as nitrogen or carbon dioxide, above the mercury column to prevent the mercury from boiling. The *radiation pyrometer* consists of a thermopile upon which the radiation from a hot source is focused by a concave mirror or lens. A sensitive galvanometer or potentiometer with a calibrated temperature scale indicates the thermo-electromotive force created by the heat on the thermopile. The *optical pyrometer* measures radiant energy by comparing the intensity of a narrow spectral band, usually red light emitted by the object, with that emitted by a standard light source (electric lamp). *Thermo-electric pyrometers* operate on the same principle as thermocouples. When measuring high temperatures, it is customary to hold the cold junction at room temperature and this may cause some error if the room temperature is above or below the calibration point. For extremely precise temperature measurements, the cold junction is usually immersed in melting ice to fix the cold junction temperature. Various forms of hand-operated and automatic cold junction temperature compensators are also available.

In the measuring of room temperatures care must be exercised to prevent the results from being affected by the body heat of the observer, by drafts from doors, windows and other openings, or by radiant heat from some local source such as a radiator or wall. All glass thermometers should be mercury thermometers with engraved stems. The total graduations of the thermometers should be from 20 to 120 F, in one degree graduations. No ten degrees should occupy a space of less than one-half inch. The accuracy throughout the whole scale must be within one-half degree. The operator should take hold of the top and no part of the body, including the hand, should be nearer than 10 in. to the bulb. The thermometer should not be closer than 5 ft to any door, window, or other opening; should not be closer than 12 in. to any wall; and should be between 3 and 5 ft from the floor. A sling instrument should be used for extreme accuracy. Thermocouples or resistance thermometers may also be used for room temperature measurements, an advantage being that the operator can read temperatures from outside the room if desired, and thus eliminate the errors which might be caused by his presence close to the temperature measuring device.

For measuring duct temperatures a duct thermometer should be used, with the bulb extending into the duct at least 6 in. When the thermo-

¹Illinois Micromanometer, University of Illinois Engineering Experiment Station Bulletin No. 120, p. 91.

²Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by A. P. Kratz and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

meter is to be permanently located in the duct, a pipe flange or nipple should be used to receive the threaded portion of the thermometer stem. When the thermometer is not to be permanently located, a cork or rubber stopper may be placed around the stem to prevent errors from air leakage. Readings should be taken at various locations in a duct so due consideration may be given to temperature stratification. Other forms of temperature measuring devices may be used, but the active part must be at least 6 in. from the duct wall.

Recording instruments may be used for testing and for making continuous records of operation. Potentiometer and Wheatstone bridge recorders for thermocouples and resistance thermometers respectively may have accuracies of $\pm \frac{1}{3}$ per cent of their range, or, for example, to ± 1 F in a range of 0 to 300 F. This accuracy compares favorably with that of other forms of temperature measuring devices.

MEASUREMENT OF AIR MOVEMENT

The quantity, velocity and pressure of air moved by a fan or flowing through a duct or grille may be determined by various methods. The instruments in common use are the Pitot tube, anemometer, direct reading velocity meter, and Kata-thermometer, the latter being suitable for low air velocities and being commonly used for measurements at points where the air is not confined in a duct. Electrical anemometers are also available, operating on the principle of measurement of the variation of resistance of a hot wire cooled to various degrees by air velocities past the wire. The use of calibrated nozzles, orifice plates, and Venturi meters are recognized methods, which, however, have little application in connection with ventilation practice.

Pitot Tube

This usually consists of two tubes, one within the other, which when properly held in the air stream will register the total or impact pressure and the static pressure, respectively. If these tubes are connected to opposite sides of a water column, or other type of manometer, the recorded pressure will be the differential or velocity pressure. Volume measurements may thus be made in a duct of known area. Pitot tube measurements are preferably used for air velocities exceeding 20 fps. Volumetric determinations from Pitot tube readings should take into account the barometric pressure and the temperature and humidity of the air measured.

Air flow in ventilation practice is generally in the turbulent range. When stratification of velocity, vortex motion, or violent eddy currents of air in ducts exist, accurate velocity pressure measurements are difficult. To insure accuracy a straight section of duct from 5 to 10 times its own diameter is desirable in order to straighten out the air currents. If it is necessary to take Pitot tube readings in shorter sections of straight duct, the results must be considered subject to some doubt and checked accordingly. For accurate work it is necessary to make a traverse of the duct, dividing its cross section into a number of imaginary equal areas and taking a reading in the center of each, the average of the velocities corresponding to these pressures giving the true velocity in the duct.

Anemometer

This instrument is delicate, and requires frequent calibration when accuracy is desired. The vanes of the instrument should never be touched and it should never be held in air having a velocity greater than that for which it is calibrated. Readings taken directly in a fan inlet or discharge are likely to harm the instrument because of excessive velocities. In duct measurements the same procedure is followed as for the Pitot tube. The anemometer usually reads directly in linear feet. To obtain the velocity in feet per minute, the reading must be divided by the elapsed time in minutes.

The following procedure for obtaining anemometer readings is based on research conducted at *Armour Institute of Technology* in cooperation with the A.S.H.V.E. Research Laboratory³.

Supply Grilles. The surface of the grille should be marked off into a number of equal areas approximately 6 in. square. A 4-in. anemometer should be used and should be held at the center of each section in contact with the grille (or as close as possible) for a period of time sufficient to insure an average reading. In the case of supply grilles, the instrument should always be held with the dial facing the operator. The average of the corrected readings should then be used in the following formula to obtain the flow in cubic feet per minute:

$$cfm = CV \frac{A + a}{2} \text{ or } \frac{CVA(1 + p)}{2} \quad (2)$$

where

V = average of corrected anemometer readings, feet per minute.

A = gross area of grille, square feet.

a = net free area of grille, square feet.

p = percentage of free area of grille expressed as a decimal.

C = a coefficient that varies with the velocity from grille and may vary slightly with type of grille. For average use, with supply grilles, C can be taken as 0.97 at velocities from 150 to 600 fpm, and as 1.00 at higher velocities.

Particular care should be exercised in the case of long, narrow grilles. The nature of the approach sometimes results in there being a narrow strip along the top or bottom of the grille through which no air will be flowing. This may be detected by holding the anemometer completely out of the air stream and then moving it slowly inward over the grille until the vanes just start to move. The distance which the vanes extend over the grille opening at this moment will indicate the width of the dead strip. Only the remaining portion of the grille should be considered in making the calculations for gross and free area.

Exhaust Grilles. The surface of the grille should be marked off and readings taken in the same manner as with supply grilles, except that the instrument should be held with the dial facing the grille, and in contact with it. The traverse should be taken at a uniform rate, allowing sufficient time in each space to minimize the percentage of error. In the case of exhaust grilles it is found that the formula

$$cfm = KVA \quad (3)$$

³Measurement of Flow of Air through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, Vol. 37, 1931, and Vol. 39, 1933).

in which

- V = average indicated velocity obtained by the anemometer traverse.
 A = gross area of grille, square feet.
 K = coefficient determined by experiment. For average use, with exhaust grilles, K may be taken as 0.8 for all usual velocities.

This formula is of advantage, especially with ornamental grilles, in that the free area need not be measured.

The flow of air through registers and grilles is of considerable importance, being frequently the only convenient method of measuring the volume of supply air to a room. While duct measurements, if available, are more dependable, grille measurements provide a fairly accurate method, if care is taken in the technique of using the anemometer.

Kata-Thermometer

The Kata-thermometer can be used to determine air velocities provided the walls and surrounding objects are at or near the room temperature. Especially at low velocities it constitutes a useful instrument for readily detecting drafts.

The instrument is essentially an alcohol thermometer with a bulb approximately $\frac{5}{8}$ in. in diameter and $\frac{1}{2}$ in. long with a stem 8 in. long reading from 100 F to 95 F, graduated to tenths of a degree. To take readings the bulb is heated in water until the alcohol expands and rises into a top reservoir. The time in seconds required for the liquid to fall from 100 F to 95 F is recorded with a stop watch and this time is a measure of the rate of cooling.

The dry Kata loses its heat by radiation and by convection so for constant velocities the time of cooling is a function of the dry-bulb temperature of the surrounding air. The wet Kata, which has a cloth covering fitted snugly around its bulb, loses heat by radiation, convection, and evaporation, and for constant velocities its rate of cooling is a function of the wet-bulb temperature of the air irrespective of the dry-bulb temperature or relative humidity. It does not follow, however, that the difference in rate of cooling of the dry and the wet Kata is caused by evaporation. A change in the wet-bulb temperature produces a change in the surface temperature of the wet Kata which in turn affects the heat lost by radiation and by convection.

Several precautions should be taken to obtain the best results with this instrument:

1. To obtain velocity readings use the dry Kata since the error in timing is reduced.
2. The instrument should be heated and allowed to cool two or three times before recording the final time of cooling. The first reading is not reliable.
3. All traces of moisture must be removed from the dry Kata before timing to eliminate error introduced by evaporation.
4. Use only the formula applying to a particular instrument. Each Kata receives an individual calibration.

HUMIDITY MEASUREMENT

The sling psychrometer is the recognized standard instrument for determining humidities. In order to obtain accurate readings considerable

skill is required on the part of the operator. The wicking and water must be clean and the temperature of the water should be slightly above the wet-bulb temperature of the surrounding air. The psychrometer should be swung rapidly and several and frequent observations should be made to see that the wet-bulb temperature has become stationary before the final reading is noted. Care should be taken that the wet-bulb has reached a minimum temperature, but the wick must still be moist. Standard psychrometric tables should be used⁴.

In making wet-bulb measurements below 32 F the same procedure is followed as above 32 F. The water is liquid at the start, but as the sling is operated it will freeze rapidly enough so that in quickly giving up the latent heat of fusion, the indicated wet-bulb temperature may drop below the actual wet-bulb temperature. After the liquid on the bulb has become thoroughly frozen the wet-bulb temperature will rise to normal. A very thin film of ice is more desirable than a thick film. Care must be taken to read the temperatures in the region below 32 F accurately because the spread between the wet- and dry-bulb is small.

In taking humidity readings in ducts it is usually impracticable to use a sling psychrometer. For this work the stationary hygrometric arranged for bolting on to the side of the duct, with two bulbs extending into the duct, will be found very convenient. Owing to the velocity of the air passing over the bulbs within the duct an accurate reading will be secured, corresponding to that given by the sling psychrometer.

Various forms of humidity recorders are available, some merely recording wet- and dry-bulb temperatures, and others recording relative humidity directly. Any form of wet- and dry-bulb device must have sufficient air velocity over the thermometer bulbs to insure accurate readings; this velocity should be secured by a fan if the air is not itself in motion, as in a duct. For extremely low humidities, or for humidity measurements above 212 F, a thermal conductivity method is available⁵.

CARBON DIOXIDE DETERMINATION⁶

At ordinary concentrations carbon dioxide is not harmful. The amount of carbon dioxide in the air is a convenient index of the rate of air supply, and of the distribution of the air within rooms. Unequal carbon dioxide concentrations in parts of a room indicate improper air distribution.

The Petterson-Palmquist apparatus has been generally accepted as the standard device for the determination of carbon dioxide in air investigations. The principle involved is the measurement of a given volume of air, the absorption of the contained carbon dioxide in a caustic potash solution, and the remeasurement of the volume of air at the original pressure in a finely graduated capillary tube, the difference in volume representing the absorbed carbon dioxide. (See Report of Committee on Standard Methods for Examination of Air, *American Public Health Association*, Vol. 7, No. 1; *American Journal of Public Health*, Jan., 1917.)

⁴Psychrometric Tables for Vapor Pressure, Relative Humidity and Temperatures of the Dew Point; U. S. Department of Agriculture, Weather Bureau, Washington, D. C.

⁵Gas Analysis by Measurement of Thermal Conductivity, H. A. Daynes, Cambridge Press, 1933.

⁶Indices of Air Changes and Air Distribution, by F. C. Houghton and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).

A thermal conductivity method may also be used to measure carbon dioxide in air over a range of 0 to 1.5 per cent⁵.

Where field conditions are such that this apparatus may not be conveniently used, as in street cars, air samples may be collected in clean bottles having mercury-sealed rubber stoppers, and these may be subjected to laboratory analysis.

DUST DETERMINATION

Many laboratory methods have been developed to measure the dust in the air. These involve the collection of dust on sticky plates, on filter paper, in water, on porous crucibles, or by electric precipitation, and the subsequent determination of the amount of dust by microscopic counting, weighing, or titration. While there is no standard method, the Hill dust counter, using a microscope, the impinger⁷, using chemical changes in water, and the Lewis sampling tube⁸, involving the analytical weighing of a porous crucible, are accepted. All test results should be accompanied by the name of the instrument used as great variation in counts with the different instruments will be obtained. The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has developed a code⁹ for the testing and rating of air cleaning devices used in general ventilation work.

FLUE GAS ANALYSIS

The analysis of flue gases by chemical means is made with the *Orsat apparatus*. A solution of *KOH* is used to absorb the CO_2 . Free oxygen is absorbed by a mixture of pyrogallic acid and *KOH*. The solution for absorbing the CO is cuprous chloride. The apparatus consists of a burette surrounded by a water jacket, to receive and measure the volume of gas. The burette is connected by a manifold of glass to *pipettes* containing liquids for absorbing CO_2 , O_2 and CO .

Various forms of automatic indicating and recording gas analysis devices are available, operating on either chemical or physical principles. Such devices are convenient for plant operation.

MEASUREMENT OF SMOKE DENSITY

Relative smoke density is usually measured by comparison with the Ringelmann Chart (Fig. 1). In making observations of the smoke issuing from a chimney, four cards ruled like those in Fig. 1, together with a card printed in solid black and another left entirely white, are placed in a horizontal row and hung at a point 50 ft from the observer and conveniently in line with the chimney. At this distance, the lines become invisible, and the cards appear to be of different shades of gray, ranging from very light gray to almost black. The observer glances from the smoke coming from the chimney to the cards, which are numbered from 0 to 5, determines which card most nearly corresponds with the color of the smoke, and makes a record accordingly, noting the time. Observa-

tions are made continuously during one minute, and the estimated average density during that minute recorded. The average of all the records made during a boiler test is taken as the average figure for the smoke density during the test, and the entire record is plotted on cross-section paper in order to show how the smoke varied in density from time to time.

Smoke Recorders

Smoke recorders are available which give a much more accurate indication of the amount of smoke being produced than does the Ringelmann Chart. They all depend upon projecting a beam of light through the smoke flue or through a separate compartment from which a sample of the flue gas is drawn continuously. The light of the beam which passes

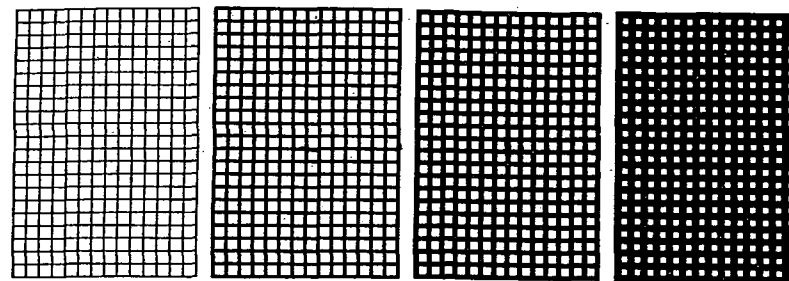


FIG. 1. RINGELMANN SMOKE CHART

through without being absorbed by the smoke is measured to determine the smoke density. Most of these instruments make use of a photo-electric cell or a thermopile to measure the relative amount of light which has not been absorbed. Standard electrical instruments serve for indicating or recording.

MEASUREMENT OF RATE OF HEAT TRANSMISSION

The standard methods of testing built-up wall sections are by means of the *guarded hot-box*¹⁰ and the *guarded hot-plate*¹¹. The *Nicholls heat-flow meter*¹¹ may be used for testing actual walls of buildings.

It would be obviously impossible to determine the air-to-air heat transmission coefficients of every type of wall construction in use with the heat-flow meter, the guarded hot-box or the guarded hot-plate on account of the great amount of time involved. Hence, the method of computing the coefficients from the fundamental constants must be resorted to in most cases. The guarded hot-plate is used to determine the fundamental constants. The heat-flow meter, guarded hot-box and guarded hot-plate tests can be used to good advantage in checking the accuracy of the computed values.

⁵Public Health Bulletin, No. 144, 1925, U. S. Public Health Service.

⁸Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933).

⁹See 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).

¹⁰See Standard Code for Heat Transmission through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928) and Report of the Committee on Heat Transmission, National Research Council.

¹¹See Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924).

If the hot-box or hot-plate methods are used, tests are usually run under still air conditions, which means there is no wind movement over the surfaces of the wall during the test. In the hot-plate method of test the inside surface coefficient is eliminated by the plate's being in direct contact with the wall. In practice, some wind movement over the exterior surface of the wall should always be allowed for; hence, still-air coefficients cannot be used over the outside of the building during the heating season. Moreover, still-air transmission coefficients cannot be corrected to provide for moving-air conditions by applying a single constant factor. Computed coefficients of transmission for various types of construction are given in Chapter 5.

EUPATHEOSCOPE

The eupatheoscope affords a means of evaluating the combined effect of radiation and convection in a given environment in terms of a standard environment and in some terms related to human comfort. See Chapter 38.

PROBLEMS IN PRACTICE

1 ● What is the corrected barometric pressure of the atmosphere at 32 F when a mercurial barometer reading of 29.51 in. Hg, is determined in a room having a temperature of 91 F?

Substitute in Equation 1. $h = 29.51 [1 - 0.000101 (91 - 32)]$.
 $h = 29.33$ in. Hg.

2 ● What advantages other than its sensitiveness, has the U-tube draft gage or manometer for measurement of low pressures?

Inherent accuracy without calibration and low cost of the essential parts, which are glass tubing and an ordinary scale.

3 ● Are thermocouples as accurate as mercury thermometers?

Within the range which can be measured with both instruments (below 1000 F) either one may be made as sensitive as the service requires. The accuracy of a thermocouple temperature measurement depends chiefly on: (1) an accurate calibration of the wire, (2) the sensitiveness of the electrical instrument, (3) accurate cold-junction control, and (4) proper placement of the sensitive junction.

4 ● When an anemometer is used for measuring the air discharged from a grille or register, does it read the velocity through the gross face area or the velocity through the net free area?

Neither. If either of these velocities is required, it should be calculated by means of Equation 2.

5 ● Do common errors made in humidity determination produce a result that is too high or too low?

A higher relative humidity than the true value is likely to be found, either because there is insufficient velocity over the wet-bulb or because the reading is not taken at the right time.

6 ● What is the purpose of the carbon dioxide determination?

It is an index of the adequacy of fresh air supply and also an indicator of air distribution.

Chapter 44

TERMINOLOGY

Glossary of Physical and Heating and Ventilating Terms Used in the Text, Standard Abbreviations, Conversion Equations, Drafting Symbols, A.S.H.V.E. Codes

Absolute Humidity: See *Humidity*.

Absolute Pressure: The sum, at any particular time, of the gage pressure and the atmospheric pressure.

Absolute Temperature: The temperature of a substance measured above *absolute zero*.

Absolute Zero: The temperature (-459.6 F) at which the molecular motion of a substance theoretically ceases. This is the temperature at which the substance theoretically contains no heat energy.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second per second.

$$a = \frac{v}{t}$$

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body. In the fps system this is 32.174 feet per second per second.

Adiabatic: An adjective pertaining to or designating variations in volume or pressure not accompanied by gain or loss of heat. When a substance undergoes adiabatic expansion, since it does not receive heat from without, the work which it does is at the expense of its internal energy, and therefore its temperature falls; similarly, when it is adiabatically compressed its temperature rises.

Adsorption: The adhesion of the molecules of gases or dissolved substances to the surfaces of solid bodies, resulting in a concentration of the gas or solution at the place of contact.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air 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, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Air Infiltration: The inleakage of air through cracks and crevices, and through doors, windows and other openings, caused by wind pressure or temperature difference.

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

Anemometer: An instrument for measuring the velocity of moving air.

Atmospheric Pressure: The pressure exerted by the atmosphere in all directions, as indicated by a barometer. *Standard atmospheric pressure* is considered to be 14.7 lb per square inch, which is equivalent to 29.92 in. of mercury.

Baffle: A plate or wall for deflecting gases or fluids.

Blast: This word was formerly used to denote forced air circulation, particularly in connection with central fan systems using steam or hot water as the heating medium. As applied in this sense, the word *blast* is now obsolete.

Boiler: A closed vessel in which steam is generated or in which water is heated.

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 $970.2 \times 34.5 = 33,471.9$ Btu per hour.

British Thermal Unit: The *mean* British thermal unit is $\frac{1}{180}$ of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F. It is substantially equal to the quantity of heat required to raise 1 lb of water from 63 F to 64 F. One Btu = $\frac{1}{3415}$ kwhr.

By-pass: A pipe or duct, usually controlled by valve or damper, for short-circuiting fluid flow.

Calorie: The *mean* calorie is $\frac{1}{100}$ of the heat required to raise the temperature of 1 gram of water from Zero C to 100 C. It is substantially equal to the quantity of heat required to raise one gram of water from 14.5 C to 15.5 C.

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 distribution ducts. See Chapters 9 and 22.

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Transmission: The amount of heat (Btu) transmitted from air to air in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

Column Radiator: A type of direct radiator. This radiator has not been listed by manufacturers since 1926.

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

Comfort Zone (Average): The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. **Comfort Zone (Extreme):** The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 3.)

Concealed Radiator: See *Convactor*.

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 F between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material 1 in. thick for a difference in temperature of 1 F between the two surfaces of the material.

Conductor (heat): A material capable of readily conducting heat. The opposite of an insulator or insulation.

Constant Relative Humidity 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.

Control: Any manual or automatic device for the regulation of a machine to keep it at normal operation. If automatic, it is considered that the device is motivated by variations in temperature, pressure, time, light, or other influences.

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. Convection may be *natural* or *forced*.

Convactor: A concealed *radiator*. A heating unit and an enclosure or shield located either within, adjacent to, or exterior to the room or space to be heated, but transferring heat to the room or space mainly by the process of convection. If the heating unit is located exterior to the room or space to be heated, the heat is transferred through one or more ducts or pipes; see Chapter 30.

Corrosive: Having the power to wear away or gradually change the texture or substance of a material.

Decibel: The standard unit for noise or sound intensity. One decibel is equal to ten times the logarithm to the base *e* of the ratio of the sound intensities.

Degree-Day: A unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exist as many degree-days as there are degrees Fahrenheit difference in temperature between the average outside air temperature, taken over a 24-hour period, and a temperature of 65 F.

Dehumidify: To remove water vapor from the atmosphere; to remove water vapor or moisture from any material.

Density: The weight of a unit volume, expressed in pounds per cubic foot. $d = \frac{W}{V}$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Diffuser: A vaned device placed at an air supply opening to direct the air flow.

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 Radiator: Same as *radiator*.

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 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.

Draft Head (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 Air: Air with which no water vapor is mixed. This term is used comparatively, since in nature there is always some water vapor included in air, and such water vapor, being a gas, is dry.

Dry-Bulb Temperature: The temperature of the air indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air.

Dry Return: 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 *wet return*.

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: The total or impact pressure. This is the sum of the radial pressure and the velocity pressure at the point of measurement.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity,

and movement of the air. Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Enthalpy: Total heat or thermal potential.

Entropy: The logarithmic probability of a state. It is the integration between two absolute temperatures of the quotient of the quantity of heat divided by the absolute temperature at the condition at which the temperature is taken. It is, therefore, a numeric which explains a difference in conditions between two points in a heat cycle.

Entropy, which can vary with temperature, volume, or pressure, is constant during adiabatic expansion in a reversible cycle or during isentropic expansion in an irreversible cycle. Entropy is a function of the unavailable energy in any system.

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.

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined. It is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932.)

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932.)

Extended Heating Surface: See *Heating Surface*.

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being soldered to the core or by both pressure and soldering. An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector.

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

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = \frac{WV}{gt}$.

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also, a fire-pot.

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.*)

Gage Pressure: 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.

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 defined above. (*A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers.*)

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

Grille: A perforated covering for an air inlet or outlet usually made of wire screen, pressed steel, cast-iron or plaster. Grilles may be plain or ornamental.

Heat: A form of energy generated by the transformation of some other form of energy, as by combustion, chemical action, or friction. According to the molecular theory, heat consists of the kinetic and potential energy of the molecules of a substance. The addition of heat energy to a body increases the temperature or the kinetic energy of motion of its molecules (*sensible heat*) or increases their potential energy of position but does not increase the temperature, as when melting or boiling occurs (*latent heat*).

Heat Capacity: The amount of heat (Btu or calories) required to raise the temperature of a body of any mass and variety of parts one degree (Fahrenheit or centigrade). This will depend on the masses and specific heats of the various parts of the body.

Therefore

$$S = m_1 s_1 + m_2 s_2 + m_3 s_3 \dots \text{etc.}$$

where

S is the heat capacity and m_1, m_2, m_3 , and s_1, s_2, s_3 stand for the masses and corresponding specific heats of the parts, respectively.

Heating Medium: A substance such as water, steam, air, electricity

or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface (or extended surface):* Heating surface having air on both sides and heated 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.*)

Heat of the Liquid: The sensible heat of a mass of liquid above an arbitrary zero.

Horsepower: A unit to indicate the time rate of doing work equal to 550 ft-lb per second or 33,000 ft-lb per minute. (One horsepower = 745.8 watts. In practice this is considered 746 watts.)

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

Humidify: To add water vapor to the atmosphere; to add water vapor or moisture to any material.

Humidity: The water vapor mixed with dry air in the atmosphere. *Absolute humidity* refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. *Specific humidity* refers to the weight of water vapor in pounds carried by one pound of dry air. *Relative humidity* is a ratio, usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. Relative humidity is either the ratio of the actual partial pressure of the water vapor in the air to the saturation pressure at the dry-bulb temperature, or the ratio of the actual density of the vapor to the density of saturated vapor at the dry-bulb temperature. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space.

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

Hygrostat: Same as humidistat.

Inch of Water: A measure of pressure which refers to the difference in the heights of the legs of a water filled manometer.

Insulation (heat): A material having a relatively high heat-resistance per unit of thickness.

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.

Latent Heat: See *Heat.*

Laws of Thermodynamics: The *first law* states that the total energy of an isolated system remains constant and cannot be increased or diminished by any physical process whatever. The *second law* states that no change in a system of bodies that takes place of itself can increase the available energy of a system.

Manometer: An instrument for measuring pressures; essentially a U-tube partially filled with a liquid, usually water, mercury, or a light

oil, so the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second.

$$m = \frac{W}{g}$$

Mb, Mbh¹: Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour.

Mechanical Equivalent of Heat: The mechanical energy necessary to produce 1 Btu of heat energy. $J = 777.5$ ft-lb.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol: The unit of weight for gases. It is defined as m lb where m denotes the molecular weight of a gas. For any gas the volume of 1 *mol* at 32 F and standard atmospheric pressure is 358.65 cu ft and the weight of a cubic foot is 0.002788 m lb.

Neutral Zone: The level within a room or building at which the pressure is exactly equal to the outside barometric pressure.

One-Pipe Supply Riser (steam): A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System (hot water): A hot water system in which the water flows through more than one heating unit before it returns to the boiler; consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

One-Pipe System (steam): A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe, usually in opposite directions. 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.

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

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

Panel Warming: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room, so that the heating process takes place mainly by radiation from the wall, floor or ceiling surfaces to the objects in the room.

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

Potentiometer: An instrument for measuring or comparing small electromotive forces.

Power: The rate of performing work, expressed in units of horsepower, one of which is equal to 550 ft-lb of work per second, or 33,000 ft-lb per minute.

¹These symbols were approved by the A.S.H.V.E., June, 1933.

Prime Surface: See *Heating Surface*.

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmosphere as to moisture. *Psychrometry:* The branch of physics that treats of the measurement of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit exposed to view within the room or space to be heated. A radiator transfers heat by radiation to objects "it can see" 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. *Concealed Radiator:* See *Convector*.

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

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

Register: A grille with a built-in multiblade damper or shutter.

Relative Humidity: See *Humidity*; see also discussion of relative humidity in Chapter-1.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reversed-Return System (hot water): A hot water heating system in which the water from several heating units is returned along paths arranged so that all circuits composing the system or composing a major subdivision of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to permit egress of air.

Saturated Air: Air containing as much water vapor as it can hold without any condensing out; in saturated air, the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature.

Sensible Heat: See *Heat*.

Smoke: 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.

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The quantity of heat, expressed in Btu, required to raise the temperature of 1 lb of a substance 1 F.

Specific Volume: The volume, expressed in cu ft, of one pound of a substance. $v = \frac{1}{d} = \frac{V}{W}$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point.

Square Foot of Heating Surface (equivalent): Equivalent direct radiation (EDR). By definition, that amount of heating surface which will give off 240 Btu per hour. The *equivalent* square feet of heating surface may have no direct relation to the actual surface area.

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

Standard Air: As defined by A.S.H.V.E. codes, *standard air* is air weighing 0.07488 lb per cubic foot, which is air at 68 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.92 in. of mercury. (Most engineering tables and formulae involving the weight of air are based on air weighing 0.07495 lb per cubic foot, which is dry air at 70 F dry-bulb with a barometric pressure of 29.92 in. of mercury. The error involved in disregarding the difference between the above two weights is very slight and in most instances may be neglected.)

Static Pressure: The compressive pressure existing in a fluid. It is a measure of the potential energy of the fluid.

Steam: Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. Steam in contact with the water from which it has been generated may be *dry saturated* steam or *wet saturated* steam. The latter contains more or less actual water in the form of mist. If steam is heated, and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated*.

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

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

Superheated Steam: See *Steam*.

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

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction, and convection from a surface to the air or liquid surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid.

Synthetic Air Chart: A chart for evaluating the air conditions maintained in a room.

Therm: Symbol used in the gas industry representing 100,000 Btu.

Thermal Resistance: The reciprocal of *conductance*.

Thermal Resistivity: The reciprocal of *conductivity*.

Thermodynamics: The science which treats of the mechanical actions or relations of heat.

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Ton of Refrigeration: The extraction of 12,000 Btu per hour.

Ton Day of Refrigeration: The heat removed by a ton of refrigeration operating for one day; 288,000 Btu.

Total Heat: A thermodynamic quantity, variously called heat content, thermal potential, enthalpy. It is the heat required per unit mass (Btu per lb) to raise a given substance to a given point from an arbitrary datum point. It is the sum of the heat of the liquid, the latent heat, and any miscellaneous heat which may be present.

Total Pressure: The sum of the static and velocity pressures in a fluid. It is a measure of the total energy of the fluid.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

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.

Underfeed Distribution System (hot water): A hot water heating system in which the main flow pipe is below the heating units.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit Air Conditioner: A piece of equipment designed to provide simultaneous control of at least four of the seven functions (page 237) involved in summer and winter air conditioning. The apparatus is compactly housed in a cabinet placed within or immediately adjacent to the rooms served. The parts comprising a unit air conditioner are assembled at the point of manufacture, and the performance of the assembly is the responsibility of the manufacturer. See Chapter 12.

Unit Cooler: A cooling device, usually comprising an extended-surface element and a motor-driven fan mounted integrally in a housing, located within or adjacent to the room served. Generally no ducts are attached to inlet or outlet. The refrigerant is brought to the unit from an outside source, and the fan drives air over the cooling element.

Unit Heater: A heating device, usually comprising an extended-surface element or a gas burner, mounted with a motor-driven fan in a housing, located within or adjacent to the room served. Generally, no ducts are attached to inlet or outlet. The fluid for heating is brought to the unit from an outside source, and the fan drives air over the heating element. Unit heaters are used primarily in industrial applications.

Unit Ventilating-Heater: A ventilating and heating device comprising a motor-driven fan, an extended-surface heating element and usually a filter, mounted in a housing, located within or adjacent to the room served. Outdoor air is obtained through a dampered direct connection or a short duct from a nearby wall or window opening. Provision

for partial recirculation is usually made. If a humidifier is included, such a filter-equipped device becomes a winter-type unit conditioner. Unit ventilators are used primarily for offices, schools, and places of public assembly.

Up-Feed System (steam): A steam 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.

Vapor: Any substance in the gaseous state.

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. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Vapor Pressure: The equilibrium pressure exerted by a vapor in contact with its liquid.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Pressure: The pressure corresponding to the velocity of flow. It is a measure of the kinetic energy of the fluid.

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*.)

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. If 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, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers and filters.

Wet-Bulb Temperature: The lowest temperature which a water wetted body will attain when exposed to an air current. This is the temperature of adiabatic saturation.

Wet Return: 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 *Dry Return*.

ABBREVIATIONS²

Absolute.....	abs
Acceleration, due to gravity.....	g
Acceleration, linear.....	a
Air horsepower.....	air hp
Alternating-current (as adjective).....	a-c
Ampere.....	amp
Ampere-hour.....	amp-hr
Area.....	A
Atmosphere.....	atm
Average.....	avg
Avoirdupois.....	avdp
Barometer.....	bar.
Boiler pressure.....	bp
Boiling point.....	bp
Brake horsepower.....	bhp
Brake horsepower-hour.....	bhp-hr
British thermal unit.....	Btu
Calorie.....	cal
Centigram.....	cg
Centimeter.....	cm
Centimeter-gram-second (system).....	cgs
Change in specific volume during vaporization.....	vg
Cubic.....	cu
Cubic foot.....	cu ft
Cubic feet per minute.....	cfm
Cubic feet per second.....	cfs
Decibel.....	db
Degree.....	deg or °
Degree centigrade.....	C
Degree Fahrenheit.....	F
Degree Kelvin.....	K
Degree Réaumur.....	R
Density, Weight per unit volume, Specific weight.....	d or ρ (rho)

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

Diameter.....	D or diam
Direct-current (as adjective).....	d-c
Distance, linear.....	s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, Vapor in contact with liquid.....	Subscript g
Entropy (The capital should be used for any weight, and the small letter for unit weight.).....	S or s
Feet per minute.....	fpm
Feet per second.....	fps
Foot.....	ft
Foot-pound.....	ft-lb
Foot-pound-second (system).....	fps
Force, total load.....	F
Freezing point.....	fp
Gallon.....	gal
Gallons per minute.....	gpm
Gallons per second.....	gps
Gram.....	g
Gram-calorie.....	g-cal

²From compilations of abbreviations approved by the American Standards Association, Z, 10 a, c, f, and i. As a general rule the period is omitted in all abbreviations except where the omission results in the formation of an English word.

³It is recommended that the abbreviation for the temperature scale, F, C, K, be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted; as 68 F.

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}
Horsepower	hp
Horsepower-hour	$hp-hr$
Inch	$in.$
Inch-pound	$in-lb$
Indicated horsepower	$i hp$
Indicated horsepower-hour	$i hp-hr$
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight.)	U or u
Kilogram	kg
Kilowatt	kw
Kilowatt-hour	$kwhr$
Length of path of heat flow, thickness	L
Load, total	W
Mass	m
Mechanical efficiency	e_m
Mechanical equivalent of heat	J
Melting point	mp
Meter	m
Micron	μ (mu)
Miles per hour	mph
Minute	min
Molecular weight	$mol. wt$
Mol	mol
Ounce	oz
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
Revolutions per minute	rpm
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor	<i>Subscript f</i>
Specific gravity	$sp\ gr$
Specific heat	$sp\ ht$ or 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
Square foot	$sq\ ft$
Square inch	$sq\ in.$
Temperature (ordinary) F or C. (<i>Theta</i> is used preferably only when t is used for Time in the same discussion.)	t or θ (<i>theta</i>)
Temperature (absolute) F abs or K. (Capital <i>theta</i> is used preferably only when small <i>theta</i> is used for ordinary temperature.)	T or Θ (<i>capital theta</i>)
Thermal conductance ⁴ (heat transferred per unit time per 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 per unit area per degree) C_a

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

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

Thermal conductivity (heat transferred per unit time per unit area, and per degree per unit length) k

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

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer (heat transferred per unit time per unit area per 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 per unit area per 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 (degrees 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

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

Watt w

Watthour whr

Weight of a major item, Total weight W

Weight rate, Weight per unit of power, Weight per unit of time w

Work (total) W

CONVERSION EQUATIONS

Fahrenheit degrees = 9/5 (centigrade degrees + 32).

Centigrade degrees = 5/9 (Fahrenheit degrees - 32).

Absolute temperature, expressed in Fahrenheit degrees = Fahrenheit degrees + 459.6. In heating and ventilating work, 460 is usually used.

Absolute temperature, expressed in centigrade degrees = centigrade degrees + 273.1.

Power, Heat, and Work

1 ton refrigeration

= 12,000 Btu per hour
200 Btu per minute

Latent heat of ice

= 143.33 Btu per pound

1 Btu	=	$\begin{cases} 777.5 \text{ ft-lb} \\ 0.293 \text{ watthours} \\ 252.02 \text{ mean calories} \end{cases}$
1 watthour	=	$\begin{cases} 2,655.2 \text{ ft-lb} \\ 3.415 \text{ Btu} \\ 3600 \text{ joules} \\ 860.648 \text{ mean calories} \end{cases}$
1 mean calorie	=	$\begin{cases} 0.003968 \text{ Btu} \\ 3.085 \text{ ft-lb} \\ 0.0011619 \text{ watthours} \end{cases}$
1 kilowatt (1000 watts)	=	$\begin{cases} 1.3405 \text{ horsepower} \\ 56.92 \text{ Btu per minute} \\ 44,252.7 \text{ ft-lb per minute} \end{cases}$
1 horsepower	=	$\begin{cases} 0.746 \text{ kilowatt} \\ 42.44 \text{ Btu per minute} \\ 33,000 \text{ ft-lb per minute} \\ 550 \text{ ft-lb per second} \end{cases}$
1 boiler horsepower	=	33,471.9 Btu per hour

Weight and Volume

1 gal (U. S.)	=	$\begin{cases} 231 \text{ cu in.} \\ 0.13368 \text{ cu ft} \end{cases}$
1 British or Imperial gallon	=	277.274 cu in.
1 cu ft	=	$\begin{cases} 7.4805 \text{ gal} \\ 1728 \text{ cu in.} \end{cases}$
1 cu ft water at 60 F	=	62.37 lb
1 cu ft water at 212 F	=	59.76 lb
1 gal water at 60 F	=	8.34 lb
1 gal water at 212 F	=	7.99 lb
1 lb (avdp)	=	$\begin{cases} 16 \text{ oz} \\ 7000 \text{ grains} \end{cases}$
1 bushel	=	1.244 cu ft
1 short ton	=	2000 lb
1 long ton	=	2240 lb

Pressure

1 lb per square inch	=	$\begin{cases} 144 \text{ lb per square foot} \\ 2.0416 \text{ in. mercury at 62 F} \\ 2.309 \text{ ft water at 62 F} \\ 27.71 \text{ in. water at 62 F} \end{cases}$
1 oz per square inch	=	$\begin{cases} 0.1276 \text{ in. mercury at 62 F} \\ 1.732 \text{ in. water at 62 F} \end{cases}$
1 atmosphere	=	$\begin{cases} 14.7 \text{ lb per square inch} \\ 2116.3 \text{ lb per square foot} \\ 33.974 \text{ ft water at 62 F} \\ 30 \text{ in. mercury at 62 F} \\ 29.921 \text{ in. mercury at 32 F} \end{cases}$
1 in. water at 62 F	=	$\begin{cases} 0.03609 \text{ lb per square inch} \\ 0.5774 \text{ oz per square inch} \\ 5.196 \text{ lb per square foot} \end{cases}$
1 ft water at 62 F	=	$\begin{cases} 0.433 \text{ lb per square inch} \\ 62.355 \text{ lb per square foot} \end{cases}$
1 in. mercury at 62 F	=	$\begin{cases} 0.491 \text{ lb per square inch} \\ 7.86 \text{ oz per square inch} \\ 1.131 \text{ ft water at 62 F} \\ 13.57 \text{ in. water at 62 F} \end{cases}$

Metric Units

1 cm	=	0.3937 in.
1 in.	=	2.54 cm
1 m	=	3.281 ft
1 ft	=	0.3048 m
1 sq cm	=	0.155 sq in.
1 sq in.	=	6.45 sq cm
1 sq m	=	10.765 sq ft
1 sq ft	=	0.0929 sq m
1 cu cm	=	0.061 cu in.
1 cu in.	=	16.39 cu cm
1 cu m	=	35.32 cu ft
1 cu ft	=	0.0283 cu m
1 liter	=	1000 cu cm = 0.264 gal
1 kg	=	2.2046 lb
1 lb	=	0.4536 kg
1 metric ton	=	2205 lb (avdp)
1 gram	=	980.59 dynes = 0.002205 lb
1 kilometer per hour	=	0.6214 mph
1 gram per square centimeter	=	$\begin{cases} 0.0290 \text{ in. mercury, at 0 deg C} \\ 0.394 \text{ in. water, at 15 C} \end{cases}$
1 kg per square centimeter (metric atmosphere)	=	14.22 lb per square inch
1 gram per cubic centimeter	=	$\begin{cases} 0.03614 \text{ lb per cubic inch} \\ 62.43 \text{ lb per cubic foot} \end{cases}$
1 dyne	=	0.00007233 pounds
1 joule	=	$\begin{cases} 10,000,000 \text{ ergs} \\ 0.73767 \text{ ft-lb} \end{cases}$
1 metric horsepower	=	$\begin{cases} 75 \text{ kg-m per second} \\ 0.986 \text{ hp (U. S.)} \end{cases}$
1 kilogram-calorie (large calorie)	=	$\begin{cases} 1000 \text{ gram-calories (small calorie)} \\ 3.97 \text{ Btu} \end{cases}$
1 kilogram-calorie per kilogram	=	1.8 Btu per pound
1 gram-calorie per square centimeter	=	3.687 Btu per square foot
1 gram-calorie per square centimeter per centimeter	=	1.451 Btu per square foot per inch
1 gram-calorie per second per square centimeter for a temperature graduation of 1 deg C per centimeter	=	$\begin{cases} 2903 \text{ Btu per hour per square foot} \\ \text{for a temperature graduation of} \\ 1 \text{ deg F per inch of thickness.} \end{cases}$

SYMBOLS FOR HEATING AND VENTILATING DRAWINGS*

1. The objects of this standard set of symbols are to insure the correct interpretation of drawings and to conserve drafting room time by establishing simple and unmistakable symbols for the component parts of the heating and ventilating systems. In preparing the list of symbols an effort has been made to follow existing practice in so far as possible but the list cannot be expected to match exactly the existing practice of every drafting room.

2. Simplicity, ease of execution and unmistakable identification were carefully considered in selecting the symbols. Uncommon fittings and appliances such as vacuum pumps, separators, etc., have purposely been omitted in order to produce a list which can be easily remembered. It is assumed that when the scale of the drawing permits, the valves and fittings will be drawn to scale and a conventional representation is then unnecessary.

*From A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, edition of 1929.

3. High pressure steam supply pipe	
4. Low pressure steam supply pipe	
5. Hot water pipe—flow	
6. Return pipe—steam or water	
7. Air vent line	
8. Flanges	
9. Screwed union	
10. Elbow	
11. Elbow—looking up	
12. Elbow—looking down	
13. Tee	
14. Tee—looking up	
15. Tee—looking down	
16. Gate valve	
17. Globe valve	
18. Angle valve	
19. Angle valve—stem perpendicular	
20. Lock shield valve	
21. Check valve	
22. Reducing valve	
23. Diaphragm valve	

24. Diaphragm valve—stem perpendicular	
25. Thermostat	
26. Radiator trap—elevation	
27. Radiator trap—plan	
28. Expansion joint	
29. Column radiator—plan	
30. Column radiator—elevation	
31. Wall radiator—plan	
32. Wall radiator—elevation	
33. Pipe coil—plan	
34. Pipe coil—elevation	
35. Indirect radiator—plan	
36. Indirect radiator—elevation	
37. Supply duct—section	
38. Exhaust duct—section	
39. Butterfly damper—plan (or elevation)	
40. Butterfly damper—elevation (or plan)	
41. Deflecting damper—square pipe	
42. Vanes	
43. Air supply outlet	
44. Exhaust outlet	

A.S.H.V.E. CODES

The following codes and standards relating to the design, installation, testing, rating, and maintenance of materials and equipment used for the heating and ventilation of buildings, have been adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Air Cleaning Devices	A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work	June, 1933	A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933
Air Purity	Synthetic Air Chart	June, 1917	A.S.H.V.E. TRANSACTIONS, Vol. 23, p. 607, and THE GUIDE, 1931
Boilers (testing)	Standard and Short-Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2)	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929
Boilers (testing)	A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3) ^a	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929
Boilers—Oil Fuel (testing)	A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931
Boilers (rating)	A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand Fired Boilers	January, 1929 Revised April, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930; p. 42
Concealed Gravity Type Radiation	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Hot Water Section)	June, 1933	A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933
Convectors	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)	January, 1931	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367
Ethics	Code of Ethics for Engineers	January, 1922	A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 6 (See frontispiece THE GUIDE, 1936)
Fans	Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers	May, 1923 Revised June, 1931	A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407 ^b
Garages	Code for Heating and Ventilating Garages	June, 1929 Revised January, 1935	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 355 A.S.H.V.E. Reprint

^aOriginally adopted by the National Boiler and Radiator Manufacturers Association.

^bAlso, see *Heating, Piping and Air Conditioning*, August, 1931, p. 743.

CHAPTER 44—TERMINOLOGY

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Heat Transmission Through Walls	Standard Test Code for Heat Transmission through Walls	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253
Minimum Requirements	Code of Minimum Requirements for Heating and Ventilation of Buildings, Edition—1929	June, 1925	A.S.H.V.E. Codes
Pitot Tube	Code for Use of Pitot Tube	January, 1914	A.S.H.V.E. TRANSACTIONS, Vol. 20, 1914, p. 211
Radiators	Code for Testing Radiators	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 18
Unit Heaters	Standard Code for Testing and Rating Steam Unit Heaters ^c	January, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165
Unit Ventilators	A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25
Vacuum Heating Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps	June, 1934	A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934
Ventilation	Report of Committee on Ventilation Standards	August, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 383

The following Codes and Standards have been endorsed or approved by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	SPONSORED BY	REFERENCE
Chimneys	Standard Ordinance for Chimney Construction	National Board of Fire Underwriters	Chapter 14, THE GUIDE, 1931
Piping Systems	Identification of Piping Systems ^d	American Society of Mechanical Engineers	<i>Heating, Piping and Air Conditioning</i> , July, 1929
Warm Air Furnaces	Standard Code Regulating the Installation of Gravity Warm Air Furnaces in Residences	National Warm Air Heating and Air Conditioning Association	National Warm Air Heating and Air Conditioning Association, Columbus, Ohio

^cAdopted jointly by the Industrial Unit Heater Association, and the A.S.H.V.E.

^dAdopted November, 1928. Sponsored by (1) American Society of Mechanical Engineers, (2) National Safety Council.

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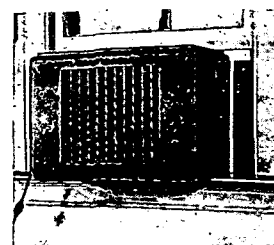
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Aeriet Air Conditioner Company

A Subsidiary of AUTOMATIC PRODUCTS CORPORATION

Manufacturers Air Conditioning—Heating—Ventilating—Humidifying Equipment

Executive Offices: 210 S. Clark Street, Chicago, Ill., U. S. A.



Aeriet Window Unit

AERIET AIR CONDITIONING UNITS

AERIET (Air Cooled) WINDOW UNIT

Unit may be installed in any average double-hung sash window. Rests on window sill. Installation complete in few minutes. Operates from any convenience outlet. No special wiring or water connections required. A compact efficient standardized comfort cooling unit for limited service.

AERIET CEILING UNITS

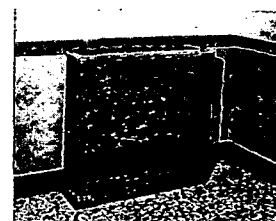
Designed for installation where floor space is limited. Lighting may be combined when desired. Aeriet Ceiling Units are for comfort cooling service only. They are not adaptable for heating. Compact, efficient, economical to install and operate. Capacities one to five tons. (Melting Ice Equivalent.)



Ceiling Aeriet with Lights



Ceiling Aeriet without Lights



Aeriet Floor Unit

AERIET FLOOR UNITS (Remote Supply)

Scientifically developed and properly engineered mechanical units combining every facility and requirement for complete year round air-conditioning. Filtering, washing, temperature and humidity control with positive circulation. Remote service required.

AERIET FLOOR UNITS (Self-Contained)

Complete year round air-conditioning facilities in a single compact unit. Heating and humidifying may be omitted when not desired. Self-contained Aeriet Air Conditioning Units are water-cooled. Furnished in capacities one-half to five tons. (Melting Ice Equivalent.)

AERIET HEATING AND HUMIDIFYING UNITS

For Every Industrial, Commercial, Institutional and Domestic Requirement

Write for Catalogue

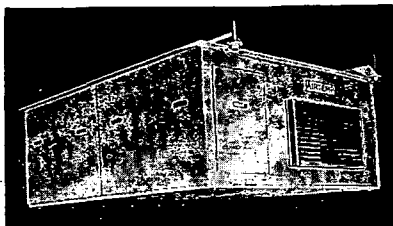
Airtemp Incorporated

8021 Conant Avenue, Detroit, Mich.

Subsidiary of
Chrysler Corporation

Complete Air Conditioning Systems for all Types of Buildings
— Restaurants — Stores — Hotels — Clubs — Theatres —
Residences — Office Buildings — Hospitals — Offices

AIRTEMP • AIR • CONDITIONERS

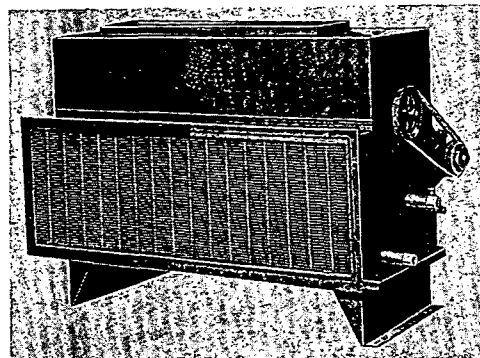


Suspended Type Air Conditioners

A suspended conditioner to be used for delivery of conditioned air directly into areas to be conditioned or for use with duct systems. Also provides for filtering and circulation of air during mild weather. Quiet in operation and highly efficient in performance.

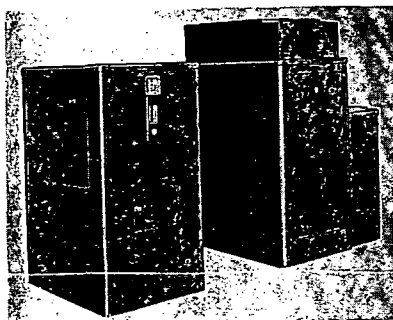
Airtemp Residential Conditioner

A concealed Year Round Comfort Unit for apartments and residences. Heats and Humidifies in winter. Cools and Dehumidifies in summer. Also provides for filtering and proper air circulation. Adaptable to Zone Type Installations. The design lends itself to installation where space is limited. Made in various capacities.



Airtemp Year Round Conditioning Systems

Airtemp Year Round Residential Conditioning System for basement installation. Designed for automatic control of summer and winter air conditions. Heating equipment can be installed and cooling system added later if desired. The boiler burner unit includes provision for domestic hot water supply.



AMERICAN RADIATOR COMPANY

40 West 40th Street, New York, N. Y.

Division of AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

AMERICAN RADIATOR CONDITIONING SYSTEMS

AMERICAN RADIATOR COMPANY announces a line of conditioning equipment for use with radiator heating. Available at present are two basement type units, one of which is suitable for homes up to 12,500 cu ft, while the other will provide for buildings up to 25,000 cu ft.

These units provide for winter humidification, air circulation, ventilation and filtration of air.

Air mains conduct conditioned air from the conditioning unit to the rooms where it is desired to introduce this air. It is not necessary to run air mains to all rooms in order to provide for adequate conditioning service in the entire house.

The units are equipped with a tempering coil which is connected to the heating boiler. This tempering coil serves to warm the conditioned air so that it can be introduced into the rooms without sensation of draft. The units are regularly furnished for steam, vapor and high temperature hot water systems. For gravity hot water systems a tempering coil of adequate capacity can be furnished on special order.

Control arrangements can be either fully automatic, semi-automatic or manual. The fully automatic control regulates the relative humidity and permits the fan and spray to operate only when there is sufficient heat in the tempering coil.

Later on the units will be available with provision for summer cooling. A cooling coil of suitable capacity will be furnished on special order, which can be used either with a direct expansion refrigerant or with chilled water.

The heating system operates entirely independently of the conditioner so that any interruption in the service of the conditioner will not affect the operation of the heating system.

Design and installation guide and complete information on these units can be obtained from branches of the American Radiator Company or by writing direct to American Radiator Company, 40 West 40th Street, New York, N. Y.

WRITE FOR COMPLETE DATA, DIMENSIONS AND RATINGS.
(See American Radiator Company pages 861-865 and Subsidiaries)

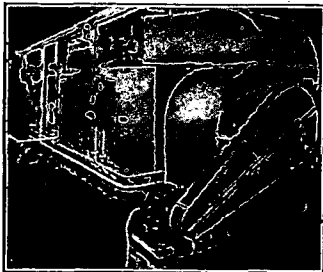
American Blower Corporation-Detroit Canadian Sirocco Co., Ltd.-Windsor

Division of American Radiator and Standard Sanitary Corp.

Branch Offices in Principal Cities of United States and Canada

SIROCCO PRODUCTS

Manufacturers of Air Conditioning, Heating, Ventilating, Drying, Dust Collecting, Dust Separation, Mechanical Draft, Pneumatic Conveying and all types of air handling equipment for more than 50 years



A typical Sirocco System of Air Conditioning for large buildings, auditoriums or factories

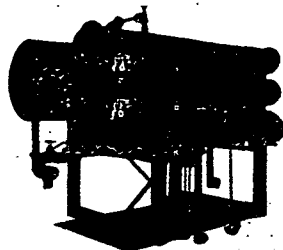
Sirocco Systems

Sirocco Systems for cooling, heating, humidifying, dehumidifying and purifying air, in all classes of business and various manufacturing processes, permit control, as desired, of temperature, humidity and air motion. Component parts: Sirocco or American HS fan and motor, dehumidifier, tempering coils, preheaters, reheaters, spray and circulating water pumps, temperature and humidity control apparatus and a "Ross" Decalorator, or mechanical refrigeration with accessories. Bulletin No. 2727.

Steam Refrigeration

(Decalorators)

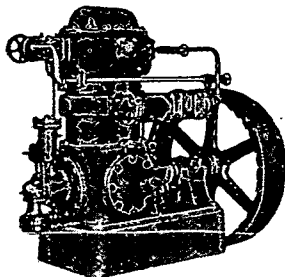
Decalorators are built and furnished in sizes varying in cooling capacity from 24,000 B.t.u. to 48,000,000 B.t.u. per hour. Chilled or decalorized water is produced by the practical application of a well known physical law, namely: water under high vacuum will vaporize at low temperatures. Decalorators will produce chilled water at a temperature of 38° F or above. The Decalorator has no moving parts and water is the only refrigerant used. Completely described in Bulletin No. 2927.



"Ross" Decalorator

Mechanical Refrigeration

For installations involving mechanical refrigeration, the motor-driven reciprocating compressor type of refrigerating machine is supplied. These compressors are available for direct expansion service, or with integral water cooling heat exchanger for use with chilled water as the cooling medium. This well constructed machine is designed for use with Freon as the refrigerant, and can be furnished in capacities from 1 to 500 tons. Complete details and specifications will be furnished upon request.

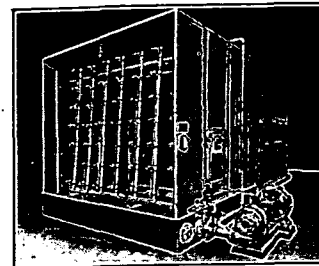


Compressor

American Blower Corporation-Detroit Canadian Sirocco Co., Ltd.-Windsor

Division of American Radiator and Standard Sanitary Corp.

Sirocco Dehumidifiers

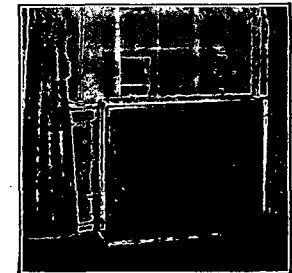


Sirocco Dehumidifiers and Washers

This line of dehumidifiers, air washers, scrubbers and purifiers, commonly referred to as the Sirocco Washer, is available in standard sizes and designs, constructed of galvanized iron casing and eliminators with welded iron tank. Capacity, 1000 to 300,000 cu. ft. of air per minute. These washers are built in lengths from 4 ft. 0 in. to 24 ft. 0 in. with one to six stages of sprays, depending upon the duty to be performed. Also built of special metal when required. Bulletin No. 3523.

Sirocco Conditioners Series "O"

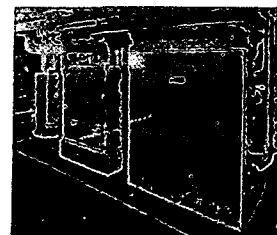
Series "O" Sirocco Conditioners are made in four sizes for comfort cooling of restaurants, shops, hospitals, offices, etc. Units are provided with three-speed switch automatically controlled by means of a thermostat, and can be equipped with wall box for outside air intake. Air filter and humidifier furnished extra. For winter heating, units are connected to hot water boiler. Complete data in Bulletin No. 2127.



Series "O" Conditioner for cooling and heating offices, restaurants and hospitals, etc.

Sirocco Conditioner Series "R"

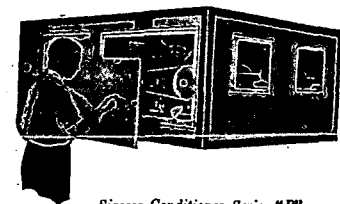
The Series "R" Conditioner is a complete conditioning unit for controlling the temperature, humidity, motion and cleanliness of air in the home. For winter heating, is directly connected to boiler (gas, oil or coal fire). For summer cooling, uses tap, ice cooled or refrigerated water. Equipped with a Sirocco electrical control for regulation of temperature and humidity. Conditioned air is distributed by means of ducts. For details refer to Bulletin No. 1127.



Series "R" Conditioner for Homes

Sirocco Conditioner Series "B"

The Sirocco Conditioner Series "B" is a unit conditioner and when used with a Decalorator (steam vacuum refrigeration) or mechanical refrigeration, constitutes a simple and complete air conditioning system for department stores, restaurants, coffee shoppes, offices and similar applications. Bulletin No. 3527.



Sirocco Conditioner Series "B"

BETZ UNIT AIR COOLER CO.

KANSAS CITY, MISSOURI

Licensed under Betz Air Conditioning System Patents

COOLS AND DEHUMIDIFIES
IN SUMMERHEATS AND HUMIDIFIES
IN WINTER

COMPLIES WITH ALL REQUISITES OF YEAR AROUND AIR CONDITIONING

Unicools cool and dehumidify in summer, heat and humidify in winter, circulate air at all seasons. The air is cleansed and purified by washing out dust, odors and toxic gases. **Unicools** provide central station performance plus individual zone control of temperature and humidity. **Unicools** are designed for installation in, or adjacent to, the space to be air conditioned. The **Unicool System** is especially adapted for multiple installations in both new and existing buildings, particularly office buildings, hotels, apartment buildings, hospitals, etc. **Unicools** are supplied in summer with refrigerated water through insulated pipe lines from a central plant. In winter, normal temperature water is circulated through the same lines for humidifying and cleansing. Steam for heating is supplied through separate lines.

Hourly Capacities and Dimensions of Unicools

No.	Hourly Btu Capacities		Normal Air Delivery	Fan H.P.	Outside Dimensions		
	Cooling	Heating			Width	Depth	Height
1	19,000	26,400	600	1/40	22"	18"	7' 6"
2	53,000	68,500	2150	1/2	22"	30"	8' 0"
3	91,000	119,000	3000	3/4	34"	30"	8' 6"
4	152,000	204,000	6000	1 1/2	46"	36"	10' 0"
5	318,000	415,000	13000	3	48"	66"	13' 0"

No. 1 **Unicools** are equipped with propeller type fans, larger sizes with squirrel cage fans. All sizes available for either right or left hand pipe connections and Nos. 2, 3, 4, 5 with vertical or horizontal air discharge.

Unicools conserve fuel in winter by providing proper humidification and uniform circulation of heated air within the occupied space. Fin type heating elements are optional.

Unicool Systems provide the advantages of a central refrigeration plant, combined with an air washer and fan, without the necessity of large expensive supply and return air ducts. In addition they make possible individual control of temperature, humidity and air movement at remote points.

With the **Unicool System** automatic zone control of temperature, humidity and air movement is definitely accomplished regardless of sun location or varying human load.

Unicools are vertical spray type units requiring relative small floor space, are constructed with double metal housings with 1" of rigid insulation interposed to prevent condensation and loss of refrigeration. **Unicools** are durably constructed to assure years of service. Openings in rear of **Unicools** for admission of outside air connections, are optional.

Unicools may secure their refrigerated water from refrigerated shell and tube coolers, from our special constructed insulated water coolers, from steam ejector plants, centrifugal water vapor refrigeration units, or from ice. **Unicools** may be used for evaporating cooling and for purification of air where natural sources of cold water are available. For moderate size installations **Unitanks** (specially designed water coolers) can be furnished in five standard sizes. For larger installations drawings and specifications of special tanks will be furnished for construction on the job.

UNICOOLS ARE COMPLETE CONDITIONERS, THEIR VALUE PROVEN BY YEARS OF FIELD SERVICE. THEY ARE EFFICIENT IN OPERATION AND A QUALITY PRODUCT

Clarage Fan Company

Kalamazoo, Michigan

Sales Engineering Offices in All Principal Cities (Consult Telephone Directory)

CLARAGE

AIR HANDLING AND CONDITIONING EQUIPMENT

For Nearly a Quarter Century Clarage has been a leading manufacturer of air handling and conditioning equipment. There is a Clarage fan, unit or system to meet every need, from the simplest ventilating or cooling job to the most exacting temperature and humidity control installation.

Whatever your ventilating, unit heating, cooling, air cleaning, humidifying, dehumidifying or complete air conditioning problem, Clarage can meet your requirements successfully and economically.

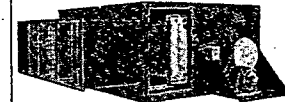
The Experience of Clarage Engineers covers almost every conceivable type of installation, commercial, industrial and public building. Clarage equipment is used in the largest industrial plants, office buildings, auditoriums, theatres, hotels, restaurants, retail stores, hospitals, churches and schools. And lately to the Clarage line has been added the Duotherm, a complete air conditioning unit for fine homes.

Clarage Vortex Control System is an important development in air conditioning. It delivers only the volume of conditioned air required by the sensible heat load existing at any given time. Compared to the conventional system, first and operating costs are definitely lower.

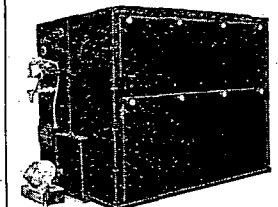
Architects, Engineers and Contractors will find our service specially helpful. This Company is an independent manufacturer, and to specify and use our apparatus does not involve any binding agreements or license fees whatsoever. Your inquiry for complete data on any Clarage product is invited.



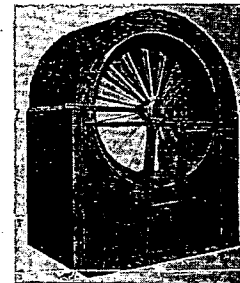
Vortex Control System for commercial and industrial conditioning



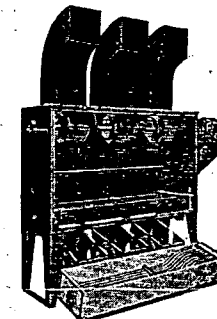
Moditherm Unit, complete conditioning plant designed for ceiling or wall installation



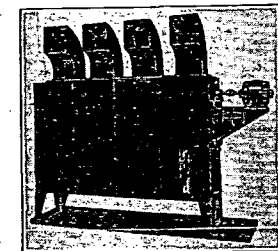
Duotherm Unit, complete conditioning plant for fine homes, installed in basement



Type HV Ventilating Fan with Clarage Vortex (constant speed) Volume Control



Uniterm Unit Heater with Synchrotherm Control as standard equipment



Uniterm Unit Cooler for product cooling and refrigeration

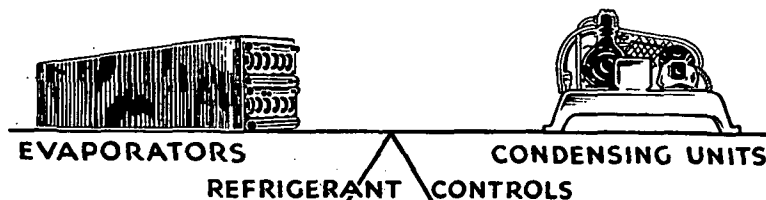
Frigidaire Corporation

Dayton, Ohio

Outlets in All Principal Cities

Air Conditioning Division

FRIGIDAIRE AIR CONDITIONING SYSTEMS ARE BALANCED



Any air conditioning system employing Frigidaire equipment—whether a Unit system or a Central Duct System—gives Balanced Performance because Frigidaire Condensing Units, Unit Conditioners, Evaporators for Duct Systems and Refrigerant Controls are designed and manufactured by Frigidaire for operation one with another.

There is a complete line of Frigidaire equipment available for providing either summer air conditioning or year 'round air conditioning for any home, office, business establishment, institution or industrial plant. There are Frigidaire unit conditioners—self-contained, remote floor-

type, suspended type and concealed type—for providing any or all functions of air conditioning. In addition there is a complete line of Frigidaire evaporators, condensing units, auxiliary coils and refrigerant controls for all types of central duct systems.

Unit Systems—Self-Contained and Remote



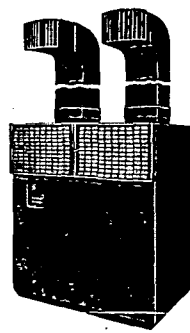
A large capacity suspended type unit conditioner

Frigidaire Unit Conditioners are of four types—the self-contained floor models, remote floor models, suspended models and concealed suspended models. Of the self-contained type there are seven models ranging from a unit of $\frac{3}{8}$ ton capacity for single room applications such as rooms in residences, offices, etc., to a 3-ton unit particularly adaptable for business establishments where it is desirable to provide air conditioning equipment that can be

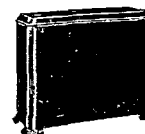
easily moved to a new location.

The line of remote floor type conditioners includes models for both summer air conditioning and year 'round air conditioning. Several units, for condition-

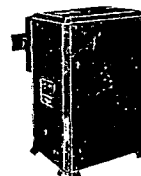
All Frigidaire Air Conditioning equipment employs FREON the safe refrigerant developed by Frigidaire, General Motors, and DuPont Engineers.



Large capacity self-contained unit conditioner



Typical floor-type remote unit conditioner for year 'round air conditioning



Typical floor-type, self-contained unit conditioner, approximately $\frac{1}{4}$ ton capacity

ing one large space or several small spaces can be operated from a single condensing unit located in the basement, spare room, closet or other convenient location. The remote floor units

are often specified when a selective system is desired, for example in a residence where air conditioning is required in several rooms but not necessarily in all rooms at the same time. Special Frigidaire refrigerant control valves installed in a system of this kind makes it possible to operate several units on a selective basis from a condensing unit of less capacity than the total capacity of the unit conditioners.

In addition to floor type conditioners, there is a complete line of suspended conditioners—particularly adaptable for applications where floor space is at a premium. These models can be suspended from the ceiling, placed above window decks or in recesses or concealed behind ornamental grilles.

A line of concealed suspended units also is available for installation in closet spaces, the conditioned air being delivered through a grill in the wall.



A typical remote floor-type unit installation, showing several conditioning units operated by a single condensing unit which is located in the basement

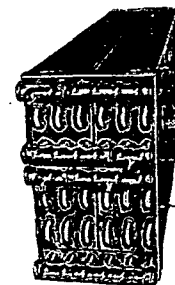
Equipment for Central Duct Systems

A complete line of evaporators, condensing units, and refrigerant controls for central duct systems, designed and balanced to operate one with another, are included in the Frigidaire line of air conditioning equipment.

5-Way Engineered Evaporators

Frigidaire evaporators for central duct systems have five design engineering features:

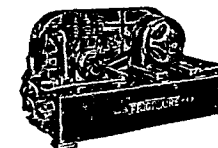
1. Low pressure drop which insures great efficiency and low operating cost.
2. Counter-flow construction to insure full dehumidifying and cooling capacity.
3. Specially constructed for air conditioning service; incorporating large surface area with minimum overall dimensions.
4. Low air resistance insuring low operating cost.
5. Matched—to operate efficiently, singly or in groups, with Frigidaire Condensing Units.



Cross section of a typical Frigidaire evaporator for duct system installations showing the manifold type of construction

Frigidaire Condensing Units

Probably no more convincing argument in favor of the dependability of Frigidaire Air Conditioning condensing units can be offered than the statement that suspended between the wheels on the big majority of air conditioned railroad cars in America, will be found a Frigidaire Condensing unit.



One of the Frigidaire 4-cylinder Water Cooled Condensing Units

There is a complete line of Frigidaire Air Conditioning Condensing units—in various capacities—both water-cooled and air-cooled.

A Frigidaire duct system installation, employing Frigidaire Condensing Units and Evaporators, in a printing establishment



Detailed specifications on the complete line of Frigidaire Air Conditioning equipment can be obtained from your local Frigidaire Air Conditioning Representative or from Frigidaire Corporation, Dayton, Ohio.

Kelvinator Corporation

MANUFACTURERS OF ELECTRIC REFRIGERATION SINCE 1914

Detroit, Mich.

CANADA: LONDON, ONTARIO

ENGLAND: LONDON

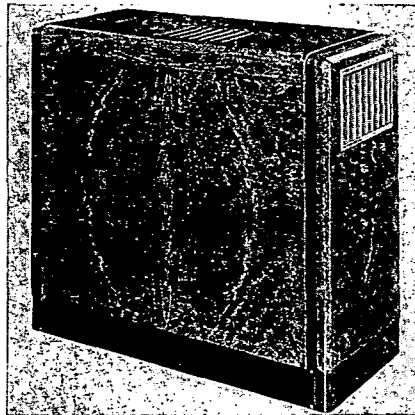
Sales and Service Facilities

In All Principal Cities Throughout the World

Manufacturers of Equipment for:

Air Conditioning, Automatic Heating, Water Cooling, Milk Cooling, Beverage Cooling, Household Refrigeration, Electric Ranges, Truck Refrigeration, Ice Cream Cabinets, Commercial Refrigeration

AIR CONDITIONING

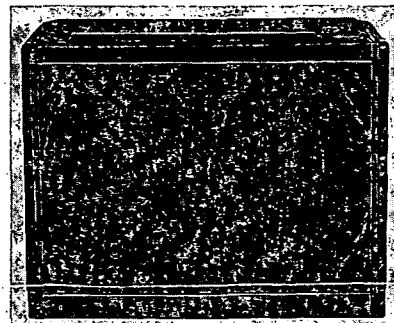


SELF-CONTAINED ROOM COOLER (WATER COOLED)

Self-Contained Units are available in several models to provide room cooling for homes and small offices. Capacities range from $\frac{1}{3}$ to 1 ton of refrigeration per 24 hours. The water cooled condensing unit, cooling and dehumidifying coils, the centrifugal fan, the multi-speed fan motor, air filter, etc., are all located within the cabinet unit. The cabinet is made of high-grade furniture steel finished in rich walnut color and grain design to harmonize with almost every type of surrounding. Top, front and rear panels are removable for inspection. Panels are lined with insulation and isolation materials to prevent the units from sweating and to eliminate all transmission of mechanical noise. All parts exposed to moisture are rust proofed.

FLOOR CABINET AIR CONDITIONING UNITS

Floor Cabinet Air Conditioning Units are available in seven models and are ideally suited to homes and offices for single or multiple installations. The Down-Draft cooling and dehumidifying coils, centrifugal fans, quiet operating multi-speed fan motor, air filters, etc., are contained in the cabinet. The cabinet is similar in finish and construction to that of the Self-Contained Room Coolers. The condensing unit and motor is located at some remote place so that the same unit can be used for several Floor Cabinet units in simultaneous or alternate operation. Heating and humidifying equipment can be added in these cabinet units.



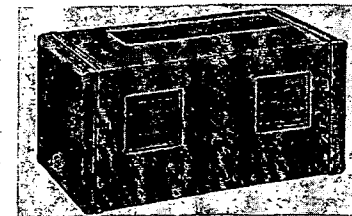
SPECIFICATIONS—FLOOR TYPE SERIES

Model* No.	Fan Data					Over All Dimensions			Net Weight Lb. Approx.
	H.P.	R.P.M. (Max.)	Speeds	C.F.M. (Max.)	No. of Fans	Width Inches	Depth Inches	Height Inches	
FT55DU	1/40	960	1	240	2	34 1/4	14 1/2	27 1/2	185
FT80DU	1/40	1140	3	275	2	34 1/4	14 1/2	31 1/2	230
FT110DU	1/20	1140	3	390	2	42 1/4	14 1/2	31 1/2	290
FT160DU	1/10	1140	3	535	2	42 1/4	17	34 1/2	330
FT50DC	1/40	1140	3	275	2	34 1/4	14 1/2	31 1/2	235
FT75DC	1/20	1140	3	390	2	42 1/4	14 1/2	31 1/2	295
FT100DC	1/10	1140	3	535	2	42 1/4	17	34 1/2	335

*Models with the suffix "DU" are controlled by a thermostat operating with a liquid line solenoid. Models with the suffix "DC" employ a humidistat as well as a thermostat to obtain a closer control of dehumidification.

Capacities range from $\frac{1}{3}$ to 1 $\frac{1}{4}$ tons dependent on temperatures of entering air, location and size of condensing unit used. Consult local dealers for exact selection.

C AND CA SUSPENDED UNITS



C Type Suspended Unit

Suspended units are available in 12 models affording a wide variety of sizes and capacities ranging from $\frac{1}{2}$ to 10 tons of refrigeration. The Down-Draft cooling and dehumidifying coils, air filters, and quiet operating centrifugal fans and motors are inclosed in a sturdy sound absorbing cabinet which is furnished with a prime coat only to permit a finishing coat to match individual surroundings. Heating and humidifying equipment can be added within the units.

C AND CA SUSPENDED UNITS—SPECIFICATIONS

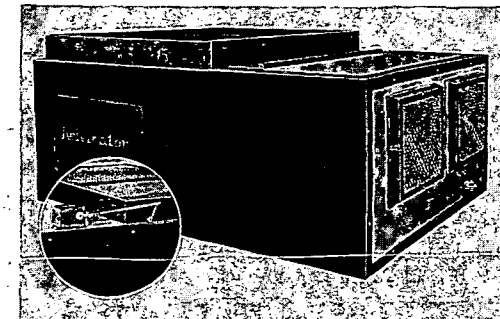
Model* No.	Fan Data			Overall Dimensions (Inches)			Approx. Net Weight Lb.
	C.F.M.	R.P.M. (Max.)	Motor H.P.	Width	Depth	Height	
C70DU	265	1140	1/40	30 1/8	17 1/8	15 1/8	175
C125DU	378	1140	1/20	31 1/8	22	19 1/8	215
C200DU	663	1120	1/10	46 1/4	22	19 1/8	375
C375DU	1430	1070	1/4	46 1/4	33	22 1/8	450
CA400DU	1200	825	1/3	46 1/4	52 1/2	26 1/4	590
CA550DU	1820	825	1/2	46 1/4	53 1/4	26 1/4	620
CA600DU	1820	550	1/2	50 1/4	65 1/4	30 1/4	820
CA900DU	2800	660	1/2	50 1/4	65 1/4	30 1/4	835
CA275DC	1200	825	1/3	46 1/4	52 1/2	26 1/4	610
CA400DC	1820	825	1/2	46 1/4	53 1/4	26 1/4	620
CA500DC	1820	550	1/2	50 1/4	65 1/4	30 1/4	820
CA600DC	2800	660	1/2	50 1/4	65 1/4	30 1/4	835

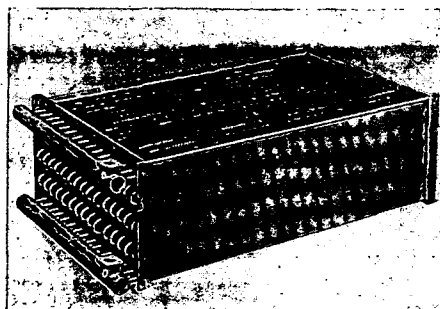
*Models with the suffix "DU" are controlled by a thermostat operating with a liquid line solenoid. Models with the suffix "DC" employ a humidistat as well as a thermostat to obtain a closer control of dehumidification.

Refrigeration capacity ranges from $\frac{1}{2}$ to 10 tons, dependent on temperature of entering air, location and size of condensing units used. Consult local dealer for Exact Selection.

CS CENTRAL STATION UNITS

Central Station units are available in several models for capacities greater than those of the Suspended units. These units are sectionalized to permit a wide range of air filter, cooling and dehumidifying, heating and humidifying, and air distribution capacities.





Down-Draft Duct Coil

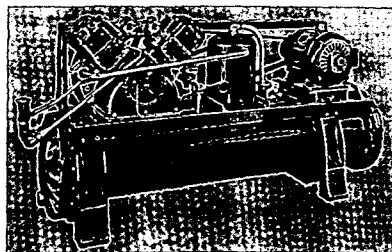
DS DOWN-DRAFT DUCT COILS

A complete line of Down-Draft Coils are available for installation in existing ventilating and air washing systems to convert such systems to summer air conditioning and cooling. These coils are also applicable to new central station plants. Coils are offered in nominal tube lengths of 20 in., 29 in., 36 in., and 48 in. with capacities per coil ranging from 1 to 10 tons, and multiple arrangements of coils for any desired capacity. The down-draft feature of this coil insures immediate drainage of the condensate formed in dehumidification, and uniform distribution of refrigerant and maximum capacity of the coil.

CONDENSING UNITS

Kelvinator offers a complete line of air cooled and water cooled Condensing units, ranging in the air-cooled series from 1/4 H. P. to 5 H. P., and in the water-cooled units from 1/4 H. P. to 20 H. P.

In its more than 21 years of refrigeration engineering and research, Kelvinator has developed condensing unit design and construction which is of highest quality, in conformity with the best engineering practice, and each unit has been specially devised for the particular requirements of its job.

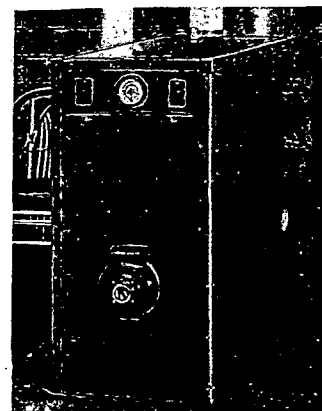
**CONDENSING UNITS—SPECIFICATIONS—AIR COOLED MODELS**

Model No.	H.P.	No. of Cycles	Comp. R.P.M.	Refrigeration Capacity Tons per 24 Hours*	Overall Dimensions (Inches)			Approx. Net Weight Pounds
					Width	Depth	Height	
B625	1/4	2	415	0.24	25	17 1/8	18 3/8	148
G633	1/2	2	315	0.32	25 1/2	17 3/8	19 1/8	183
H650	1/2	2	340	0.46	28	19 1/8	19 3/8	215
F675	3/4	2	330	0.70	30 3/4	23 3/8	21 3/8	366
K6100	1	2	310	0.96	35 3/8	24	23 1/4	408
R6150	1 1/2	2	310	1.33	40	26 1/8	26 3/8	537
Y6200	2	2	280	1.92	40	28 3/8	27 3/8	617
T6300	3	2	280	3.08	44	27 3/8	29 3/8	750
T6500	5	2	375	4.12	44	29 3/8	29 3/8	850

WATER COOLED MODELS

Model No.	H.P.	No. of Cycles	Comp. R.P.M.	Refrigeration Capacity Tons per 24 Hours*	Width	Depth	Height	Approx. Net Weight Pounds
WB625	1/4	2	465	0.30	25 3/8	14 1/8	18 3/8	169
WG633	1/2	2	365	0.40	25 3/8	14 1/8	19 3/8	195
WH650	1/2	2	410	0.59	29	15 3/8	19 3/8	200
WF675	3/4	2	400	0.85	30 3/4	19 1/4	21 3/8	380
WK6100	1	2	400	1.36	35	19 1/2	22 3/8	413
WR6150	1 1/2	2	400	1.88	40 3/8	22 3/8	26 3/8	511
WY6200	2	2	375	2.55	40 3/8	23 3/8	27 3/8	605
WT6300	3	2	300	3.27	44 3/8	23 3/8	29 3/8	835
WT6500	5	2	440	4.80	55 3/8	23 3/8	29 3/8	1200
WU6750	7 1/2	2	440	7.00	72 3/8	25 3/8	34 1/2	1960
WU61000	10	2	650	9.95	72 3/8	25 3/8	43 3/8	2035
WV61500	15	4	440	14.00	92 1/8	28 3/8	43 3/8	3095
WV62000	20	4	650	19.90	93 3/4	31	46	2185

*Approximate one point rating capacity based on 80 F. Condenser Water Temperature or 90 F. Condenser Air with 37 lb Average Suction Pressure. Consult Dealers for Exact Selection.

KELVINATOR AUTOMATIC HEATING**BOILER BURNER UNITS (STEEL)**

Boiler Burner (Steel) Model No. KB-1

Kelvinator Boiler-Burner Unit offers the heating engineer a compact unit built of welded steel and thoroughly insulated. Exterior metal jacket is rust-proofed, finished with light gray lacquer and trimmed in polished chromium.

Outstanding boiler features are cylindrical combustion chamber, single pass tube bank, spiral baffles and thorough insulation.

BOILER BURNER UNITS (CAST IRON)

Exceptionally compact and attractive in appearance, these units provide an unusually efficient sectional cast iron, oil burning boiler, a domestic water heater and oil burner assembled as one unit inclosed in an attractive bonderized steel jacket trimmed with chromium. Moulded combustion chambers, conforming to the natural size and shape of the flame, quickly develop the maximum amount of heat contained in the fuel. The sectional design permits easy installation without altering stairways or doors.

BOILER BURNER UNITS—110 V.—60 CYC.

Model No.	Fuel Oil Capacity per Hour	Square Feet Radiation	
		Steam	Hot Water
KB1 (Steel)	1.22 Gal.	565	850
KB2 (Steel)	2.00 Gal.	925	1400
KB3 (Cast Iron)	1.66 Gal.	780	1170
KB4 (Cast Iron)	2.6 Gal.	1230	1845
KB5 (Cast Iron)	3.5 Gal.	1635	2450

Steel units have constant electric ignition and line voltage controls. Cast Iron units have intermittent electric ignition, low voltage A. C. controls for 115 V. D. C., line voltage 115 V. D. C. controls.

CONVERSION TYPE UNITS

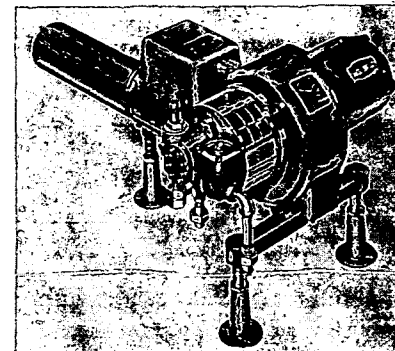
Model No.	Fuel Oil Capacity Gallons per Hour		Square Feet Steam Radiation	
	110 V., 60 Cyc., 115 V. D. C.	110 V., 25 or 50 Cyc.	110 V., 60 Cyc., 115 V. D. C.	110 V., 25 Cyc. or 50 Cyc.
K-1	2.5	2.00	750	550
K-1A	2.0	1.75	600	500
K-2	4.5	4.25	1300	1200
K-3	7.5	6.50	2200	1900

Intermittent electric ignition low voltage A. C. controls, line voltage D. C. controls.

All Units Have: Split Phase Motor; Pressure Fuel Atomization; and require A.O.B.A. No. 120 D3 Fuel Oil.

CONVERSION TYPE UNIT—**MODEL No. K-1**

Kelvinator Conversion Type Pressure Burner Units for conversion of existing heating plants feature advanced design for long life, economy and trouble-free heating service. Features are: fuel control unit, noiseless, self - priming, non - pulsating pump, adjustable air shutter, scientific fan and housing construction for noiseless operation, directly connected motor with overload protection, air turbulator for complete combustion. These and other features indicate the thoroughness with which Kelvinator adheres to the best engineering practice.

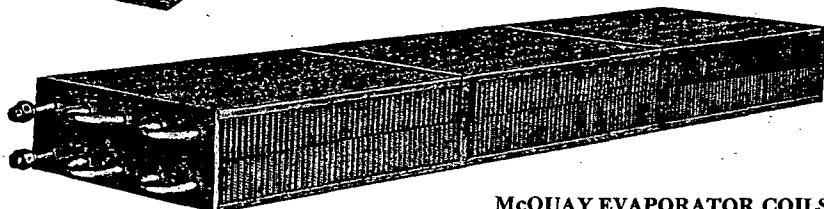
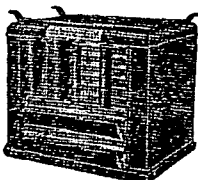
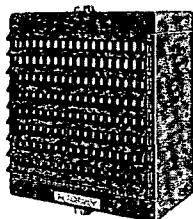
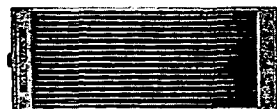
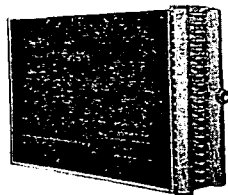


McQuay, Incorporated

1600 Broadway, N.E., Minneapolis, Minn.

UNIT HEATERS — BLAST HEATERS — CONVECTION RADIATION
AIR CONDITIONING COILS — EVAPORATOR COILS — UNIT COOLERS

AIR CONDITIONING—BLAST COILS



McQuay Extended Surface Coils for Air Conditioning, Central Fan Heating, and for all commercial drying purposes are the result of our many years of experience in the manufacture of heat transfer surfaces. Constructed entirely of non-corrodible materials they are durable, light in weight, and compact, affording quick economical installation. Specially designed fins are attached to the round seamless tubes under hydraulic expansion, forming a permanent frictional contact between tube and fin, without the use of solder, unaffected by constant expansion and contraction. Tubes are properly orificed permitting a uniform distribution of steam or water into each tube. Tubes are brazed, by special process to round, heavy gauge, drawn copper tube headers, insuring a lasting and trouble-free heat transfer surface. The completed unit is encased in a heavy gauge galvanized housing with provisions for simple and economical duct connections. Since most applications of coils for Air Conditioning and process drying are of a special nature, we offer the assistance of our Engineering Department in solving your various problems.

McQUAY UNIT HEATERS

McQuay Unit Heaters are scientifically designed, engineered, and employ a heating element of the same design as the Air Conditioning and Blast Heater coils. McQuay Unit Heaters are available in twenty sizes ranging from 80 to 1500 sq. ft. of direct radiation making it convenient to select the proper size heater for any heating requirement.

McQUAY UNIT COOLERS

Available in seven sizes for Walk-In Coolers, Beer Storage Rooms, Truck Refrigeration, etc. For use with all types of refrigeration machines using Methyl Chloride, Freon, Sulphur Dioxide, and other similar refrigerants. Units for Ammonia also available.

McQUAY COMFORT COOLERS

Available for Water or Brine and Direct Expansion refrigerants. Numerous sizes with capacities to fit your requirements. Floor and suspended blower type units also available.

McQUAY EVAPORATOR COILS

McQuay coils are designed for practically all phases of commercial refrigeration. Fins made with a special spun collar hydraulically attached to tinned copper tubes, forming a permanent frictional contact between tube and fin, without the use of solder, and unaffected by constant expansion and contraction. All joints are brazed, assuring against leaks ordinarily occasioned by soldered joints. McQuay method of manufacturing makes possible a flexible line of special coils tailor-made to fit your specific requirements. McQuay coils for ammonia installations are also available for all purposes.

Bulletins descriptive of all McQuay products available upon request

Parks-Cramer Company

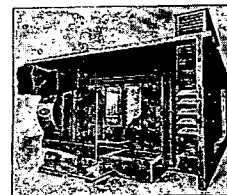
Fitchburg, Mass.

Charlotte, N. C.

Complete Air Conditioning Systems including Heating,
Cooling, Humidifying, De-humidifying, Ventilating,
Refrigeration, Air Filtering and Air Washing

CERTIFIED CLIMATE

Certified Climate—Central Station Air Conditioning



Central Station

For health and comfort in Hospitals, Art Galleries, Auditoriums, Restaurants, Stores and Homes. Centrally located apparatus supplies adequate moisture with positive pre-determined air change through air washers, heaters—or refrigeration for cooling. Automatic Regulation essential.

Automatic Aircharger (Not Illustrated)

Balanced evaporation with air change. Insures maximum evaporative cooling in hot rooms in summer. Conserves heat in winter. Positive air circulation and uniformity of humidity and temperature. Mostly for industrial application.

Unit Air Conditioners (Not Illustrated)

Various types for industrial or non-industrial use. Similar in purpose to Central Station, but more portable.

Industrial Air Conditioning

Helps in many industries, notably, Textiles (Cotton, Wool, Worsted, Silk, Rayon, Jute); Printing and Lithographing; Cigar, Cigarette and Tobacco; Clothing; Paper and Envelope, Leather and Shoes, Wood Products; Cereals, Storage of Perishables; Ceramics; Celluloid; Glassine Paper; Starch and Dextrine; Cement.



Psychrostat

Automatic Regulation

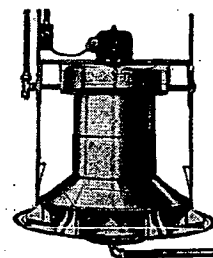
The Psychrostat for accuracy, durability, sensitivity; Hygrostat (not illustrated) where requirements are not so exacting. Psychrostat uses the Principle of the Sling Psychrometer. U. S. Government uses Sling Psychrometer in all Weather Bureau Stations. An Air Conditioning System is no better than its Regulation.

Turbo Atomizer Humidifier (Not Illustrated)

Spray generated by air under pressure. Water not under pressure and consequently no chance of leakage. Several fixed capacities. Highest efficiency of any atomizer. Popular in a wide variety of industrial plants.

High Duty Humidifier

Water under pressure generates spray. Excess water returned to filter tank and recirculated. Evaporation per unit high; two sizes of heads each with two sizes of nozzles give flexible capacity for varying conditions. Circulation increased by individual motor-driven fan. Spray thoroughly diffused and distributed over wide area.



High Duty Humidifier

Niagara Blower Company

AIR ENGINEERING EQUIPMENT AND SYSTEMS

General Sales Office: 6 East 45th Street, New York City

BUFFALO BOSTON PHILADELPHIA CLEVELAND PITTSBURGH DETROIT CHICAGO ST. LOUIS SAN FRANCISCO

PRODUCTS—Exact Control Air Conditioning, Humidifying, Dehumidifying, Drying, Moistening, Chilling, Comfort Systems, Niagara Air Conditioners, Niagara High Humidity Spray Cooler, Niagara Fan Coolers, Niagara Fan Heaters, Niagara Aluminum Cooling Coils, Heating Coils.

NIAGARA AIR CONDITIONING SYSTEMS

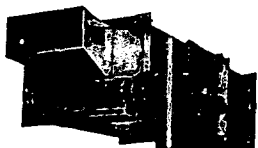
For human comfort and for all industrial applications requiring controlled climatic conditions of temperature, relative humidity, air purity and air movement.

NIAGARA AIR CONDITIONER, TYPE A—STANDARD ALL ALUMINUM

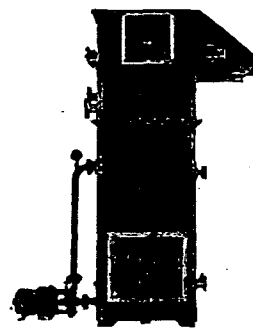
Maintains constantly, or makes any change required in temperature and relative humidity; dries or moistens within tolerance of 1 degree F. and 2 per cent R.H. in processing hygroscopic materials; cleans air most effectively; secures saturation for dehumidifying. Seven sizes. Available both in floor mounted and space saving suspended types.

NIAGARA AIR CONDITIONER TYPE C

Uses Surface Cooling Coil method for cooling and dehumidifying. Manufactured in sections so that any desired combination of air conditioning functions may be obtained; also solves installation problem in existing buildings as its sections can be brought thru any doors and erected in confined spaces.



*Niagara Type C Air Conditioner
Niagara All Aluminum Surface
Coil Air Conditioner, manufactured
in 7 sizes, both floor mounted and
ceiling suspended models*



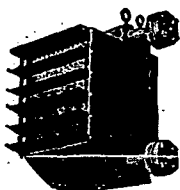
*Niagara Air Conditioner, Type A.
manufactured in 7 sizes, both floor
mounted and ceiling suspended models*

NIAGARA SURFACE COOLING COIL METHOD

Air conditioning for human efficiency and comfort. A year around operating system providing winter heating and humidifying and summer cooling and dehumidifying for offices, stores, restaurants and all places where people gather. An advanced engineering method introducing simplified apparatus and controls, for superior operating results, economy and long life, which cut operating costs.

NIAGARA FAN COOLER

Recommended for comfort cooling, process cooling, low temperature storage for dairies, fruits, meats, food products; fur storage vaults, etc. Gives complete circulation of air at desired temperature with even temperature at all points. Manufactured in seven sizes.



*Niagara Disk Fan Cooler
Manufactured in 7 sizes*



*Niagara Fan Cooler
Manufactured in 1-, 2-, 3-, and 4-fan units
and in 7 sizes—both floor mounted and
ceiling suspended types*

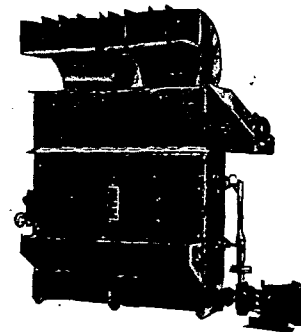
NIAGARA DISK FAN COOLER

For overhead suspension, saves space and provides the efficiency of moving air cooling for small storage areas, market coolers, etc. Seven sizes.

NIAGARA SPRAY COOLER—STANDARD ALL ALUMINUM

For all cooling applications requiring high humidity or high capacity in small space such as food product applications, especially meat chilling and storage pre-cooling and storage of fruits and vegetables. Prevents drying out, wilting. Also used in industries requiring freezing temperatures such as ice-cream manufacture and frozen food storage.

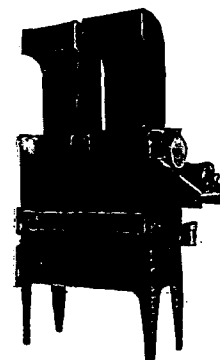
Niagara Spray Cooler is built with cooling coils in a constant brine or water spray. Maintains constant relative humidity as required. Available both in floor mounted and space saving suspended types.



*Niagara Spray Cooler
Illustration of 8-fan unit. Also manufactured
in 1-, 3-, and 4-fan units and in 7 sizes, both
floor mounted and ceiling suspended models*

NIAGARA FAN HEATERS

For the heating and ventilating of large areas, Niagara Fan Heaters put the heat immediately where needed in the working zone, give quicker heating up to working temperatures. Definitely built to the highest possible standards; welded aluminum heating coils.



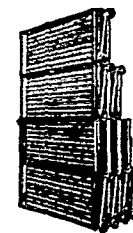
*Niagara Fan Heater
Illustration of 8-fan unit. Also
manufactured in 1-, 3-, and 4-fan
units and in 7 sizes, both floor
mounted and ceiling suspended
models*

NIAGARA HUMID HEATER—STANDARD ALL ALUMINUM

Recommended for industrial applications where heat and humidity are required as in manufacture of textiles, cordage, printing and paper-converting plants.

NIAGARA ALUMINUM COOLING COILS

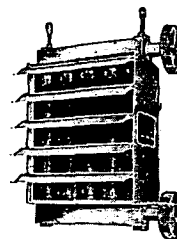
High pressure tested cooling coils are used in Niagara Fan Coolers and Niagara Surface Method air conditioning. Encased, seven standard sizes for blast cooling installations. 20-in. and 30-in. widths. Aluminum coils in aluminum cases.



*Niagara Aluminum
Heating Coils*

NIAGARA ALUMINUM HEATING COILS

For use with fan heating systems giving the advantage of aluminum, light weight and resistance to corrosion. Manufactured in two widths, 20-in. and 30-in., and in various lengths, giving a complete range of sizes. 150 lbs. working steam pressure.

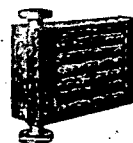


*Niagara All Aluminum
Disk Fan Heater*

Niagara Aluminum Booster Heaters are used for reheaters to control room temperature independently of fan system.

NIAGARA DISK FAN HEATERS

A most effective suspended heater. Operates with lower discharge temperature, cuts down roof and wall losses.



*Niagara Aluminum
Booster Heater
Write for Bulletin
No. 20*

Savage Arms Corporation

AIR CONDITIONING DIVISION

100 East 42nd, New York, N. Y.

ZEPHYR AIR

ZEPHYR AIR equipment includes a complete line of air washing-humidifying-cooling units for all-year service. Self-contained and remote refrigeration; direct expansion or chilled water. Capacities range from 175 to 4000 cfm and from $\frac{1}{2}$ to 10 tons refrigeration.

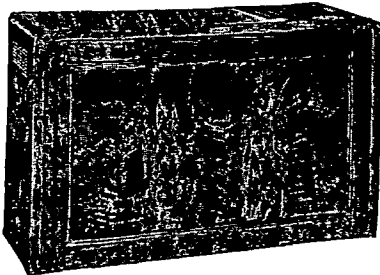
The models illustrated are typical of the line. Some are specially designed for multiple installation in office buildings, with chilled water circulated to units from central refrigeration. The special deep counterflow coil is used, requiring small water flow and minimum pipe sizes.

SALIENT ZEPHYR AIR FEATURES

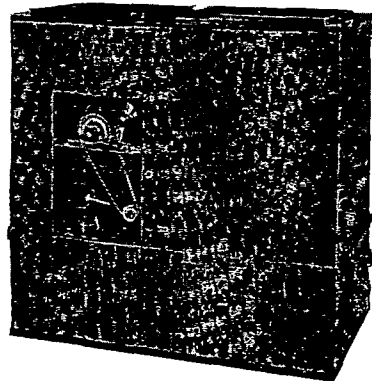
1. All-year air washing for effective removal of dust and soluble odors. In Summer, the air is washed before it is cooled, avoiding accumulation of dirt and odors on moist surface of cooling coil. In Winter the air washing provides air cleaning and adequate humidification.
2. Option of finned tube evaporator for direct expansion or special deep counterflow coil for chilled water.
3. Unusually high evaporating capacity for humidification because of dense, finely atomized spray.
4. Extreme simplicity and accessibility.
5. Outstanding for quiet operation, trouble-free performance and compactness.

ZEPHYR engineers have pioneered the field of unit air conditioning since 1926. The soundness of the patented ZEPHYR atomizing method and other distinctive features has been proved in thousands of highly satisfactory installations. Now ZEPHYR AIR equipment is produced and backed by Savage with all the advantages of Savage resources and precision manufacturing facilities.

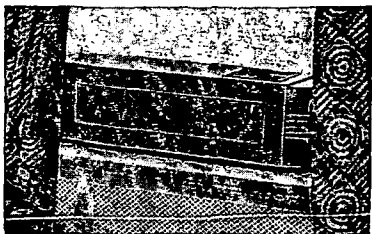
Complete specification folders on request



ZEPHYR 60—Self-Contained All-Year Model. Capacity to cool one room; to humidify 8 room house



ZEPHYR 85-86-88-90 Series—Larger Capacity Models, 600 to 4000 cfm, for cabinet or duct installations



ZEPHYR 40—Window Ventilator—Air Washer. Will evaporate from 8 to 12 gal water daily

Servel, Inc.

Commercial Refrigeration Division

Evansville, Indiana

AIR CONDITIONING

Servel Air Conditioning Units (Floor Type)

Handsome units with wood grain exterior, suitable for homes, offices, small stores, shops, etc. While these units are designed primarily for Summer conditioning (Cooling, dehumidifying, circulating and cleaning), they can be equipped with a heating coil and a spray-type humidifier for Winter use.

Servel Air Conditioning Units (Self-Contained)

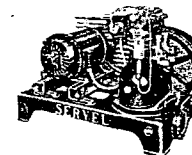
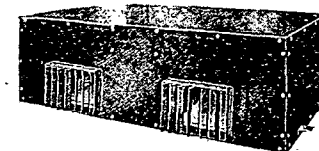
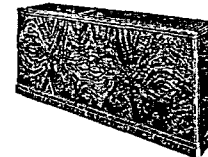
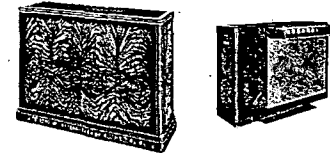
Completely self-contained, semi-portable Summer conditioning units can be installed anywhere that an electrical connection, a water line, and a drain are available. Ideal for homes, offices, and apartments where it is not practical to place the refrigerating unit at a remote point.

Servel Air Conditioning Units (Suspended-Type)

These suspended-type units are designed for restaurants, stores, beauty shops, offices, etc., where limited space precludes the use of a floor-type unit. (Can be used in a ducted system or suspended from the ceiling). Capacities from 1 to 4 tons.

Servel Refrigerating Machine Units

Offered in a complete range of sizes and models for every air conditioning need and for every operating condition, from $\frac{1}{2}$ ton to 20 tons. Available with either of two most popular refrigerants. Many exclusive features insure efficiency, economy, and durability.



This modern 30-acre plant is the home of Servel Air Conditioning

DESCRIPTIVE FOLDERS AND ENGINEERING DATA WILL BE SENT ON REQUEST

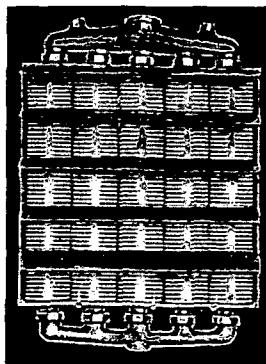
Thermal Units Manufacturing Company

64 East 25th Street, Chicago, Ill.

Representatives in All Principal Cities

PRODUCTS—Heaters, Unit; Coolers, Unit; Humidifying Units; Coils, Heating and cooling, Fans and automatic refrigerating compressors for commercial and air conditioning work

THERMAL UNIT HEATERS—Six Sizes



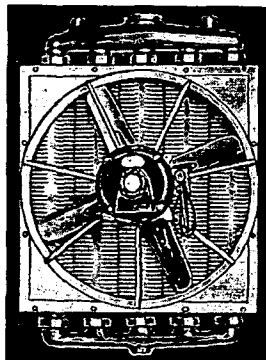
With permanent one piece—integrally cast—aluminum element. Capacities from 15,000 to 500,000 Btu per unit.

No joints, welds, Brazed or soldered connections.

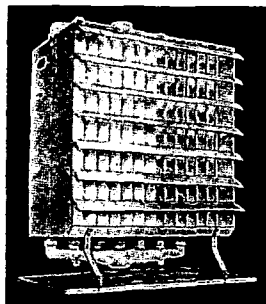
Leakproof, Freezeproof, indefinite life without servicing.

Maximum air delivery per horsepower, with lowest outlet air temperature.

Built for troubleproof service and long life.



THERMAL UNIT COOLERS—Five Sizes

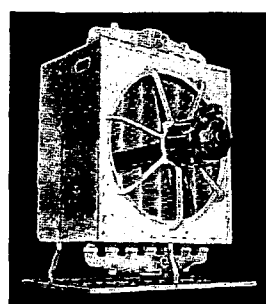


For Ammonia, Brine, Freon, water and other refrigerants. Three types: Circulating, Flooded or direct Expansion. Capacities from 1/2 ton to 8 tons refrigerating effect.

Employs the same sturdy heat transfer element as our Unit Heater above.

Automatic operation, Automatic defrosting, with controls.

Eight years of successful applications, of all types.



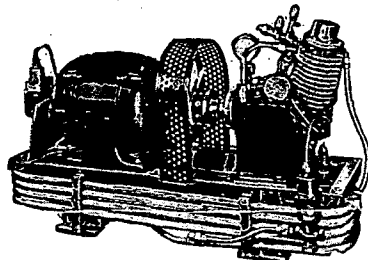
THERMAL UNIT "V8" AUTOMATIC REFRIGERATING COMPRESSORS

Simplicity - Efficiency - Economy—Using Freon or Methyl-Chloride
High Volumetric Efficiency - Low Cost - Trouble-Free Operation

Eight cylinders and pistons for smooth operation.

Silent operation for air conditioning.

Light weight, more compact for delivered refrigeration tonnage.



Direct Motor drive, vibrationless operation.

Pressure lubrication, for longer life and fewer wearing parts.

Economical operation with increased efficiency.

Catalogs, information, and Engineering data furnished on request

Westinghouse Electric & Manufacturing Co.

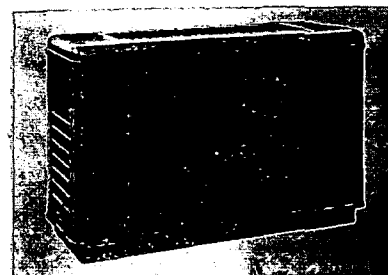
Mansfield, Ohio

Sales, engineering and service facilities available through local distributors in principal cities

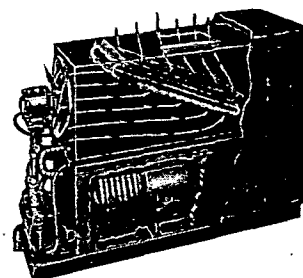


Westinghouse Air Conditioning Equipment is built *exclusively* for air conditioning service. Developed by Westinghouse Research and Design Engineers, it offers many advantages over equipment originally intended for other purposes.

Westinghouse equipment is available for either summer or year 'round air conditioning. It has been especially designed to fit into minimum space, simplifying its installation in existing buildings as well as new structures.

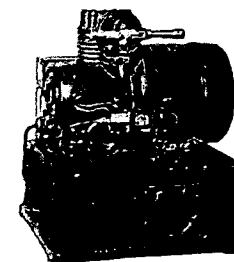


Westinghouse type EL Air Conditioning Unit for stores, offices, homes, etc. When connected to a suitable condensing unit and steam or hot water heating system, it provides automatic and complete year 'round air conditioning.

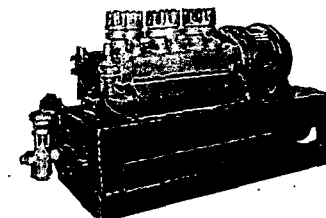


The Mobilair is a self-contained summer air conditioning unit, providing cooling, dehumidification and circulation of

air. Powered with the famous Westinghouse Hermetically-sealed mechanism, it has a capacity of 7,000 Btu per hour.



Type RW-1 Condensing Unit is especially compact in design. Capacity 12,000 Btu per hour. Motor and compressor are built in a single unit, the water cooled condenser is located in the base.



Type RW-705 Condensing Unit has a capacity of 240,000 Btu per hour. Motor mounted on compressor shaft eliminates belts, flexible couplings and other mechanical drive devices, providing extremely quiet operation and long, trouble-free service. The condenser is located in the bed plate under the compressor and motor. These condensing units are free from vibration, permitting installations to be made without expensive foundations.

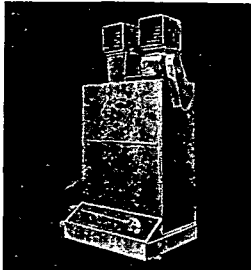
Units of other sizes are available to meet specific requirements.

York Ice Machinery Corporation

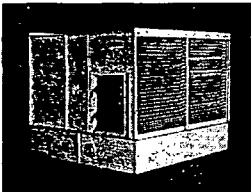
General Offices: York, Pennsylvania

Direct Factory Branches in 71 U. S. Cities

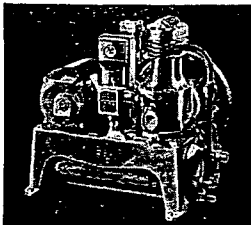
Complete Air Conditioning and Refrigerating Systems for maintaining proper atmospheric conditions for industrial processes and human comfort. Available in central and unit systems . . . from fractional tonnage up to any capacity required.



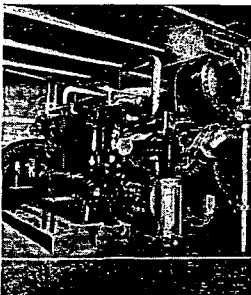
York Economizer



York Standard Dehumidifiers



York Freon Refrigerating Unit



York Water Cooling System

York Economizer:

A complete self-contained, combined forced-draft cooling tower and refrigerant condenser. Can be installed outdoors, or in ventilated space using discharge duct to outside for rejection of condenser air. Water circulating pump mounted on casing. No condensing water circulating pump or piping required. York Economizers practically eliminate usual water requirements for condensing and power for pumping water. Power requirements of compressor motor are likewise reduced for seasonal operation of refrigeration or air-conditioning plants.

York Industrial Spray Type Air Conditioner:

(Not Illustrated)

For meat chill rooms, fruit cooling and storage rooms, textile and printing plants, bakery fermentation rooms and other spaces requiring a constant temperature and relative humidity. In low temperature field, where brine spray keeps refrigerant coils defrosted, corrosive problems are particularly acute. In this York unit long life is insured by welded tank steel casings hot dipped galvanized, stainless steel eliminators, fan wheels and scrolls . . . cadmium plated fan shaft.

York Standard Dehumidifiers:

Designed especially for air-conditioning applications employing refrigeration. Suitable for theatres, public buildings, stores and office buildings as part of complete central station systems with controls, fans, motors and air distributing ducts. Available in wide range of standard sizes and capacities; also in special sizes and designs for any commercial or industrial air-conditioning requirement.

York Refrigerating Systems for Air Conditioning:

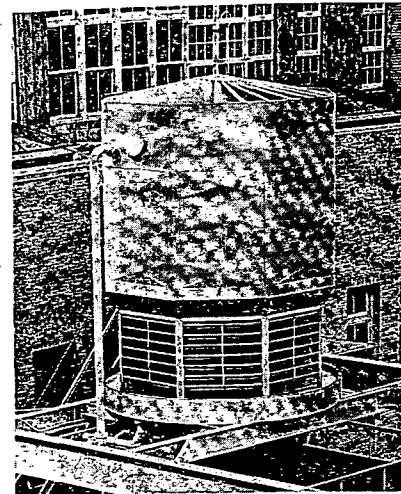
Complete refrigerating systems for use with Freon (F 12) or Ammonia. Because refrigeration for air conditioning is essentially a water cooling problem, York has developed properly balanced, standard water cooling systems for this duty. Designed especially for human comfort applications, Freon was selected as the most suitable refrigerant because of its outstanding characteristics . . . odorless, non-toxic and non-poisonous, non-inflammable and non-explosive, non-irritant and non-corrosive. Freon's thermal properties make it ideally suited for use in vertical single acting reciprocating compressors.

York Engineering Service:

Headquarters for mechanical cooling since 1885 York maintains, on instant call, expert engineers in important centers of demand. York gives you this important service and invites Architects, Engineers and Contractors to avail themselves of it.

Research Corporation

Chrysler Building, 405 Lexington Avenue, New York City



COEY MULTI-STAGE COOLING TOWER

Patented

"Water Saved is Money Made"

Gives: Controlled Cooling, Minimum Noise Level, Sprayless Operation, Architectural Harmony.

Because of its compactness, light weight, spray-free and noiseless operation, the Coey Multi-stage Cooling Tower is especially well suited to roof and basement installation in congested districts; although the highly efficient cooling provided by the multi-stage principle makes this unit desirable for installation anywhere.

The tower is built of a copper bearing steel shell, with cypress water baffles and a non-overloading reverse blade centrifugal fan rotor.

Air Movement—Non-overloading centrifugal rotor, high efficiency, variable speed.

Principle of Operation—Three pass, counter current flow.

No ice formation on fan rotor during winter operation.

No.	G.P.M.	Shape	Short Dia.	Height	Net Weight	Tons Capacity, Gas Refrig.
4	25	Hexagonal	3 ft-1 1/2 in.	6 ft-4 in.	1000 lb	10
10	80	Hexagonal	5 ft-1 1/2 in.	9 ft-2 in.	2200 lb	25
26	200	Hexagonal	8 ft-2 1/2 in.	13 ft-2 in.	6800 lb	70
50	400	Hexagonal	11 ft-1 1/2 in.	17 ft-9 in.	14500 lb	150
85	700	Hexagonal	14 ft-4 in.	21 ft-8 in.	21000 lb	225
110	900	Octagonal	16 ft-5 1/2 in.	24 ft-2 in.	28000 lb	300
150	1200	Octagonal	19 ft-0 in.	27 ft-8 in.	38000 lb	400

THE CALORIDER SYSTEM OF HUMIDITY CONTROL

Removes excess moisture by chemical absorption, with an inexpensive compound known as "Caloride." The system has two distinct functions (1) the control of humidity and air purity for industrial applications, including the drying and purification of various gases and (2) the control of effective temperature and air purity for human comfort and health.

Cottrell Electrical Precipitation Systems—Multiclone Dust Collectors

Inertia Impact Classifiers

American Moistening Company

ESTABLISHED 1888

ATLANTA, GA.
BOSTON, MASS.

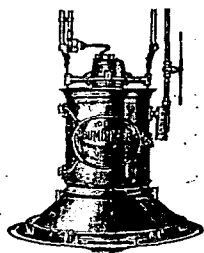
Providence, R. I.

CHARLOTTE, N. C.
GREENVILLE, S. C.

AMCO

UNIT HUMIDIFYING AND AIR CONDITIONING EQUIPMENT

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.

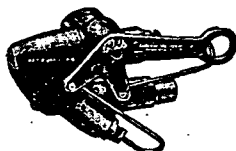


IDEAL HUMIDIFIERS—Senior Type

A high capacity unit for use where conditions require a great amount and good distribution of moisture. Motor driven fan gives wide distribution of atomized spray. Amco heads serve the triple purpose of humidifying, air washing and cooling.

IDEAL HUMIDIFIERS—Junior Type

Similar in construction to Senior Type. Used where medium capacities are required.



AMCO ATOMIZER—Self-Cleaning

Provided with lever cleaning device—an exclusive AMCO development. Atomized spray is produced by means of compressed air. There is no possibility of water leakage at the nozzle at any time—a distinct advantage of AMCO atomizers. Made in various capacities to meet any required conditions.



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.

A few of many AMCO products with a Long Record of Dependable Performance

Sectional Humidifiers.
Antex Humidifiers.
Hand Sprayers.
Mine Sprays.
Fabric and Paper Dampeners.

Mechanical Psychrometers.
Electro Psychrometers.
Sling Psychrometers.
Hygrometers.

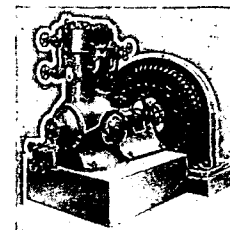
Baker Ice Machine Co., Inc.

1523 Evans Street, Omaha, Nebr.

MANUFACTURERS OF INDUSTRIAL AND COMMERCIAL REFRIGERATION
AND AIR CONDITIONING

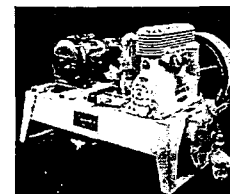
GET FULL SPECIFICATIONS ON BAKER'S COMPLETE LINE

Baker pledges full factory co-operation on problems of engineering and installation. Plants and sales offices are situated to give prompt, effective service and information regarding the complete Baker line of compressors, units, and condensers of all types and capacities. Detailed specifications for your files sent upon request.



Baker Ammonia Compressor

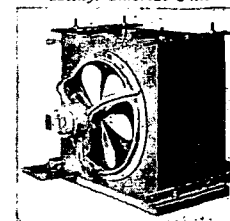
Baker Ammonia Compressors available in units to 100 tons capacity. Vertical, enclosed, single-acting type, arranged in duplex or multiple for any desired capacity. Synchronous, direct-connected or V-belt drive. Also in automatic self-contained units ranging from 1 ton to 25 tons capacity at 30 deg suction temperature, two and four cylinder types.



Baker Freon or
Methyl-Chloride Unit

Baker Freon or Methyl-Chloride Units in all capacities 1/4 to 35 tons at 30 deg suction temperature. Two and four cylinder air and water cooled types, featuring Timken bearings, double-trunk-type semi-steel pistons and full force feed lubrication. 3 to 35 tons capacity units equipped with multipass shell and tube condensers.

Baker Fan-Type Coldstream Units for refrigerated trucks and commercial air cooling. Rigidly constructed for compact, high-capacity heavy duty service. Finned coil surfaces and air velocity regulated for proper humidity and temperature conditions.



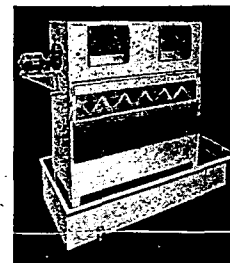
Baker Fan Type ColdStream Unit

Baker Ceiling Type Coldstream Blower Units designed for comfort cooling or commercial and industrial air cooling. Finned or galvanized bare pipe coil; fans direct-connected or V-belt driven. Also made as Floor Type Unit in a complete range of sizes from 2 to 16 tons refrigeration; 2000 to 16,000 cfm air capacity.



Baker ColdStream Blower Unit
—ceiling type, front and
rear views

Baker Coldstream Brine Spray Units for proper conditioning and humidification in packing plants, breweries, dairies, fruit cooling and storage rooms, and other applications requiring constant temperatures and relative humidity conditions. Available in gravity flow and forced draft types. Equipped with rust-proof eliminators to prevent brine or moisture escaping from the unit.



Baker ColdStream Brine Spray
Unit—forced draft type

Shell and Tube Condensers



Baker Shell and Tube
Condenser

Baker Condensers and Liquid Coolers made in all diameters and tube lengths up to 2500 sq ft of surface. Vertical, horizontal, multipass or singlepass types.

Carbondale Machine Corporation

General Offices:
Harrison,
New Jersey

CARBONDALE

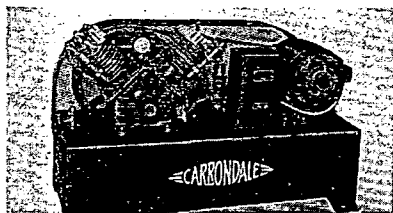
District Offices
and Representatives
in Principal Cities

AR-3621

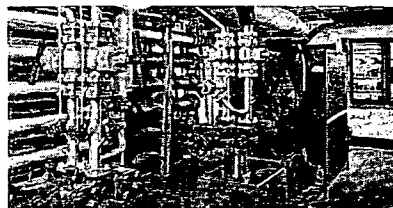
REFRIGERATION SYSTEMS FOR AIR CONDITIONING IN COMFORT COOLING OR INDUSTRIAL PROCESS

Complete refrigerating systems for use with Freon (F-12), 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 Dealer-Distributors in

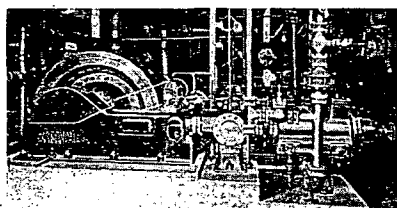
major cities to provide sales and engineering service and plan complete air conditioning systems of the central or unit type. Architects, Engineers, and Contractors are invited to consult with us. Write to Harrison, New Jersey, for bulletins covering products illustrated below.



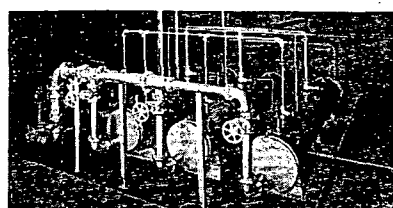
Freon or methyl chloride self-contained commercial units with motors up to 25 hp. and ratings up to 25 tons



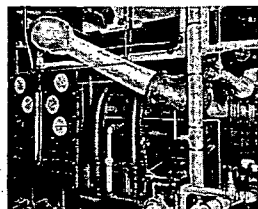
Horizontal CO₂ compressors, single and duplex, 25 to 500 hp. Complete installations with condenser and cooler units



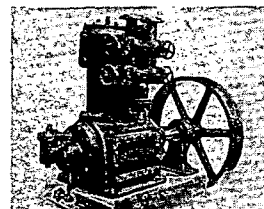
Horizontal ammonia compressors, single and duplex; belt, direct motor and steam drive; 60 to 100 tons. Capacity control features available



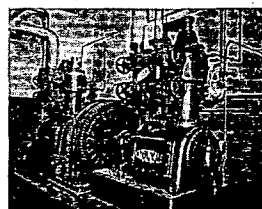
High and low side equipment: Coils, coolers, condensers, receivers, controls, pumps, valves, connections, fittings



Steam-jet vacuum cooling equipment. Chilled water 35 F. or over, 2 to 1000 tons at 60 F. Also centrifugal water vapor units, 60 to 300 tons



Vertical ammonia compressors, pressure-lubricated, roller main bearings, safety heads, patented Feather Valves, 2 to 200 tons



Special vertical Freon Compressors, pressure-lubricated, roller main bearings, safety heads, up to 150 tons

Compressor requirements above 100 tons are best met by single or duplex horizontal machines. Units up to 750 tons are available, proportioned for Freon and equipped with three-step stuffing boxes.

Carrier Engineering Corporation

AIR CONDITIONING - REFRIGERATION - INDUSTRIAL HEATING

Home Office: 850 Frelinghuysen Avenue
Newark, New Jersey

District Sales Offices: NEW YORK, PHILADELPHIA, CHICAGO, LOS ANGELES
Branch Offices and Dealers in Principal Cities

Export and Marine Division—Carrier-Brunswick-International, Inc., Newark, N. J.

ENGINEERING - SALES - SERVICE

Air Conditioning—(1) Apartment Houses, (2) Auditoria, (3) Banks, (4) Broadcasting Studios, (5) Chain Stores, (6) Clubs, (7) Department Stores, (8) Homes, (9) Hospitals, (10) Hotels, (11) Individual Rooms, (12) Libraries, (13) Offices, (14) Office Buildings, (15) Manufacturing; Ceramics, Chemical, Confectionery, Foods, Laboratory Products, Paper, Pharmaceuticals, Printing, Textiles, Tobacco, Miscellaneous, (16) Motion Picture Studios, (17) Public Buildings, (18) Restaurants, (19) Retail Stores, (20) Ships, (21) Theatres.

Refrigeration—(1) Chemical Processes, (2) Cold Storage, (3) Food Retailers, (4) Food Serving Establishments, (5) Fur Storage, (6) Marine Applications, (7) Meat Packing, (8) Product Cooling, (9) Special Applications, (10) Yachts, (11) Miscellaneous.

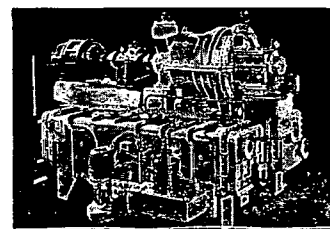
Industrial Heating—(1) Airplane Hangars, (2) Armories, (3) Auditoria, (4) Factories, (5) Garages, (6) Gymnasiums, (7) Lofts, (8) Machine Shops, (9) Markets, (10) Public Buildings, (11) Warehouses, (12) Miscellaneous.

A Complete Line of Equipment for Every Requirement

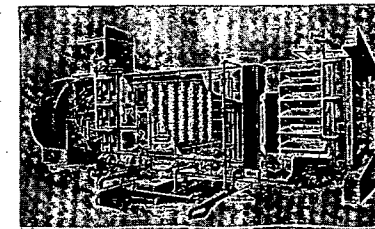
There is a Carrier system exactly fitted to each requirements and the nearest office of Carrier Engineering Corporation offers a complete service in solving any air conditioning, drying, industrial, heating or refrigerating problem. With an experience of over 25 years in the application of scientific principles every Carrier system is a finely engineered job which can be depended upon to meet the most exacting requirements. Carrier representatives are trained to analyze special problems and will survey your requirements without obligation. Engineers and architects are invited to make use of this special service.

Carrier Products—A complete line of equipment is manufactured, comprising: (1) Central Station Type air conditioning systems, (2) A complete line of unitary air conditioning equipment, (3) Industrial air conditioning units, (4) Industrial Humidifiers, (5) Individual room units, (6) Residential Air Conditioning Systems, (7) Unit Coolers, (8) Unit Heaters, (9) Centrifugal refrigerating machines, (10) Reciprocating type Refrigerating machines of all sizes and for standard and special refrigerants, (11) Special industrial and marine refrigerating machines.

Typical Carrier Equipment for Air Conditioning



Carrier Centrifugal Refrigerating Machine



Carrier De-humidifier and Air Washer

Engineering bulletins and descriptive literature on all products available upon request

Curtis Refrigerating Machine Co.

Division of
Curtis Manufacturing Co.

1959 Kienlen Avenue, St. Louis, Missouri, U. S. A.

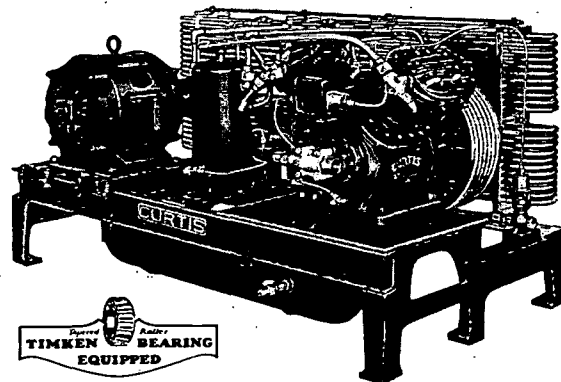
CURTIS Air-Conditioning Units

These units for air conditioning are outstandingly efficient and trouble-free, the result of 42 years' experience in building fine compressors and compressed-air machinery. The following features are the basis for the low-cost performance of Curtis Air-Conditioning Units:

Extra Capacity—Slow Operating Speed—Experienced Design—Low Upkeep—Rugged Construction—Fine Materials and Precision Workmanship.

Curtis enjoys the highest type capital and credit rating—A Curtis product won't become an "orphan."

Specify Curtis and be sure of a fine job.



Model No.	FWH 100-AC	FWH 150-AC	FWH 200-AC	FWH 300-AC	FWH 500-AC	FWH 750-AC	FWH 1000-AC	FWH 1500-AC
Motor Size (H.P.)	1	1½	2	3	5	7½	10	15
Bore (In.)	2½"	2½"	2½"	3"	3½"	3"	3½"	3½"
Stroke (In.)	2½"	2½"	2½"	3½"	3½"	3½"	3¾"	3¾"
Compressor Speed (R.P.M.)	375	375	515	360	425	425	450	600
Charge Lbs. Refrigerant F-12	10	12	12	14	18	20	20	20
Net Weight Lbs.	480	510	525	724	856	1375	1500	1550

Note: Models FWH-300-AC and larger have Timken Roller Bearing equipped compressors. Models FWH-750-AC and larger compressors are 4-cylinder "V" type design. Units supplied for Methyl Chloride instead of Freon if preferred.

Frick Company

(Incorporated)

Air Conditioning, Refrigerating
and Ice-Making Equipment

Waynesboro, Penna.

KANSAS CITY
LOS ANGELES
MEMPHIS
NEW ORLEANS
NEW YORK
OKLAHOMA CITY
PALATKA
PHILADELPHIA
PITTSBURGH
ST. LOUIS
SEATTLE
WASHINGTON

Distributors in 100



Principal Cities

AIR CONDITIONING WITH FRICK REFRIGERATION



10-Story Office Building at Topeka, Kan., Air Conditioned Throughout with Frick Freon Refrigeration

Is the term applied to our service in supplying refrigerating equipment and engineering assistance for complete air conditioning jobs, which are handled by Frick Branches and Distributors — located in principal cities throughout the world.

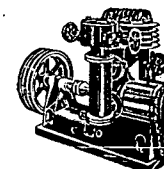
Estimates cheerfully submitted. Get data on special Frick air conditioning systems arranged for automatic direct-expansion operation. Economical, safe, highly satisfactory: let us refer you to typical installations. Ask for Bulletin 504.

FRICK FREON REFRIGERATION

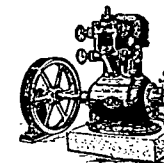
Includes the most complete line of enclosed type Freon compressors: large capacity, ample gas passages, pressure lubrication from internal pump, patented FLEXO-SEAL at shaft. Coils, coolers, condensers and controls for Freon systems. Bulletin 508.



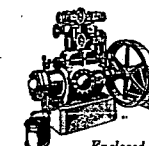
7 Frick Low Pressure Refrigerating Units Air Conditioned Dempsey's Restaurant in New York City



Frick Enclosed Freon Compressor



Enclosed Type Ammonia Compressor



Enclosed Compressor for Carbon Dioxide



Low Pressure Refrigerating Unit

AMMONIA REFRIGERATION

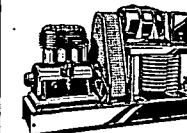
Machines in all capacities from ½ ton up. Combined units, vertical enclosed type compressors, horizontal machines: complete high and low sides. Widely used for air conditioning. Bulletins 102 to 138.



Four Frick Freon Compressors, Totalling 780 Tons Refrigeration, are used in Air Conditioning the Wings of the Administration Building of the Department of Agriculture, at Washington, D. C.

CARBON DIOXIDE REFRIGERATION

Six sizes of enclosed CO₂ compressors: smooth running, efficient and reliable machines; condensers, coolers, etc. Bulletins 118, 124, 208.



15-Ton Freon Unit for Air Conditioning Work

LOW PRESSURE REFRIGERATION

Commercial units in more than 50 sizes and types: motors of ¼ to 20 hp. Charged with Freon or methyl chloride. Air and water cooled condensers. Finned coils, fan and blower units, ice cube and beverage coolers, etc. Bulletins 97 and 98.



Dozens of Theatres are now Air Conditioned with Frick Refrigeration

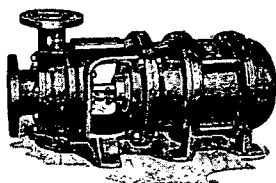
Ingersoll-Rand Company

11 Broadway, New York City

Branches or Distributors the World Over

R	BIRMINGHAM	CHICAGO	DETROIT	LOS ANGELES	PHILADELPHIA	SCRANTON
	BOSTON	CLEVELAND	DULUTH	NEWARK	PICHER	SEATTLE
	BUFFALO	DALLAS	EL PASO	NEW ORLEANS	PITTSBURGH	ST. LOUIS
	BUTTE	DENVER	HOUSTON	NEW YORK	SALT LAKE CITY	TULSA
		KNOXVILLE			SAN FRANCISCO	WASHINGTON

CAMERON PUMPS

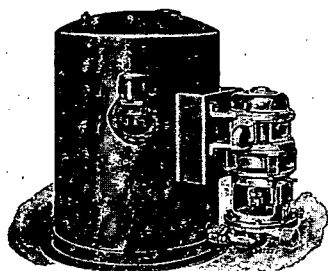


Cameron Motor pump

There is an efficient, reliable Cameron pump for every purpose. Single-stage centrifugal units range from 5 to 100,000 g.p.m., and multi-stage units (2 to 8 stages) range from 125 to 3000 g.p.m. for pressures up to and over 1500 lb.

The Cameron "Motorpump" is ideal for general service everywhere, ranging in sizes from $\frac{1}{4}$ to 40 hp. Pumps and motors are built as one compact and efficient unit on a single shaft. They can be mounted vertically or horizontally on the floor, wall or ceiling. Capacities range from 5 to 1000 g.p.m. for heads as high as 220 ft. single-stage; and from 20 to 275 g.p.m. for heads up to 500 ft. two-stage. Motors for any common current conditions; open, splash-proof, totally-enclosed, or explosion-proof types.

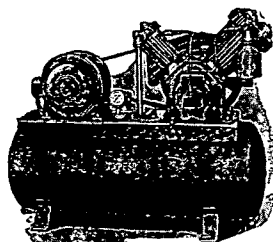
CONDENSATE RETURN UNITS



"Motorpump" Condensate Return Units are designed to return condensation from radiation systems, heaters, steam coils, etc.

The units consist of standard "Motor-pumps" mounted on 15-, 30-, or 60-gal. tanks and controlled by a float switch. They are suitable for industrial uses, for use in office buildings, apartment houses, department stores, school buildings, estates and clubs, etc. They are often-used to replace inefficient steam traps.

AIR COMPRESSORS



Type "30" Two-Stage Compressor

A "Type 30" Air Compressor is an excellent source of compressed air for use with regulating devices or control equipment, and for many other applications requiring small capacities. It is a self-contained plant, consisting of a compressor, driving motor, air receiver and automatic pressure-control switch. The compressor is either single- or two-stage, using I-R plate valves. Cylinders and inter-cooler are air cooled. Capacities range from 1.2 to 82 cu. ft. per min. Pressures range from 5 to 1000 lb. per sq. in.

Ingersoll-Rand manufactures more than 1000 sizes and types of compressors including a complete line of ammonia compressors.

OTHER I-R PRODUCTS

Surface condensers; counter-current and ejector-jet barometric condensers; steam-jet ejectors; vacuum pumps; centrifugal blowers; air aftercoolers and receivers; Diesel and gas engines; air-and electric hoists; rock drills; and pneumatic tools of many kinds.

Ingersoll-Rand

WATER-VAPOR REFRIGERATION

The use of water as a refrigerant makes the system 100 per cent safe; eliminates seasonal cost for replacing refrigerant; simplifies installation; and reduces power consumption.

Two types of I-R units—Steam-Jet and Centrifugal—permit the selection of the most economical type of equipment to meet widely varying conditions of steam or electric power, cooling water supply and refrigerating requirements.

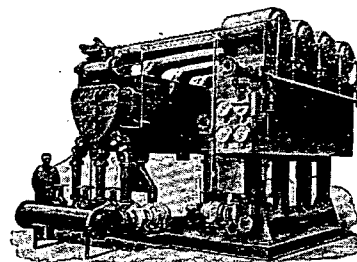
Sustained performance and overload capacity permit a full tonnage rating at the highest chilled-water temperature compatible with the service conditions. Direct evaporation eliminates scaling or fouling of evaporator tubes, and the volumetric capacity of either steam-jet boosters or the

centrifugal compressor remains practically fixed for the life of the unit. An overload of 15 or 20 per cent can be carried for peak conditions with only 3 to 5 deg. increase in chilled-water temperature.

Because it is simple, safe, and economical, I-R Water-Vapor Refrigeration has unlimited uses. It has been applied to: Air conditioning in office buildings, factories, department stores, restaurants, theaters, hospitals, hotels, museums, government buildings, and cold storage aboard ships.

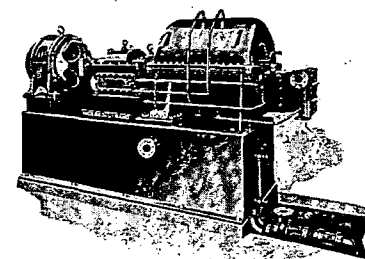
In industry, it is being used in automobile factories, chemical plants and rubber mills. Industrial applications are constantly increasing in variety and number.

STEAM-JET TYPE



Quietness and absence of vibration and moving parts characterize steam-jet units. Water, the refrigerant, is evaporated at a high vacuum and the vapor is compressed by highly efficient steam-jet boosters. The water chilled by this evaporation is the cooling medium. The condenser may be of surface or barometric type. Exclusive I-R features simplify control. Sizes from 10 tons upward.

CENTRIFUGAL TYPE



Centrifugal compressors developed by Ingersoll-Rand specially for this service, replace the steam jets where sufficient steam is not available or condensing water is limited. They can be furnished with electric-motor or highly efficient steam-turbine drive. An important feature is self-regulation; units simply float on the load, giving inherently reduced power consumption with reduced demand. Sizes from 50 to 300 tons.

Norge Division

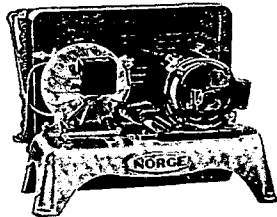
Borg-Warner Corporation

606-670 E. Woodbridge Street, Detroit, Michigan

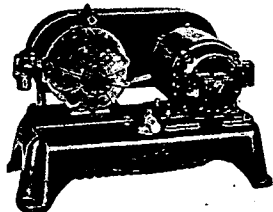
Electric Refrigerating Equipment for Any Commercial Purpose

The Norge line of commercial refrigerating equipment meets every demand from smallest delicatessen refrigerator to the large air conditioning job. Specially designed coils are quickly available if needed. A competent staff of highly

trained refrigeration engineers is maintained to insure that everyone who invests in Norge refrigeration for commercial use gets exactly the type of refrigeration he needs.

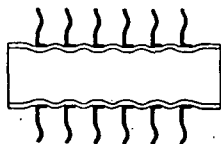


NA-150, 1 1/4 hp air-cooled Rollator Compressor

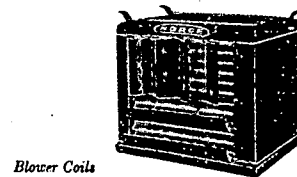


NW-33, 3/4 hp water-cooled Rollator Compressor

Rollator Compressor Units—Condensing units with motors powered from 1/4 hp to 1 1/2 hp—air cooled and water cooled. Methyl Chloride used as standard refrigerant.



Norge Coil Construction—This diagram shows how fins are rigidly and permanently bonded onto tubes by hydraulic expansion, thus improving heat absorbing capacity.

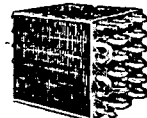


Blower Coils

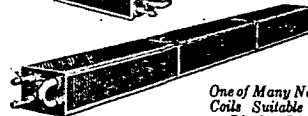
Blower Coils—Especially adapted to beer keg storage, dry type milk coolers and similar applications. Fan motors from 1/80 hp to 1/4 hp available for any type current.



Market Cooler Coil



Grocery or Restaurant Refrigerator Coil



One of Many Norge Coils Suitable for Display Case

Cooling Coils—Standard coils meet every requirement for temperature control and humidity. For most installations, coils are made of 1 in. heavy copper tubes with individual aluminum fins.

Reciprocating Compressors—Condensing Units with motors powered from 1 to 15 hp, using Methyl Chloride or Freon as standard refrigerant. Also available with gasoline engine power.

See also Pages 847 and 892

Universal Cooler Corporation

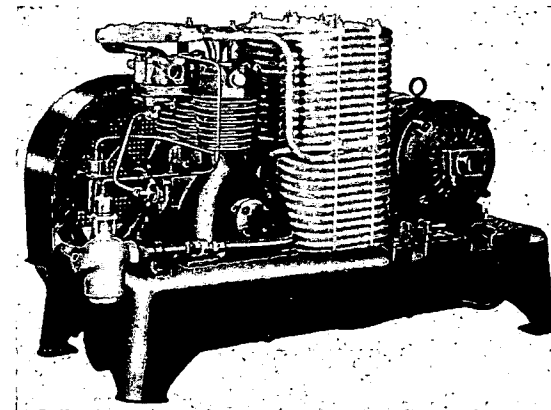
DETROIT



MICHIGAN

14 Years Experience in Commercial Refrigeration

REFRIGERATING machines built by the UNIVERSAL COOLER CORPORATION are meeting the exacting requirements of the air conditioning industry. The line is complete—from the small units for self-contained room coolers to the refrigerating machines for both comfort and industrial cooling. These units can be supplied charged with either Freon or Methyl Chloride, and are applicable to practically all types of lowside equipment.



Model W1500FH 15 Hp

UNIVERSAL Cooler condensing units have been developed by engineers fully conversant with the requirements of the air conditioning industry. During the past few years a great many machines have been installed, and the 1936 line reflects the experience gained on applications of every kind.

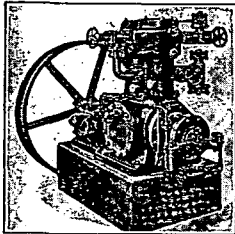
The executive, sales and manufacturing policies which have guided its destinies are as enduring a part of Universal Cooler as its physical properties. Customers will receive the highest character of service—the public an outstanding degree of product dependability—always.

The Vilter Manufacturing Company

Since 1867

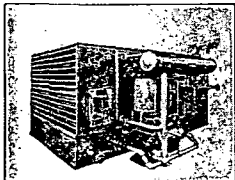
Milwaukee, Wisconsin

For nearly seventy years the name Vilter has been identified as a pioneer in the development of equipment for industrial refrigeration, ice making, cooling, air conditioning, etc.

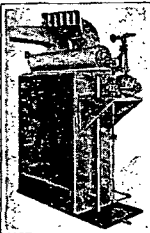


Vilter Compressors are offered in a complete range of sizes from one ton to several hundred tons. These are available in designs for use of any of the commonly known refrigerants.

Self-contained units or individually mounted compressors, condensers, etc., according to the application. V-belt or synchronous motor drive; hand or automatic control.

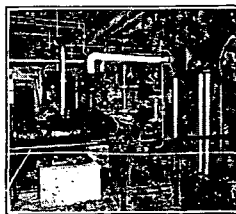
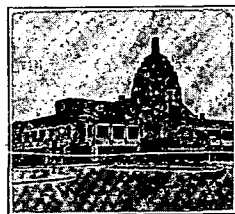
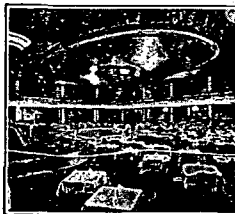


Washers—Designed for industrial air conditioning. Positive control of humidity, temperature and circulation of air. Automatic or hand operated. Eliminators incorporated for entrained moisture removal. Odors and dust removed by water sprays. No filter replacement. Low cost of operation. Good for low or high temperature cooling.



Unit-Type Air Conditioners—Mono-Unit air conditioners built in complete range of sizes and types, from the small ceiling unit to large floor units. Dry type, above 35 F, with quiet Plexiform type fan; spray type, below 35 F, with Plexiform type fan; junior, above 35 F, with disc type fan.

Super-Capacity Cooler—High heat transfer from ammonia to water. Exclusive float mechanism assures flooded, trouble-free operation. Unit completely galvanized and built into galvanized iron tank. Completely non-rusting. No danger of freezing up or of breaking pipes. Shell and tube type water or brine coolers also available.



Wittenmeier Machinery Company

AIR CONDITIONING ENGINEERS AND CONTRACTORS

850 N. Spaulding Avenue

Chicago, Ill.

103 Park Avenue
New York, N. Y.

AMERICAN REFRIGERATING CO.
DETROIT, MICHIGAN
H. J. KELLY
NEW ORLEANS, LA.

WITTENMEIER MACHINERY CO.
COLUMBUS, OHIO
WITTENMEIER MCH'Y. CO. OF CANADA, LTD.
HAMILTON, ONTARIO

Wittenmeier, a name associated with Refrigeration in all its phases and applications for over 35 years, offers complete Air Conditioning and Refrigerating Systems for industrial processes and bodily comfort, either in Central or Unit Systems from 1/4 ton capacity up.

Refrigerants. CO₂—Freon—Ammonia—Methyl Chloride—Steam—Water Vapor.

When the subject turns to Refrigeration, immediately the name **Wittenmeier** comes to mind. Credit for introducing and pioneering the Carbonic Safety System belongs to **Wittenmeier**. To **Wittenmeier** also belongs the credit for having developed this system to its present high state of efficiency. Add to this the distinction of having made the very first installation for "Comfort Cooling" and it at once becomes apparent why the name **Wittenmeier** occupies such an enviable position in the industry and why the **Wittenmeier Machinery Company** is classed with Reliable and Responsible Contractors. While our standing in the Industry is reflected in the thousands of CO₂ installations that we have made throughout the country, we are, nevertheless, fully equipped and qualified to install any of the systems listed above.

Wittenmeier, with Air Cooling and Conditioning Experience that dates back to 1908, can offer helpful suggestions. Architects, Engineers, Contractors and others should avail themselves of this service. Whether room, office, shop, restaurant, bank, theatre, auditorium or large office building, there is a **Wittenmeier** System best suited for the purpose. Let us make a survey of your requirements and give you our unbiased recommendations of the system best suited for your needs. There is no obligation.

Wittenmeier made the first Hotel Air Cooling Installation. Congress Hotel, Chicago, Ill.—1908.

Wittenmeier made the first Theatre Cooling Installations. Riviera and Central Park, Chicago, Ill.—1918.

Wittenmeier made the first Dance Hall installation. The Trianon Ballroom, Chicago, Ill.—1921.

Other outstanding **Wittenmeier** installations are Michigan Theatre, Fox Theatre, Book-Cadillac Hotel, Detroit, Mich.; Board of Trade, Hotel Sherman, Southtown Theatre, Aragon Ballroom, Chicago, Ill.; Manufacturer's Trust Co., Hotel New Yorker, Capitol Theatre, New York, N. Y.; Loew's State Theatre, United Artists Theatre, Los Angeles, Calif.; New Majestic Theatre and Municipal Auditorium, San Antonio, Texas; Saenger Theatre, Criminal Court Building, Orpheum Theatre, New Orleans, La.; Federal Reserve Bank, Dallas, Texas; Fox Theatre, St. Louis, Mo.

There is no substitute for Experience.

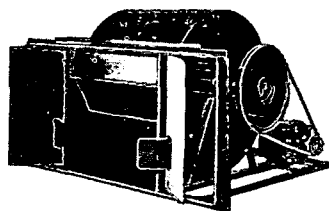
Wittenmeier Continuously Since 1897

Air Controls, Inc.

Div. of
The Cleveland Heater Co.

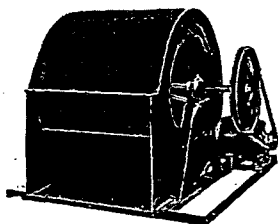
1937 West 114th Street, Cleveland, Ohio

Manufacturers of AIR-PAK Forced Air Heating and Air Conditioning Units, A. C. Blowers and Airate Air Circulators



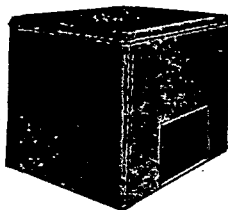
A. C. BLOWER WITH LOUVRES

A high efficiency blower for forced air heating and air conditioning installations. Automatic Louvres prevent damage to furnace due to overheating and are essential for safety on oil, gas and coal fired furnaces.

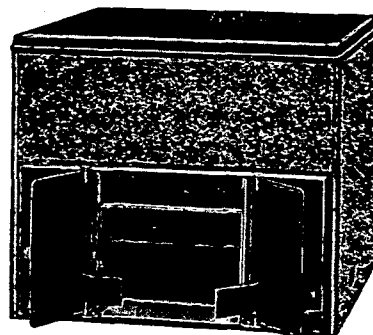


A. C. BLOWER WITHOUT LOUVRES

A. C. Blowers are quiet and are designed to deliver proper amount of air whether filters are dirty or clean. They occupy small space but deliver large volume of air. Send for bulletin describing them.



As shown above the AIR-PAK can be obtained without louvres when so desired. Send for Simplified Selector Chart and Performance Data.



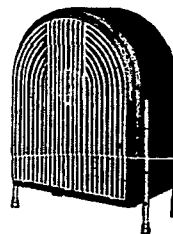
THE AIR-PAK

The AIR-PAK blower-filter unit for forced air heating and conditioning is encased in an attractive red casing with black top. Casing comes in panels and can be assembled in 10 min. without screws, bolts or slip joints. Filters have high cleaning capacity and low resistance. Blower is quiet and free from vibration. All moving parts are rubber mounted. Self-aligning bearings require oiling once a year. Casing has removable back which permits cool basement air to be drawn through filters for summer use.

The AIR-PAK is equipped with the famous Patented Automatic By-pass Louvres which permit unobstructed circulation by gravity when blower is not running.

THE AIRATE

A large volume circulator for commercial or attic ventilation. Attractive, quiet, powerful and economical. Write for bulletins. No. 29 and S. C. 21.



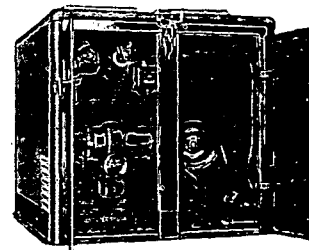
Delco Appliance Corporation

Subsidiary of General Motors
Rochester, N. Y.

DELCO-HEAT

Delco-Heat Conditionair

One Compact, Completely Harmonized Unit—Filtering the air which is circulated in the home is one of the most important functions of air conditioning from the standpoint of health. It removes a large percentage of bacteria, pollen and dust from the air. It reduces the amount of cleaning required in the residence and adds life to fabrics and drapes. Humidification of the air, in the winter, is important to health. It reduces the chances of colds and other winter ailments. It also prevents deterioration of woodwork. Furniture, with the proper amount of moisture, does not dry up and become loose at the joints. Summarizing: The Delco-Heat Conditionair filters the air, humidifies, automatically heats and circulates the air.



Ample Heating Surface to Maintain Temperature Desired—The especially designed combustion chamber and heat passages are unique in construction. They expose a large heating surface to the air as it is circulated over the stream-lined unit. Rapid heating with great economy of fuel results. The Delco-Heat principle of combustion gets full heating effects from low cost domestic fuel oil.

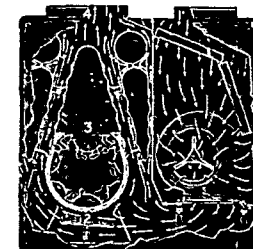
Completely Automatic in Operation—The thermostat, set to the desired temperature, controls the operation of the Delco-Heat Conditionair.

Central Plant Duct System—The Delco-Heat Conditionair requires a central plant duct system. A main distributing duct, from which branch ducts go to individual rooms, is taken from the top of the unit. Return ducts are recommended through

which to carry the cooled air from the house back to the Conditionair.

Blown Glass Filter Units

The operation is simple. The air passes down from the cold air return duct, and travels through the filters, which remove dust, pollen and bacteria. Air passes to the cascade type humidifier and the heating compartment where it is automatically heated. Air in the home is cleaned approximately eight to ten times per hour. Each filter unit has an area of 384 sq. in. and is 2 in. thick. The filtering material consists of an upper layer of fine strands of blown glass and a lower layer of extremely fine strands of blown glass. The upper layer of glass strands is approximately twice as thick as the lower layer. Strands are covered with a special adhesive material. Models DA-2, and GA-2 have four filter units. DA-3 has nine filters. Model DA-1 has two filters.



Delco-Heat Conditionair (Oil or Gas Burning)

Automatic Controls for Conditionair

Delco-Heat Conditionair is available oil-fired in three models; model for natural, artificial or mixed gas also available. Sturdiness, positive alignment of moving unit assures quiet operation due to freedom from vibration and rattle. No part of the Delco-Heat mechanism is in the combustion chamber, thus warping, burning out, and corrosion are eliminated.

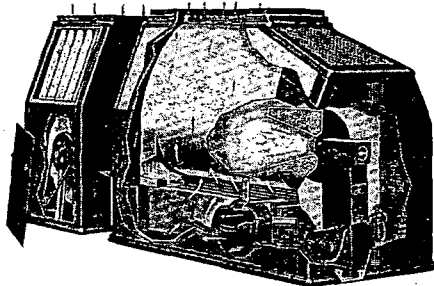
All controls needed for automatic heating are standard with the Delco-Heat Conditionair.

See also Pages 876 and 890

Gar Wood Industries, Inc.

AIR CONDITIONING DIVISION
7924 Riopelle Street, Detroit, Mich.
Licensed Distributors in All Principal Cities

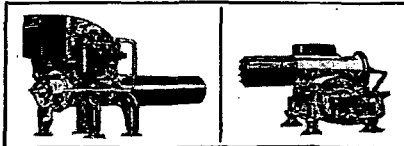
TEMPERED-AIRE UNIT



Tempered-Aire heating and air conditioning equipment includes filters, blower, humidifier, furnace (with "Economizer") and integral oil burner of pressure atomizing type. Cloth filters can be easily laundered. All units can be equipped with a water heating coil for use in winter. The installation of an auxiliary duct for intake of outside and basement air in summer is desirable. Air can then be drawn from the basement during the day and from the outside at night to provide ventilation and cooling during warm weather. Change from basement air to night air may be manually or automatically controlled.

Ratings and Dimensions	No. 102	No. 103	No. 104	No. 105	No. 105-C
Btu 1 Hour at Bonnet.....	120,000	165,000	225,000	300,000	400,000
Btu 1 Hour at Grilles.....	100,000	135,000	185,000	245,000	
Air Delivery, CFM.....	1000 to 1575	1375 to 2150	1875 to 2950	2500 to 3925	3000 to 4500
Heating Surface, Firebox, Square Feet.....	30	33	44	55	55
Heating Surface, Economizer, Square Feet.....	60	99	132	165	165
Total Heating Surface.....	90	132	176	220	220
Overall Length, Inches.....	96 3/4	117 3/4	140 1/4	156 1/4	156 1/4
Overall Width, Inches.....	40	40	40	40	40

CONVERSION OIL BURNER—Provides automatic oil burning equipment for coal fired heating plants. Pressure atomizing type, handle heavier, cheaper grades of fuel. Sturdy, noiseless, easily accessible.



Model "K"

Extremely flexible, due to system of triple range blowers and two motor speeds. Handles capacities from 412 to 2500 sq ft net steam radiation.

Model "H"

An efficient, low cost unit, to supply automatic oil heat for small homes. Handles capacities up to 625 sq ft net steam radiation.

MODEL "Q" AUTOMATIC OIL FIRED WATER HEATER—Provides quick, convenient, less expensive and unlimited supply of domestic hot water for industrial and commercial buildings. Complete, quiet. Burns low priced No. 3 fuel oil.

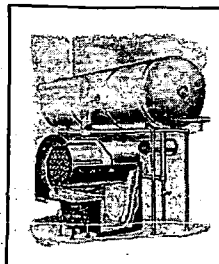
Capacities and Dimensions

Two sizes, 200 and 300 gal capacity. 100° F rise per hour.

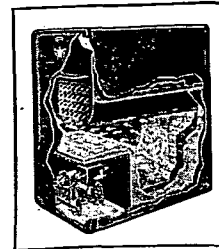
Overall height 48 in.

Overall width 28 1/4 in.

Length 49 in. and 61 in. respectively.



MODEL "R" BOILER-BURNER UNIT



A compact, fire-tubular steam or hot water heating boiler, with an integral oil burner.

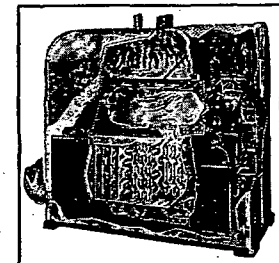
Boiler built of heavy rust-resisting boiler plate, electrically welded. The combustion chamber walls are carried clear to the crown sheet. The result is a considerably higher temperature of fire than is found in the conventional boiler, and better combustion of No. 3 oil.

Since there are no water legs, the bulk of the heating surface is concentrated in the firetubes, or secondary heating surface, resulting in maximum extraction of heat from the hot gas which leaves the firebox and consequent low stack temperature, high efficiency and operating economy.

Ratings and Dimensions	R475	R750	R1000	R1400	R1800
Maximum Net Steam Load, Square Feet.....	475	750	1000	1400	1800
Maximum Net Hot Water Load, Square Feet.....	760	1200	1500	2240	2880
Maximum Gross Steam Load, Square Feet.....	712	1125	1500	2100	2700
Maximum Gross Hot Water Load, Square Feet.....	1140	1800	2400	3360	4050
Heating Surface, Square Feet.....	52	68	84	118	154
Overall Width, Inches.....	30 3/4	30 3/4	37 1/4	37 1/4	37 1/4
Overall Length, Inches.....	53	65	59 3/4	66	80

MODEL "J" BOILER-BURNER UNIT

A deluxe, down-draft, steel, steam or hot water heating boiler of unique design, equipped with its oil burner, all beautifully finished to give discriminating owners not only an efficient but handsome unit. The design of boiler gives a maximum of heating surface resulting in almost complete heat transfer. Gar Wood external indirect water heaters are available for both Model R and J boilers, to provide year round domestic hot water.



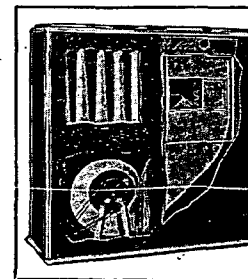
Ratings and Dimensions	No. 5	No. 8	No. 12	No. 16	No. 16-35
Maximum Net Steam Load, Square Feet.....	500	800	1200	1600	2800
Maximum Net Hot Water Load, Square Feet.....	800	1280	1800	2400	4480
Maximum Gross Steam Load, Square Feet.....	750	1200	1800	2400	4200
Maximum Gross Hot Water Load, Square Feet.....	1200	1920	2880	3840	6720
Heating Surface, Square Feet.....	72	105	152	199	199
Overall Width, Inches.....	31 1/2	31 1/2	31 1/2	31 1/2	31 1/2
Overall Length, Inches.....	42 3/4	50 1/4	60 3/4	71 1/4	71 1/4

INDIRECT AIR CONDITIONING CABINET

Combined with "R" or "J" boilers, it provides heating, humidifying, filtering and circulation of air.

The air drawn by blower is first cleaned by dry cloth filters, then forced through the humidifying chamber, where proper amount of healthful moisture is added by passing through a warm vapor mist, regulated by a room humidistat. The air is then warmed to correct temperature by passage through the copper blast heater. Standard units, in capacities from—

111,000—497,000 Btu Hour at bonnet
89,000—397,000 Btu Hour at grilles



GENERAL ELECTRIC

COMPANY

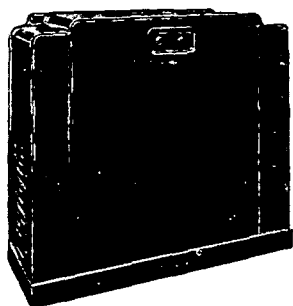
AIR CONDITIONING PRODUCTS

Air Conditioning Department, Bloomfield, N. J.

FOR AIR CONDITIONING, INCLUDING AUTOMATIC HEATING

A Complete Line of Equipment for Summer, Winter, and Year 'Round Application

G-E UNIT ROOM AIR CONDITIONERS

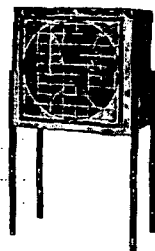


For Summer

Type FC-2 cools, dehumidifies, ventilates, filters. Requires no piping connections or special wiring. Has three hermetically sealed condensing units. Other G-E types have condensing units remotely located, and may be installed in multiples with one condensing unit.

G-E Store Cooler

Wall or ceiling mounted. Cools, dehumidifies, and circulates air. Has G-E Aphonic type fan. (Types AG-10-12 and 15).

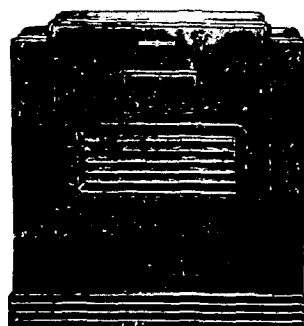


G-E Air Circulator

Removes overheated air by day; draws in cool outdoor air at night. Effectively installed at attic windows in homes.

G-E Condensing Units

Exceedingly flexible range—1 hp to 20 hp—to accurately fit any need and assure economical operation.



For Year 'Round

G-E units to provide automatically cooled, dehumidified air in summer, and warmed, humidified air in winter are available either with condensing unit built in (Type FR-1) or remotely located (Types AD-3 and AD-4).



G-E Air Conditioner for Warm Air Systems (Types AA-3 and AA-4)

Combined with G-E Oil or Gas Furnace, provides heated, humidified, filtered, and circulated air. Fan draws air from return duct through steel wool filters, passing over steam heating surface and humidifying screens

to delivery ducts. Humidifying water is heated by indirect coil, flow controlled by magnet valve operated by humidistat. Cooling equipment for summer can be added easily.

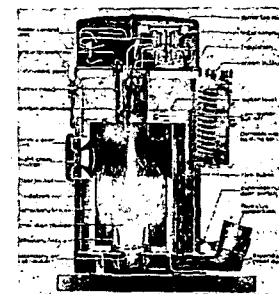
GENERAL ELECTRIC

COMPANY

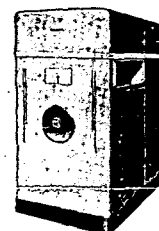
AIR CONDITIONING PRODUCTS

Air Conditioning Department, Bloomfield, N. J.

G-E Oil Furnace—(Two sizes—Types LA-4 and LA-5)—Fully coordinated—Boiler, burner, water heater, controls in one enclosed unit, all made and guaranteed by G-E. Can burn cheap grades of oil—to produce low cost heat. Employs entirely new principle of oil atomization. Heavy arc-welded steel boiler with water-backed combustion chamber. Low standby losses. Sub-atmospheric pressure prevents escape of odors. Large, non-clogging oil orifice. Non-drip nozzle is air cooled. All controls built-in, including four second self-checking flame detector. Expansion relief door with electric cut-out. Synchronous motor-driven primary control and electric clock thermal control. All rotating parts on one shaft, automatically oiled, completely sealed in. Burner head, motor compressor, controls separately and quickly removable.



G-E Gas Furnace—All parts by G-E, fully coordinated for high efficiency. Plain or de luxe jacket. RM (small residential) Series, 4 sizes, A.G.A. steam ratings 320 to 800 sq ft. RK (larger residential) Series, 6 sizes, A.G.A. steam ratings 660 to 1760 sq ft. CK (commercial) Series, 14 sizes, A.G.A. steam ratings 1980 to 9680 sq ft. Automatic secondary air control on RK and CK reduces standby losses, conserves fuel. Automatic control of steam pressure, low water cut-off, and temperature limit. Safety controls are gas operated to assure positive action. Boilers meet A.S.M.E. Specifications and carry A.G.A. approval.



G-E Warm Air Conditioner, Oil Fired (Type LB-4)—Consists of combustion-heat transfer unit, oil burner unit, centrifugal fan, humidifier, filters, controls, and necessary air, oil, water, and electrical connections. Completely enclosed in attractive two-tone gray cabinet, black and chrome trim. Of direct-fired type, developed especially for residential air conditioning, it circulates clean, warm, moistened air through ducts. G-E thermal control and anti-syphon screen valve included. Total output, 133,000 Btu per hour; humidifying, 10 lb per hour. Extremely quiet; electrically welded air-tight furnace—no place for gases and odors to escape; flame detector shuts off oil in less than four seconds; burns cheap oil on exclusive G-E impact-expansion principle; sealed-in-steel motor, self-lubricated; burner nozzle air cooled.

See also Pages 968 and 969

Gilbert & Barker Mfg. Co.

SUBSIDIARY OF STANDARD OIL CO. (N. J.)

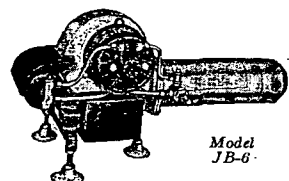
"Gilbarco" Automatic Oil Heating and Air Conditioning Equipment
Springfield, Massachusetts

GILBARCO FLEXIBLE-FLAME OIL BURNERS FOR ALL TYPES OF HEATING PLANTS

Complete Range of Models

700 to 10,000 Sq Ft of Steam Radiation

1120 to 16,000 Sq Ft of Hot Water Radiation



Model
JB-6

tem. Due to the flexibility of the flame, the "Gilbarco" Burner is enabled to more completely fill the fire box with a heat-giving radiant flame.

Each "Gilbarco" Burner combines the following definite advantages: 1. Flexible Flame Combustion—assuring tailor-made application. 2. Forced draft—insuring complete combustion. 3. Radiant type flame—gives great heat output. 4. Constant electric ignition. 5. Separate air and oil controls, insuring complete combustion. 6. Oil filter in oil line—assuring clean oil at all times. 7. Radio interference eliminator. 8. Burner installed outside boiler insures long life, easy inspection and service and general operating efficiency. 9. Quiet operation—no gears, belts or disturbing noisy mechanisms.

SPECIFICATIONS OF AND CAPACITIES FOR 60 CYCLE MOTOR BURNERS (Write to us for capacities of D.C. and odd cycle motor burners)

MODEL NUMBER	SI-190	*JB4	*JB4	*BS10	*CS20	*CS30
Max. oil Capacity per hour, U. S. Gallons	2	3.15	5.25	11.0	15.00	25.00
Total Steam Radiation (Radiation, piping, and pickup)	700	1200	2000	4400	6000	10,000
Total Hot Water Radiation (Radiation, piping, and pickup)	1120	1920	3200	7040	9600	16,000
Ignition	Continuous electric single transformer	Continuous electric single transformer	Continuous electric single transformer	Continuous electric single transformer	Continuous electric two transformers	Continuous electric two transformers
Motor (size)	1/12 H. P.	1/8 H. P.	1/5 H. P.	1/6 H. P.	1/3 H. P.	1/3 H. P.
Controls	Stock mounted Protectorelay and Thermostat	Stock mounted Protectorelay and Thermostat	Stock mounted Protectorelay and Thermostat	Protectorelay Protectorelay and Thermostat	Protectorelay Protectorelay and Thermostat	Protectorelay Protectorelay and Thermostat
RPM	1150	1150	1750	1750	1750	1750

GILBARCO SERIES "A" BOILER-BURNER UNITS

There are three models for both steam and hot water systems in the "A" series which range in capacity from 350 ft of net standing radiation to 550 ft of net standing radiation for steam and 560 ft to 880 ft of net standing radiation for hot water. The boiler, especially designed for oil fuel, is of cast iron, which can be moulded into the most effective kind of heat-absorbing surfaces. "A" series models are completely automatic and the high efficiencies which are attained in the boiler are due to the fact that the heat gases are compelled to be in direct contact with the heat-absorbing surfaces at all times. All series "A" Gilbarco Boiler-Burner Units are designed to include: Built-in hot water coil and aquastat; Low-water cut-off on steam systems; Pre-cast refractory combustion chamber; "Pressuretrol" on steam systems; Surface aquastat on hot water systems.

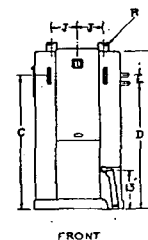


Outside view of the "B" Series Gilbarco Boiler Burner Unit. The lower-center panel on the "A" Series is perfectly plain.

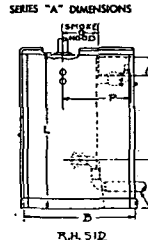
*SERIES "A" RATINGS

No.	STEAM		HOT WATER		DOMESTIC WATER-COIL CAPACITY		CHIMNEY	
	Gross	Net	Gross	Net	Steam	Hot Water (at 150°)	Size Ins.	Height Ft.
A-19	548	350	876	560	40 gallons	20 gallons	8 x 12	30
A-20	700	450	1120	720	40 "	20 "	8 x 12	30
A-22	860	550	1376	880	40 "	20 "	8 x 12	30

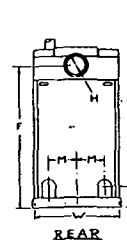
SERIES "A" DIMENSIONS



FRONT



R.H. SIDE

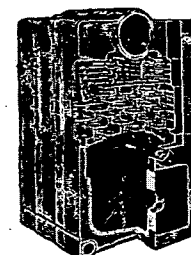


REAR

	A	B	C	D	E	F	G	H	J	K	L	M	N	W	P	R	S
A19	59K	42	47	44	23 1/2	31H	18 1/2	8	7 1/2	16 1/2	54 1/2	12H	3 1/2	34K	25	2-3/4	2-3/4
A20	59K	45	47	44	23 1/2	31H	17 1/2	9	8 1/2	16 1/2	54 1/2	12H	3 1/2	34K	26	2-3/4	2-3/4
A22	59K	47	47	44	23 1/2	31H	18 1/2	9	9H	16 1/2	54 1/2	12H	3 1/2	34K	27	2-3/4	2-3/4

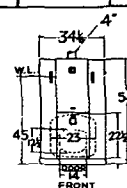
GILBARCO SERIES "B" BOILER-BURNER UNIT

There are five models for both steam and hot water systems in the "B" series which range in capacity from 500 ft to 1300 ft of net standing radiation for steam, and 800 ft to 2080 ft of net standing radiation for hot water. Like the "A" series units, the boilers of the "B" series are of cast iron construction insuring peak efficiency and exceptional durability. The specially designed boiler combines the following important efficiency factors: (1) Extended fin type heating surfaces. (2) Low draft loss. (3) Large combustion chamber. (4) Quick steaming. (5) Water-backed combustion chamber. (6) Ground joints between sections. (7) Unique gas travel. (8) Positive internal water circulation. (9) Large steam liberating surface. (10) Compact size.

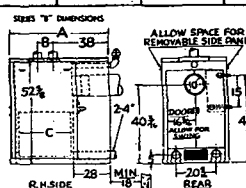


*SERIES "B" RATINGS

No.	STEAM		HOT WATER		DOMESTIC WATER-COIL CAPACITY		CHIMNEY	
	Gross	Net	Gross	Net	Steam	Hot Water (at 150°)	Size Ins.	Height Ft.
B-3	780	500	1248	800	100 gallons	50 gallons	8 x 12	30
B-4	1095	700	1743	1120	100 "	50 "	8 x 12	30
B-5	1405	900	2248	1440	100 "	50 "	8 x 12	35
B-6	1720	1100	2752	1760	100 "	50 "	12 x 12	35
B-7	2030	1300	3246	2080	100 "	50 "	12 x 12	40



FRONT



R.H. SIDE

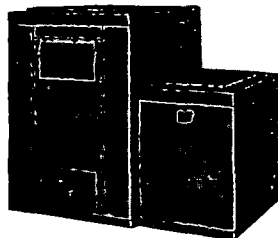
	B3	B4	B5	B6	B7
MINIMUM CLEARANCE OF	3	4	5	6	7
A	51"	59"	67"	75"	83"
B	0"	0"	16"	24"	32"
C	15K	25K	31K	39K	47K

*NOTE.—All domestic water heater ratings are based on 50°-160° F. rise in three hours at a minimum boiler water temperature of 180° F. In hot water boilers it will be proportionately less depending on operating water temperature. Net rating means actual standing, cast iron radiation, piping and pick up losses having been allowed for. It does not, however, include domestic hot water load, and this should be deducted at the rate of 0.75 sq. ft. (steam) or 1.2 sq. ft. (water) for each gallon of storage heater capacity. Efficiency—Series "A" boilers show a test efficiency of 77%, Series "B" boilers show a test efficiency of 80%, plus over-all at rated capacity for continuous firing.

NOTE.—When ordering be sure to specify whether the unit is for a steam or hot water system

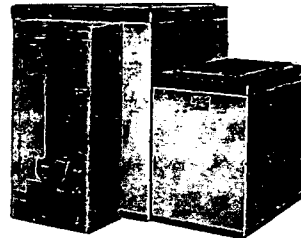
GILBARCO SERIES "F" AIR CONDITIONING UNITS HEAT - CLEAN - HUMIDIFY - CIRCULATE

"Gilbarco" series "F" Air Conditioning Systems are efficient, dependable and unusually economical in operation. They are specially designed for oil fuel and develop a high degree of heating efficiency. The oversized radiator within the furnace, with its long, retarded gas travel insures a great amount of heat with economical fuel consumption. The filtering and humidifying equipment is engineered according to advanced principles of this new science and the fan and motor are of sufficient capacity and durability to insure many years of efficient service.

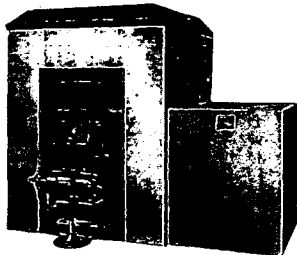


De Luxe Models FBD22120 and FBD27200

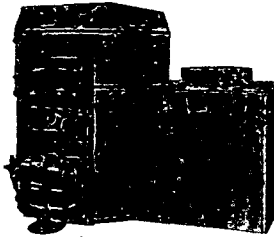
For
Specifications
See
Below



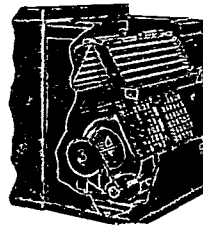
Models FBS27150, FBS32250, FBS32257, FBS32300, FBS32400, FBS32475



Model FBS27100



Model FBR27100



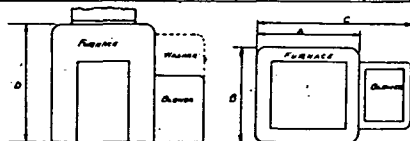
Filter-Washer-Blower Unit

SPECIFICATIONS "GILBARCO" SERIES "F" AIR CONDITIONERS

UNIT NUMBERS*		B.T.U.	Burner	Burner†	BLOWER					DIMENSIONS (Inches)						Smoke	CHIMNEY		
Without Washer	With Washer	at Register	Model No.	Enclosed	C.F.M. Max.	Fan Diam.	Motor H.P.	Filters #	A	B	C	D	E	F	Pipe Diam.	FINISH	Size Ins.	Height Ft.	
FBR27100		100,000	S2	No	1400	12	1/2	2	16 x 25	46	72	58			8"	Galvanized	8 x 8	30	
FBS27100		100,000	S2	No	1400	12	1/2	2	16 x 25	45	69	58			8"	Sage Green, Ripple Enamel	8 x 8	30	
FBD22120	FBDW22120	120,000	S2	Yes	2000	12	1/2	4	20 x 20	43	51	71 1/2	61		8"	Green Flintflex	8 x 8	30	
PBS27150	FBSW27150	150,000	S2	No †	2200	16	1/2	6	16 x 25	50	53	78 1/2	61	26	14"	8"	Gray Flintflex	8 x 10	30
FBD27200	FBDW27200	200,000	S2	Yes	2500	16	1/2	5	16 x 25	54	63	82 1/2	61		9"	Green Flintflex	8 x 12	30	
FBS32250	FBSW32250	250,000	J86	No †	3400	18	1/2	6	20 x 20	61	61	97	63	26	27"	9"	Gray Flintflex	10 x 12	30
FBS32300	FBSW32300	300,000	J86	No †	4000	18	1/2	9	20 x 20	61	61	97	63	26	27"	9"	Gray Flintflex	10 x 12	30
FBS32400		400,000	J86	No †	5000	21	1/2	12	16 x 20	61	67	97	71 1/2	26	27"	10"	Gray Flintflex	12 x 12	32
FBS32475		475,000	J86	No †	6000	21	1	12	16 x 20	61	67	97	71 1/2	26	27"	10"	Gray Flintflex	12 x 16	34
FBS32600		600,000	BS10	No						55	120	91	—		12"	Galvanized	16 x 16	36	
FBS32800		800,000	BS10	No						55	120	91	—		12"	Galvanized	16 x 20	40	
SPECIAL																			

*F indicates "F" Series
B = Blower
R = Round Case
S = Square Case
D = De Luxe Case Finish
W = Washer

†Vestibule Burner Enclosure can be furnished at extra cost
‡C.F.M. for Normal Static Pressure and Fan Speed
§Viscous Type Renewable Filters

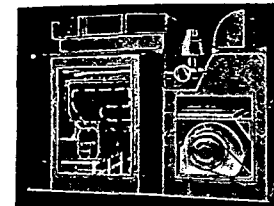


Henry Furnace & Foundry Co.

3471 East 49th Street

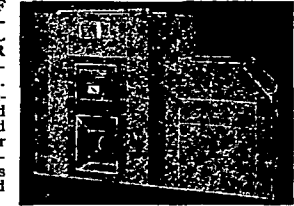
Cleveland, Ohio

MONCRIEF GAS FIRED AIR CONDITIONERS, OIL FIRED AIR CONDITIONERS AND COAL FIRED AIR CONDITIONERS



MONCRIEF GAS FIRED AIR CONDITIONER. Approved by American Gas Association. Burns mixed, artificial or natural gas.

MONCRIEF ARISTOCRAT OIL FIRED AIR CONDITIONER. Specially designed and constructed for oil burner service. Accommodates any standard oil burner.



Heats, cleans, humidifies and distributes the air through forced circulation. Refrigerating and dehumidifying equipment for summer cooling can be readily incorporated if desired.

High efficiency in heating and low operating cost are achieved by means of special construction of burners and heating sections. See table below.

The product of 38 years' experience.

Made in three sizes:

No. 200 delivers at register 118,465 Btu.

No. 240 delivers at register 150,360 Btu.

No. 300 delivers at register 200,475 Btu.

Develops as high as 83 per cent efficiency at bonnet.

Complete catalog and table of dimensions and capacities furnished on request.

MONCRIEF GAS FURNACES FOR AIR CONDITIONING SYSTEMS

Furnace No.	Size of Casings in Inches							No. of Burners	Dia. Flue Pipe	Size Gas Line	Heating Surface	Casing Free Area	Velocity Thru Furnace	Btu Input	Forced Air Btu at Register	C.F.M.	Blower Size	Motor H.P.	No. of Filters	Velocity Thru Filters	Gravity Btu at Register	Gravity Pipe Area
	Type B and Type C Width	Type B Length	Type B Less Blower Length	Type C Less Blower Length	Type C with Blower Length	Type B Height	Type C Height															
50	12	44	..	..	48	..	1	3	3/8	2160	233	...	30,000	...	...	...	...	...	...	...	19,125	170
100	19	44	52	85	51 1/2	54	2	4	1	4320	376	215	60,000	42,750	560	110	1/4	2	105	38,250	281	
150	26	44	52	85	51 1/2	54	3	5	1	6480	438	277	90,000	64,125	840	110	1/4	2	158	57,375	422	
200	34	44	52	88	51 1/2	54	4	6	1	8640	584	277	120,000	85,500	1120	112	1/4	3	140	76,500	562	
250	41	44	52	88	51 1/2	54	5	7	1 1/4	10800	730	277	150,000	106,875	1400	114	1/4	4	131	95,625	703	
300	49	44	52	88	51 1/2	54	6	8	1 1/4	12960	976	248	180,000	128,250	1680	212	1/2	4	158	114,750	844	
350	56	44	52	88	51 1/2	54	7	8	1 1/4	15120	1146	246	210,000	149,625	1960	212	1/2	5	147	133,875	984	
400	64	44	52	88	51 1/2	54	8	9	1 1/4	17280	1168	277	240,000	171,000	2240	214	1/2	5	168	153,000	1124	
450	71	44	52	88	51 1/2	54	9	10	1 1/4	19440	1314	276	270,000	192,250	2520	214	1/2	5	183	172,000	1265	
500	78 1/4	44	52	88	51 1/2	54	10	10	1 1/4	21600	1376	293	300,000	213,750	2800	214	1/2	6	175	191,250	1406	
600	94 1/2	44	52	88	51 1/2	54	12	10	1 1/2	25920	1678	288	360,000	256,500	3360	120	3/4	6	210	229,500	1688	

Type B Furnace Casing has manifold exposed and is designed for gravity use. Blower and filters can be added as separate unit.

Type C Furnace Casing has manifold enclosed and is designed for blower-filter use. Can be used for gravity circulation if desired.

Forced Air Btu Ratings are based on efficiencies of 75 per cent at bonnet and 85 per cent at register. Gravity Btu Ratings are based on efficiencies of 75 per cent at bonnet and 85 per cent at register.

C.F.M. Capacities are based on 70 deg room temperature and 135 deg register temperature.

Add 11 in. to overall height for 14 in. pitched top used with Type B Casing.

See also Page 896

Air Conditioning Systems

Branches

ST. LOUIS
MEMPHIS
OMAHA
MINNEAPOLIS
SALT LAKE CITY
DALLAS

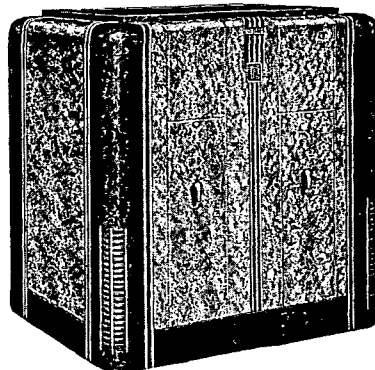
L. J. Mueller Furnace Co.

ESTABLISHED 1857

2009 W. Oklahoma Ave., Milwaukee, Wis.

Branches

LOS ANGELES
NEW YORK
BALTIMORE
PHILADELPHIA
PITTSBURGH
BOSTON

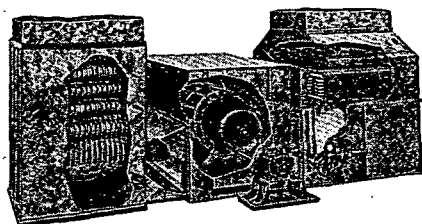


MUELLER OIL-FIRED AIR CONDITIONING UNIT

The first complete departure from conventional furnace design in a direct-fired, forced air heating and air conditioning plant. The basic design, covered by patent claims allowed and pending, secures a new standard of efficiency and reduced fuel cost. The compact simplicity of design, reducing thermal capacity, secures instant heat and eliminates parasitic losses, slow response and temperature override which occur with heavy furnaces. Heats, humidifies, filters, and circulates the air, completely controlling indoor winter climatic conditions, and if desired, cooling and dehumidifying.

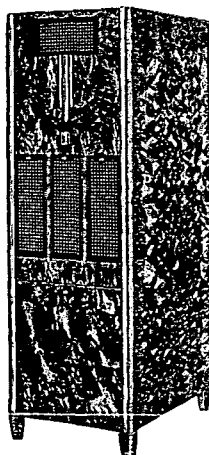
MUELLER CLIMATOR AIR CONDITIONING SYSTEMS

The equipment shown is designed to handle all functions of both winter and summer air conditioning. Heating unit may be any type Mueller furnace for use with coal, oil or gas. Where winter conditioning only is desired, dehumidification and temperature reduction equipment is omitted. Climator units are available in a range of sizes to provide complete or partial conditioning for any size residence.



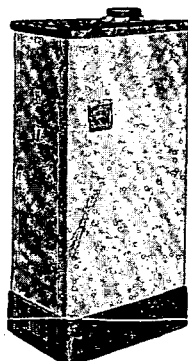
MODERNAIRE UNIT

(At left). For installation in space to be heated. Heats, filters and humidifies the air and distributes by forced circulation. Heat transfer units are Gas Era pressed steel sections. Fan operates automatically when furnace is in operation. Four sizes, with A. G. A. input ratings from 90,000 to 225,000 Btu per hour.



GAS ERA BOILERS

The Series A Gas Era Boiler, at right, is adaptable to steam, hot water or vapor heating in smaller residences. Boilers are of standardized sectional cast iron construction, equipped with completely insulated lacquered casing, concealing sections, controls and diverter. Nine sizes with A. G. A. ratings from 180 to 1,260 sq ft steam, and 290 to 2,015 sq ft hot water.



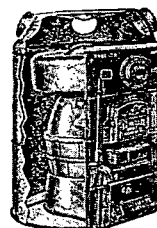
Complete literature on above units furnished upon request

L. J. Mueller Furnace Co.

Air Conditioning Systems and Furnaces

Mueller Heaters For All Fuels—All Purposes

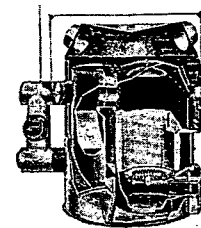
"The Most Complete Line in the Industry"



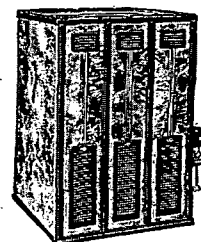
Return Flue all-cast Furnace. 18 in. to 30 in. firepots, single and double fire-door styles. Available in round, galvanized or square, lacquered casings.



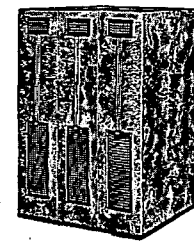
Double Radiator all-cast Furnace. Seven sizes, 20 in. to 33 in. firepot. Vast heating surface. Lacquered, square casing, or round, galvanized iron.



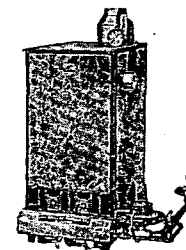
Mueller Steel Furnace. Riveted and welded. Extra heavy construction. Burns any fuel. Seven sizes, 20 in. to 34 in. drums. Square lacquered or round galvanized casing.



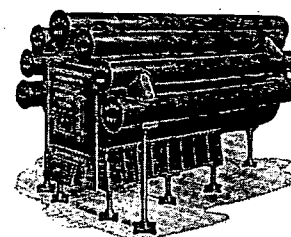
Gas Era cast iron Furnace. Sectional construction. A. G. A. input rating, 65,000 Btu per hour per section. Insulated, lacquered casing for fan or gravity.



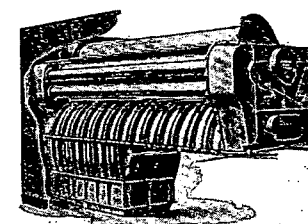
Gas Era pressed steel sectional Furnace. A. G. A. input rating 45,000 Btu per hour per section. Insulated, lacquered casing for fan or gravity operation.



Series C Gas Era Boiler, for steam, hot water or vapor heating or hot water storage. A. G. A. rating, 1,260 to 12,600 sq ft steam; 2,680 to 20,100 water.



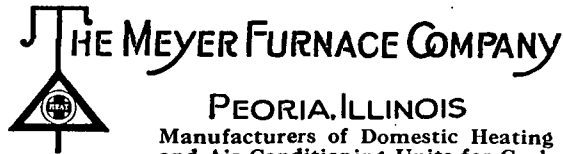
Nos. 80-A and 90-A Horizontal Tubular Heaters for large buildings. Furnished with steel jackets or for brick-setting. For coal, wood or oil burning. Capacities to 750,000 Btu per hour.



Nos. 93, 94 and 95 cast iron Horizontal Tubular Heaters are especially designed for schools, churches and similar large buildings. Capacities from 1,188,000 to 1,390,000 Btu per hour.

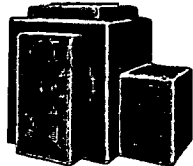
Catalog on Mueller gravity and air conditioning registers and grilles available upon request

Air Conditioning Systems



THE MEYER FURNACE COMPANY
PEORIA, ILLINOIS
 Manufacturers of Domestic Heating
 and Air Conditioning Units for Coal,
 Gas and Oil Burning

Branches and Distributors
 KANSAS CITY, MO.
 OMAHA, NEB.
 GREEN BAY, WIS.
 PITTSBURGH, PA.
 NEW ORLEANS, LA.
 DETROIT, MICH.
 ST. LOUIS, MO.
 SAN FRANCISCO, CALIF.
 DES MOINES, IOWA
 MINNEAPOLIS, M.N.



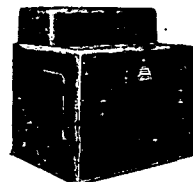
The WEIR Conditioned-Air Unit for coal burning, built around the famous WEIR Steel Furnace, is a complete unit for conditioning the air in the home during the heating season. Equipment includes automatic humidifier, renewable filter, centrifugal blower and automatic damper and blower controls.



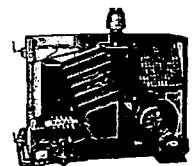
WEIR Conditioned Air Unit

WEIR Gravity Heater

No.	Grate Surface (Sq Ft)	Ratio Htg. to Grate Surface	Smoke Outlet Diam. (In.)	Gravity Circulation		Fan Circulation		
				Casing Dimen.		Rated Output		Air Delivery (CFM)
				Round (In.)	Rect'lar (In.)	At Reg. (Btu./Hour)	Pipe Area (Sq In.)	
621	1.25	41.2	9	48	47x50	54,400	400	1200
624	1.78	33.9	10	52	50x52	73,600	541	1600
628	2.32	29.2	10	54	50x56	94,100	692	2000
630	3.08	25.4	10	58	54x56	119,000	875	2500
633	3.82	22.7	10	65	56x64	138,000	1015	2700
636	4.74	19.4	10	67	56x66	160,000	1180	3000
540	6.25	19.3	12	..	..	..	..	4000
544	7.60	18.5	12	..	..	..	..	5000



WEIR Oil-Fired Air Conditioner



MEYER Gas-Fired Air Conditioner



MEYER Gravity Gas Furnace

The WEIR Oil-Fired Air Conditioner does a complete job of winter air conditioning. Designed for oil fuel and forced circulation. Features include quiet operation, dependability, safety, long life, modern appearance and high efficiency.

The MEYER Gas-Fired Air Conditioner provides complete winter air conditioning. Modern in appearance, compact and efficient with heavy gauge, welded steel, gas-tight heating section. Equipment includes automatic humidifier, renewable filters, centrifugal blower and fully automatic controls.

No.	Input at Burner (Btu/Hour)	Output at Bonnet (Btu/Hour)	Vent Diam. (In.)	Dimensions			Air Delivery 1/4 In. S.P. (CFM)	Motor Size (HP)
				W. (In.)	L. (In.)	H. (In.)		
WEIR Oil-Fired Air Conditioner								
125-A	200,000	150,000	7	48	65	48	1700	1/4
175-A	275,000	205,000	7	56	65	48	2300	1/3
225-A	350,000	265,000	7	63	65	48	3000	1/2
MEYER Gas-Fired Air Conditioner								
B-1	90,000	67,500	4	20	53	42	1000	1/6
B-1 1/2	135,000	101,250	5	27	60	44	1500	1/4
B-2	180,000	135,000	7	40	53	42	2000	1/3
B-3	270,000	202,500	8	60	53	42	3000	1/2
B-4	360,000	270,000	9	80	53	42	4000	3/4
B-5	450,000	337,500	10	100	53	42	5000	3/4
MEYER Gravity Gas Furnace								
C-100	100,000	75,000	5	38	38	60	Complete descriptive literature, including data on Summer cooling, upon request	
C-120	120,000	90,000	6	42	42	67		
C-150	150,000	112,500	6	42	42	67		

The MEYER Gravity Gas Furnace—Efficient—Economical—All steel, welded heating section—large heating surface—A.G.A. approval.

Air Conditioning and Automatic Heating Systems

Norge Division

Borg-Warner Corporation

606-670 E. Woodbridge Street, Detroit, Michigan

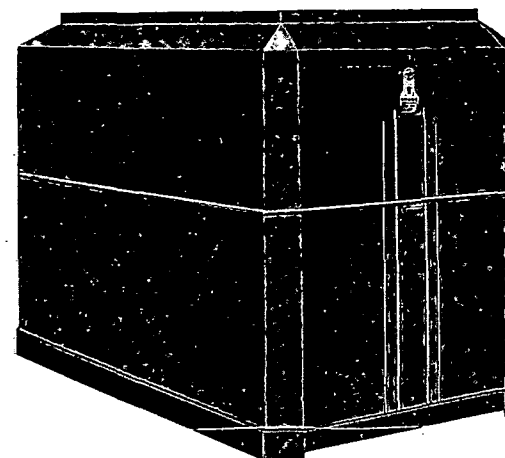
Norge Fine-Air Conditioning Furnace Unit

Newest and most revolutionary Norge development, is the Norge Fine-Air Furnace. It actually warms, humidifies, filters, circulates, purifies the air in every room in the house. Moreover, it provides plenty of hot water during the winter months, with simple adjustment to prevent steam. May easily be converted for cooling and de-humidifying during summer months. And all these advantages cost no more in operating expense than the average ordinary heating system. The Fine-Air Furnace is suitable for new homes, or may be used to replace present warm air heating equipment.

Complete Winter Air Conditioning.

A balanced matched unit with exceptionally high fuel efficiency.

- Fully automatic with oil as fuel.
- 1500-2200 cfm adjustable air delivery.
- Effective radiation surface—104 sq ft.
- Hot gas travel—35 lineal feet.
- Bonnet output up to 200,000 Btu per hour.
- Very fast heat delivery.
- Quick response to temperature controls.
- Fuel efficiencies over 80 per cent.
- Automatic air filter control.
- Correct humidification.
- Adjustable high output domestic water heater.



Oversize deep-blade fan with reserve power simplifies air duct arrangement.

Height 62 in.
 Width 40 in.
 Length 68 in.

Includes Norge Whirlator Burner as standard equipment—manufactured by Norge Heating and Conditioning Division of Borg-Warner Corporation, Detroit, Mich.

See also Pages 830 and 892

Williams Oil-O-Matic Heating Corporation

Manufacturers of Air Conditioning Equipment

Bloomington, Illinois

Service to Architects and Builders

CHICAGO, ILL., 641 N. Michigan Avenue

NEW YORK, N. Y., 1231 Graybar Building

For Williams Oil-O-Matic Oil Burner Equipment and
Ice-O-Matic Refrigeration Equipment, see File Index

Air-O-Matic Combines Heating and Cooling in One Air-Conditioning System

The Williams Oil-O-Matic Heating Corporation has developed an outstanding year 'round air conditioning system known as "Air-O-Matic." Low pressure steam, which is usually provided by an Oil-O-Matic operated boiler unit, is supplied directly to a copper finned heating coil within the central air distributing unit for heating service, which can be supplemented by direct radiation if desired. Proper provision for the addition of moisture is provided for winter heating service.

This same low pressure steam, through an especially developed absorption refrigeration unit, provides the proper degree of temperature and humidity reduction for summer comfort. A change from winter to summer operation can be effected almost instantaneously by means of a master control located in a suitable, convenient place.

The Williams Low Pressure Steam Absorption Type Unit, the outstanding feature of Air-O-Matic, has been especially developed to meet the particular requirements of air conditioning through years of research in the Williams' laboratories. It affords adequate comfort cooling facilities with unusual advantages of economical operation, mechanical simplicity, compactness and freedom from fire and toxicity hazards. Both the solvent and refrigerant are newly developed chemicals and are essentially non-toxic, non-inflammable, non-corrosive to the common metals and chemically stable under all operating conditions.

The steam requirements for one ton of refrigeration are 19 lb. (18,200 B.t.u.'s) per hour at a pressure of 10 to 12 lb. gauge. The electrical power requirements for one ton of refrigeration are 75 to 100 watt hours per hour.

Low Steam and Power Requirements

The low power requirements make possible the use of single phase current in the smaller size installation, thereby saving the expense of providing 3-phase current. For all sizes of machines any low pressure steam boiler of the proper capacity may serve to generate the required steam.

*NOTE.—Any low pressure steam boiler of proper capacity may be used to generate the required steam. A Williams Oil-O-Matic Boiler-Burner Unit or other types of automatic heating systems may be used to maintain the necessary steam pressure automatically.

The only metal to metal contact of moving parts within the refrigeration unit is between the seal face and seal seat of the solution circulating pump. This seal face operates in a bath of oil, thereby assuring proper lubrication. The maximum pressure against this seal face is only 45 lb.

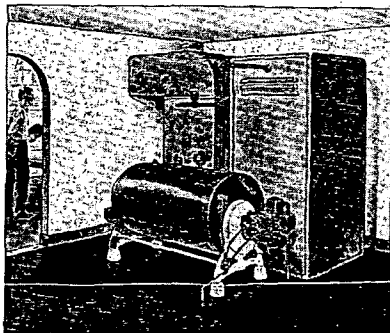
In normal operation the pressure on the low side of the unit is between 8 in. and 2 in. of vacuum and the pressure on the high side is between 22 and 28 lb. gauge.

The steam requirements remain at 19 lb. per ton hour at high or lower cooling water or evaporator temperatures within a reasonable range.

These new absorption refrigeration units are remarkably compact. For example, the 15-ton unit is 44 in. long, 44 in. wide, and 65 in. high, and the 25-ton unit is 65 in. long, 65 in. wide and 65 in. high.

There are now available seven different sized units ranging in capacity from 6 to 25 tons.

The cooling coils of the dry expansion type and the expansion valve are designed for the unique characteristics of the refrigerant.



See also Page 893

The Young Ventilating Company

2700 Woodland Avenue, Cleveland, Ohio

CONTRACTORS - ENGINEERS - MANUFACTURERS

YOUNG REGULATOR

A Convenient Control of Air Conditioning and Ventilation



and presents a neat appearance.

The Young Regulator controls the volume of air flow through a duct by opening and closing a volume damper with which it is connected by a $\frac{3}{8}$ in. square rod. Indicator and dial show accurately position of damper in duct.

For service in schools, hospital, residences, offices, banks and public buildings, the operator in charge of air conditioning and ventilating finds it easy, with this device, to establish and maintain the proper volume of air, free from interference or meddling.

The Young Regulator is simple to install, with direct connection to the damper. It is made of rust-resisting metal

For Locking a Volume Damper in a Permanent Position

With the handy wrench the damper is readily set to admit the desired amount of tempered air, and locked in that position, making it tamper proof.

The device for adjusting is the outermost nut, or head with eight sides, turned by wrench.

Behind the adjusting nut is a set nut with ten sides, of larger diameter, which, with a turn of the wrench, will lock the regulator at any desired position, or release it. This nut is difficult of access by any ordinary wrench.

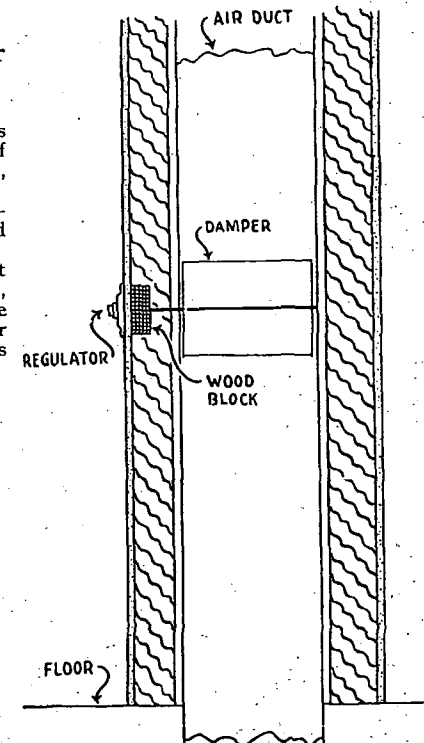
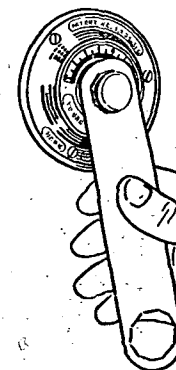


Diagram Showing Vent and Supply Ducts, in which
Damper Controls are Installed

AMERICAN AIR FILTER COMPANY INC.

1st Street and Central Avenue, Louisville, Ky.

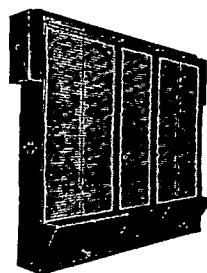
Representatives in Principal Cities

Dust Engineering—Dust Engineering is that branch of applied science which deals with the origin, nature and characteristics of the small solid air-borne particles called "dust," and the development of methods, processes and apparatus for its control or elimination.

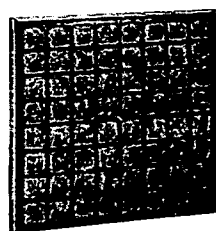
The American Air Filter Company, Inc., has had an important part in advancing the science of Dust Engineering. The efforts of its Research and Engineering Staff for the past twelve years have been devoted exclusively to the study of dust problems and the development of a complete line of air cleaning equipment for modern air conditioning, building ventilation and the control of process dust in industry.

American Air Filter products, therefore, not only embody the knowledge accumulated from years of constant research and the experience gained from designing, building and applying thousands of air filters, but are backed by ample technical and financial resources to insure their outstanding position in the Dust Engineering field.

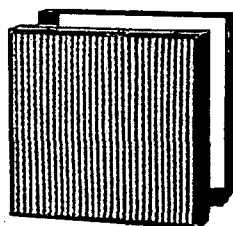
Products—American Air Filters are available for every condition, with operating characteristics and efficiencies to suit specific problems. In



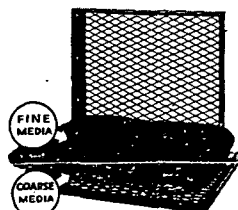
Automatic Multi-Panel Filter



Throway Air Filter



Airmat Drifilter Type K



Re-nu-Vent Filter

general, there are two distinct types based upon the "viscous film" and "dry mat" principles. Each type is made in several styles which differ in method of operation, servicing, space required and initial cost to meet the various conditions encountered in air cleaning problems. A discussion of various filter types will be found in the Technical

Data Section under "Air Cleaners."

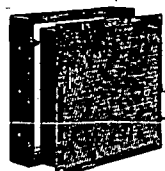
Air filters are generally used for the removal of dust, dirt, bacteria and other foreign matter from the air and are applied to general ventilation, modern air conditioning, process dust control; for air compressors and Diesel Engines; mill motors, turbo-generators and other electrical applications; and for air or gas under pressure to remove entrained oil, moisture and dirt.

Air Filters in Air Conditioning—Filtered air is today recognized as essential in modern air conditioning. There are other important factors which contribute to our comfort such as temperature, air movement and humidity, but science today emphasizes the prime necessity of pure air for health and efficiency.

Air cleaners have, of course, always been considered an integral part of large central systems. These are usually of the fully automatic type such as the Multi-Panel filter, illustrated in the accompanying photograph.

There are now available to manufacturers of unit air conditioners moderate priced unit filters, such as the Re-Nu filter, the Drifilter and the Throway filter, illustrated herewith.

The Re-Nu filter is an entirely new departure in air filter construction. It consists of a permanent metal frame provided with



Standard Viscous Unit Filter

a removable cover and renewable filter pad. The cover is easily removed without the use of tools, and filter pad can be lifted out and replaced with a new one at very small expense in less than a minute's time.

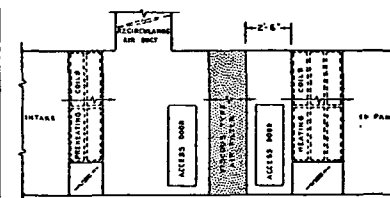
The Drifilter consists of the Airmat filtering media mounted on a supporting member and arranged in saw-tooth fashion, as illustrated. The filter media is reasonable in price and can be easily replaced when desired.

The Throway filter, as the name implies, is an inexpensively constructed unit designed to be discarded after it has served its maximum period of usefulness and replaced with a new filter unit. The filter pad is enclosed in a perforated cardboard container which makes it possible to dispose of the dirty filter by burning it in a furnace or incinerator.

There is probably no single item which costs as little and may mean as much in the design of an air conditioner as air filtration. These units are furnished in any dimensions or shapes desired. They are usually built in units handling 400 c.f.m. and from 2 in. to 4 in. thick. They are usually made in the following sizes—20 x 20 in., 16 x 25 in. and 16 x 20 in. High cleaning efficiencies can be secured, with a resistance to air flow ranging from $\frac{1}{16}$ in. to $\frac{3}{8}$ in. water gauge.

Automatic Self-Cleaning Air Filters—The American line of automatic air filters is among the most complete that has ever been offered, the most popular types being Multi-Panel, Horizontal and Phoenix Filters.

All types are furnished for either continuous or intermittent service and are available in sizes and set-ups suitable to any desired capacity or space condition.



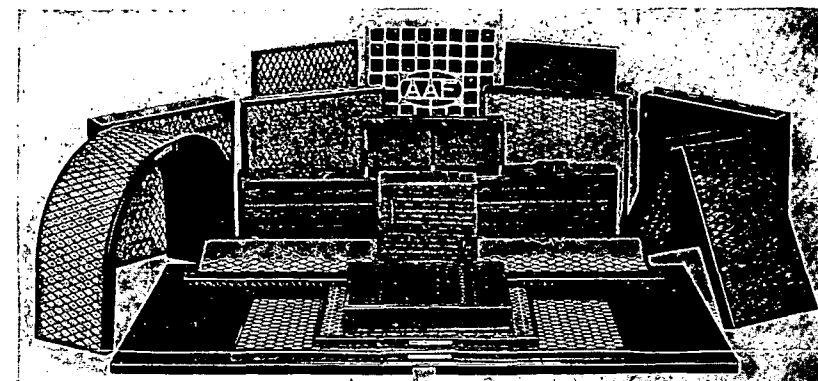
Typical Installation

Standard Viscous Unit—The American Unit Air Filter incorporates the time tested unit principle of construction. Each unit consists of a standard steel frame and interchangeable cell equipped with automatic latches to facilitate removal for cleaning and recharging.

Airmat Filter Dry Type—The filtering media in this type is the Airmat sheet, a dry filter mat composed of thin sheets of gauzy, cellulose tissue. The Airmat sheets are supported in screen pockets mounted in a unit frame of box-like construction. These unit frames can be set up to meet any capacity requirement or space condition. The Airmat sheets are renewable—their life depending on the dust condition and hours of daily service.

Airmat filters are used both for air conditioning and industrial air conditioning. In the latter field they are particularly well adapted for the recovery of valuable dusts and for abating the dust nuisance which confronts so many industrial plants.

Our standard data books and catalogues are to be found in most engineering files or libraries. We will be glad to furnish full data to engineers or manufacturers interested in this subject.



Various Types of Unit Air Filters for Air Conditioning Work

Coppus Engineering Corporation

339 Park Avenue, Worcester, Mass.

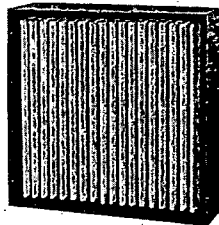
MANUFACTURERS OF AIR FILTERS, STEAM TURBINES, FORCED DRAFT BLOWERS, COOLING FANS

Coppus Unit Filter

Coppus Unit Filter—The Coppus Unit Filter is of the dry type using a removable filter glove of all-wool felt and supported by a rigid, welded distender frame which is adjustable so that after assembly the filter glove may be stretched and held tautly inside of the filter casing, giving the pockets a tapered shape so essential for an even air flow. All metallic parts are rust-proofed (Bonderized, Cadmium Plated), and Duco painted.

Specifications

Normal Rating: 800 cfm.
Resistance when clean: .2 in. W.G.
Dust Arrestance (cleaning efficiency): 99.61 per cent (Tested in accordance with A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work).
Dimensions: 20 in. by 20 in. by 6½ in.
Weight per unit: 25 lbs.



Air Intake Side of Stationary Unit Filter

Outstanding Advantages

1. It has an exceptionally high dust arrestance.
2. It maintains a high dust arrestance even under diverse conditions of neglect.
3. Its operation is not impaired by atmospheric conditions.
4. It is a Medium Air Resistance Type (Class C) according to the A.S.H.V.E. Code for Air Cleaning Devices.
5. It is easily and quickly cleaned without removing the filter element.
6. Its cost of upkeep is very low because the permanent filter element is reconditioned periodically with a vacuum cleaner.
7. It combines scientific knowledge and practical engineering methods with highest quality of material and workmanship.

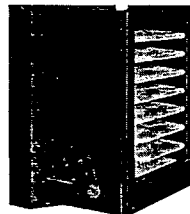


Cleaning Filter Elements with Portable Vacuum Cleaner

Coppus Dry-Matic Filter

Automatic Self-Cleaning Dry-Type Air Filter

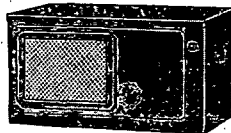
—It uses a specially woven cotton textile as filter medium arranged in the shape of an endless belt in zig-zag fashion over rolls. The horizontal pockets have a tapered shape, thus securing an even rate of air flow over the whole effective filter area. At predetermined intervals the filter curtain is automatically moved by a small geared motor over the rolls, thus passing over a suction nozzle. For certain types of dust the filter is also equipped with a beater which, at predetermined intervals, automatically reconditions the filter curtain. The cleaning and reconditioning operation



takes place while the ventilating system is in operation. Built in capacities from 2000 cfm up.

Coppus Window Air Filter

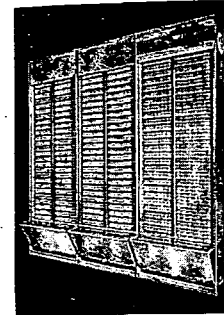
It supplies a continuous flow of filtered air, is extremely quiet in operation, keeps out street noises, eliminates dirt and dust including invisible particles, and last, but not least, is very effective against rag weed pollen in concentrations commonly found in the hay fever season. Its use is recommended for offices, homes and hospitals.



Air Filters for Compressors and Internal Combustion Engines.
Steam Turbines, Horizontal and Vertical, 1 to 150 Hp.
Forced Draft Blowers.
Portable and Cooling Ventilating Fans.

Independent Air Filter Co.

215 West Ohio Street, Chicago

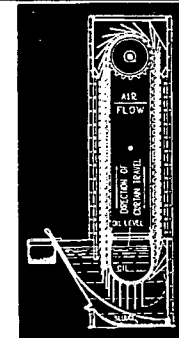


Three Section Group, 9 ft. 3 in. high by 9 ft. 0 in. wide, capacity 30,000 c.f.m. "Double Duty" filters made in capacities from 3000 to 600,000 c.f.m.

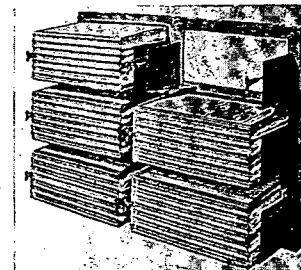
"DOUBLE DUTY"

An Air Cleaner That Cleans Itself—and Never Clogs

This type especially adapted for large installations, where automatic operation and unvarying air flow are important factors. The filtering principle is that of true impingement on viscous coated metal surfaces. Endless filter curtain composed of die stamped steel plates, heavily indented, and disposed in overlapping form. Acute deflections of individual fins, constantly coated with heavy oil, result in high cleaning efficiency. Entirely self-cleaning, regardless of dust content of air. Self-draining design of curtain plates eliminates all danger of oil entrainment.



Cross-section through filter. Sludge is removed at intervals of two to six months. Oil consumption negligible—never changed; merely replenished

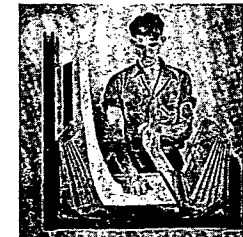


Note easy method of detaching unit from frame. Improved style locking device clamps felt seal firmly against main frame

"KOMPAK"

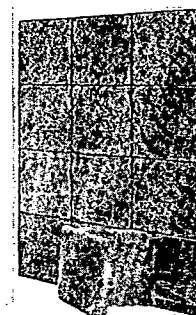
A Low Velocity, Dry Fabric Air Filter—With Renewable Medium

For general ventilation and industrial air cleaning. Filtering principle that of straining air through fabric material at low velocity. Standard filter medium high grade, fluffy cotton material, with high cleaning efficiency and remarkable dust holding capacity. Each unit contains 27 sq. ft. of filter medium, which, at normal rating of 600 c.f.m., results in velocity of 22½ f.p.m. through medium. Initial resistance 0.15 in. W. G.



METHOD OF SERVING

Entire operation—unloading, refilling and replacing—consumes at most 3 minutes per unit. Useful life of filter medium varies from three to six months, depending on conditions

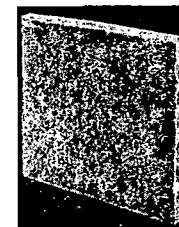


Frame installation. No pad container necessary. Only the pad is thrown away

HAIR PAD

A High Grade, Low Cost Air Filter

Filter medium composed of Keratin fibre (animal hair) thoroughly sterilized, making it sanitary and odorless. Fabricated with progressive density into a self-supporting pad of great resiliency. Oil treated to increase cleaning efficiency and dust holding capacity. For ventilating and air conditioning systems frames are made standard size 20 in. x 20 in. x 4½ in. (overall) rated at 800 c.f.m. per unit. Initial resistance with pads in double formation 0.12 W. G.

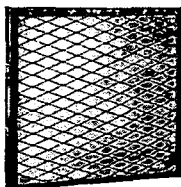


For warm air heating systems and air conditioning units pads are made in three standard sizes, 20 x 20, 16 x 25 and 16 x 20. Special sizes, any shape, cut to order

INDUSTRIAL MATERIALS DIVISION
Owens-Illinois Glass Company
 MANUFACTURERS OF DUSTOP AIR FILTERS
 Newark, Ohio

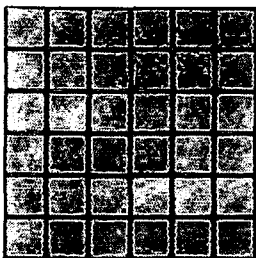
DUSTOP

The Original Replacement Type "Dustop"—the Glass Wool Air Filter of High Efficiency and Low Cost for Use in All Types of Ventilating Systems or Wherever Air is Mechanically Circulated



Dustop—The Owens-Illinois fibrous glass air filter is the modern answer to the demand for low cost, efficient air filtration. Dustop filters can be assembled in any required number of units to clean air in every type of commercial and industrial building. Dustop maintains a high efficiency in removing dust, dirt, lint, pollen, bacteria and other harmful impurities from circulated air.

Dustop is simple in design and operation—progressively packed—ready for use—easily disposed of when dirty—3 to 6 months between changes—efficient—odorless—light in weight. Because of its many outstanding features Dustop is one of the most desirable filters on the market.



Overall Size of Frame Unit

(One complete "L" or 800 c.f.m.)
 Width, 20 in. Height, 20 in. Depth,
 4 3/4 in. Uses standard 20 x 20 x 2
 Dustop Filter



Overall Size of Frame Unit

(One complete "V" or 2000 c.f.m.)
 Width, 30 in. Height, 20 in. Depth,
 27 in. Uses standard 20 x 25 x 2
 Dustop Filter

FILTERS INSTALLED in TWO Arrangements

The "L" Type for Plane Bank Arrangement Rating

Capacity per unit, 800 c.f.m.
 Maximum velocity, 300 f.p.m.
 Resistance at rated capacity (two filters in tandem),
 Clean, .25 in. w.g.; Dirty, .38 in. w.g.

The "V" Type for Limited Space Rating

Capacity per unit, 1000 c.f.m.
 Maximum velocity, 300 f.p.m.
 Resistance at rated capacity (two filters in tandem),
 Clean, .25 in. w.g.; Dirty, .38 in. w.g.

FRAME CONSTRUCTION FOR ANY C.F.M. CAPACITY

Sold by fan, blower, unit and air conditioning manufacturers and our authorized distributors—everywhere.
 Replacements—Stocked by our authorized distributors in all trading centers.

Somers Air Filter Sales Co.

7310 Woodward, Detroit, Mich.
SOMERS "HAIR GLASS" FILTER (Patented)



Fig. 1—Flat type with deep frame



Fig. 2—Flat type with flat frame



Fig. 3—"V" type with deep frame

The Somers Filter Mat requires no adhesive material to catch the dust particles of the air stream in passing between the fibers. Its construction is such that it forms an effective dust barrier that is highly efficient, the range depending on the type of pack, air stream velocity and the character of the dust. Cleaning the filter is easy and it is a most desirable construction for use where operating velocities or temperatures are relatively high or where a high degree of air purity must be secured.

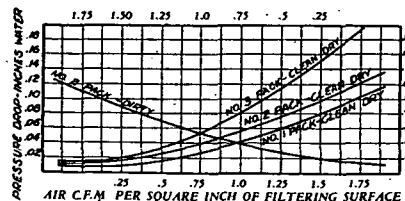
"Hair Glass" in this form reacts with practically none of the chemical elements carried in the air stream. It is easy to maintain in proper functioning condition as "Hair Glass" neither rots nor disintegrates in service. "Hair Glass" is pliable and tough, it is both odorless and non-absorptive. Consequently, the user has wide latitude in his methods of handling and cleaning the unit.

The Somers Filter does not have to be replaced as soon as it fills up with dust. The entire pack may be washed with hot water, cold water, or chemical solvents and as often as necessary, without in the least impairing its operating efficiency.

Standard frames and mesh are galvanized or alloy although other materials can be furnished if required.

The Somers filter may be assembled in different thicknesses:

No. 1 Pack—The thinnest, is used where the resistance must be kept low. It is usually employed for domestic applica-

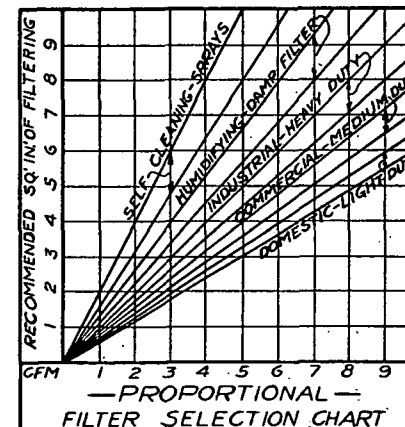


tions, recirculating systems and gravity flow.

No. 2 Pack—Is generally used for all commercial and industrial applications. It will arrest liquid particles in the air stream.

Filter units can be made to exact dimensions.

Bank holding frames can be supplied. Give outside dimensions.



CODE LETTER	FRAME TYPE	FILTER FRAME DIMENSIONS				AREA IN INCHES			FILTER ONLY		FILTER TYPE	FIGURE NO.
		HEIGHT	NOTE	WIDTH	DEPTH	PROJECTED	RATIO VEE TO ONE	FILTERING MEDIUM	WEIGHT IN POUNDS			
A	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
B	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
J	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
K	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
L	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
M	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
N	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
O	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
P	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
Q	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
R	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
S	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
T	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
U	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
V	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
W	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
X	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
Y	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
Z	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AA	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AB	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AC	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AD	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AE	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AF	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AG	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AH	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AI	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AJ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AK	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AL	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AM	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AN	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AO	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AP	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AQ	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AR	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AS	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AT	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AU	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AV	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AW	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AX	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
AY	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
AZ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BA	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BB	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BC	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BD	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BE	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BF	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BG	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BH	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BI	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BJ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BK	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BL	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BM	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BN	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BO	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BP	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BQ	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BR	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BS	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BT	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BU	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BV	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BW	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BX	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
BY	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
BZ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CA	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CB	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CC	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CD	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CE	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CF	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CG	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CH	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CI	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CJ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CK	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CL	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CM	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CN	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CO	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CP	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CQ	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CR	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CS	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CT	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CU	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CV	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CW	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CX	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
CY	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
CZ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DA	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DB	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DC	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DD	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DE	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DF	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DG	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DH	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DI	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DJ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DK	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DL	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DM	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DN	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DO	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DP	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DQ	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DR	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DS	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DT	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DU	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DV	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DW	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DX	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
DY	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
DZ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
EA	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
EB	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
EC	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
ED	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
EE	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
EF	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
EG	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
EH	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
EI	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
EJ	FLAT	20		20	4	400	1:1	400	3.25		FLAT	2
EK	FLAT	16		25	4	400	1:1	400	3.25		FLAT	2
EL	FLAT	20		20	4	400	1:1	40				

Staynew Filter Corporation

Air Filters for Buildings and Mechanical Equipment
6 Leighton Avenue, Rochester, N. Y.

Products—

Protectomotor Dry Type Positive Filters for removing dust, dirt and foreign matter from small or large volumes of air at atmospheric or at higher or lower pressures. Made in various types and sizes for buildings, windows, oxygen chamber in hospitals, furnaces, pipe lines, air compressors, diesel engines, blowers, motors, pneumatic systems, air brakes, etc.

Operation—

The efficiency of Protectomotor Air Filters is exceptionally high due to their large filtering surface within a relatively small space. The intake air currents move parallel to the filtering surface at very low velocity, so that dust and dirt are not packed on to the felt, but remain in a loose and porous condition, which permits air to pass through the accumulated dust and dirt quite as readily as through the felt itself. Dust and dirt do not enter pores of the extremely fine texture felt.

Efficiency—

Due to the low air velocity and the fine texture of the felt, practically complete removal of dust is obtained. An exceptionally high efficiency is maintained, even on fine air-floated dust. Efficiency is not greatly affected by continuous service, nor by any change in volume of air passed.

Pressure Drop—

Resistance to the flow of air is less than $\frac{1}{4}$ in. water gauge when operated at rated capacity. Less pressure drop, when required, may be obtained by using an oversize filter.

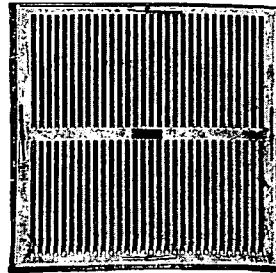
Cleaning—

Under ordinary conditions filters operate from six months to a year without attention. A special cleaning device enables the material collected on the filtering surface to be very quickly and completely removed by compressed air or vacuum cleaner. The material may be reclaimed, if valuable, without contamination, since no sticky or adhesive oils are used. No spare parts or cleaning tanks are necessary. Less than two minutes' time, per 1000 cu. ft. per minute of air capacity, required for cleaning. Filters may be cleaned while in operation, without removing filtering units.

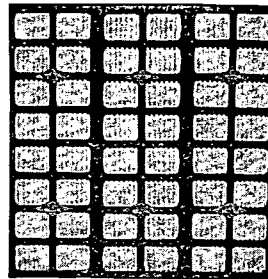
Application—

All filter units or filter assemblies are complete in weatherproof housing, ready to attach to air intake pipe for application to engines, compressors, etc. For building ventilation, equipment is easily adapted to space available.

Write for Catalog.



Protectomotor Building Filter



Multi-V-Type Building Filter



Protectosent Window Ventilator
One filter cell removed, exposing one of the fans and motors



Diagonal view showing adjustable ends storm hood, filter and deflector plate. Note the large, active surface

The Vinco Company, Inc.

305 East 45th Street

New York, N. Y.

VINCO BOILER CLEANSER

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.



Vinco for Old Systems

Annual cleaning of the old heating system adds years of life to the boiler, prevents rust deterioration and saves much fuel and fire attendance. Only half quantities given in specification table.

Our Three-Fold Guarantee

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients. Leading boiler manufacturers have placed tags on their boilers advising against use of acids or alkalis.
2. Purchase price is refunded if results are not as claimed when VINCO has been used according to directions.
3. Your time, money and comfort are further safeguarded by

Our Free Laboratory Service

Saves thousands of dollars by analyzing the boiler water before making needless mechanical changes. If desired, the boiler water is again examined after the VINCO treatment is completed, and "certified chemically correct" for boiler operation. (Write for details.)

VINCO SUPERFINE LIQUID BOILER SEAL

A different liquid leak 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.

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.

VINCO SOOT-OFF

Destroys soot from coal, oil or gas burning heating equipment. Easier, cleaner and quicker than brushing. Makes short work of jobs too hard for brush or scraper. Cleans fire pot, flues and chimney in one simple operation. Non-explosive, thoroughly safe.

Adopted By

American Radiator Co.
Burnham Boiler Corp.
Hoffman Specialty Co.
National Radiator Corp.
Petroleum Heat & Power Co.
Sarco Co.
Titusville Iron Works Co.
United States Radiator Corp.
and many others
*Vinco Distributors

What Vinco Does

VINCO permanently removes oil, grease, scale, rust and dirt from the internal surfaces and from the boiler water without the labor of blowing boilers over the top. By this thorough cleansing Vinco stops foaming, priming, surging, and slow steaming.

How Vinco Works

Each minute grain of VINCO powder absorbs several times its own weight of oil, rust and dirt. These larger grains of absorbed impurities then settle and are blown through the bottom, according to directions on each can.

Vinco Specifications for New and Remodeled Steam and Vapor Systems

Cleaning the System—Upon completion of the installation, the contractor shall clean the system by the Vinco method to remove oil, grease, rust and dirt from the boiler using, 1— lb. of Vinco, in exact accordance with manufacturer's directions.

This compound must remain in the boiler for at least 10 days of normal operation—not longer than 60 days. At the end of this period, boiler must be thoroughly drained and flushed before refilling with clean water.

In writing specification, insert in this space number of pounds of Vinco to be used in accordance with the following schedule. For systems having actual installed radiation of:

Up to 350 sq. ft.	3 lb.
351 " 600 "	5 "
601 " 1100 "	8 "
1101 " 1400 "	10 "
1401 " 1800 "	13 "
1801 " 2100 "	15 "
2101 " 2700 "	18 "
2701 " 3100 "	20 "
3101 " 3700 "	23 "
3701 " 4200 "	26 "
4201 " 4600 "	28 "
4601 " 5000 "	30 "

Above 5000 sq ft use an additional pound of Vinco for each additional 300 sq ft of installed radiation.

Vinco Specifications for Hot Water Systems
Cleaning the System—Same as above but only one-half quantities in specification table required.

MCDONNELL & MILLER

Manufacturers of McDONNELL Boiler Water Level CONTROLS

General Offices: Wrigley Building, Chicago, Ill.

*"Doing one thing well"***BASIC ADVANTAGES:**

McDonnell Boiler Water Level Controls protect low pressure steam boilers from the damage and shut-downs caused by low water. In connection with automatically fired boilers they "make the boiler water level as automatic as the firing." (Recommended by A.S.M.E. Boiler Code, Underwriter's Laboratories, Boiler Insurance Companies—Required by ordinances in over 20 cities.)

FEATURES OF CONSTRUCTION:

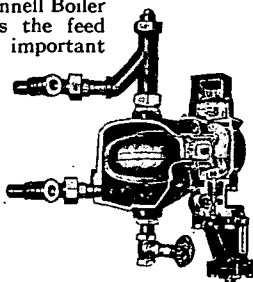
In all McDonnell Boiler Water Feeders the feed valve and all important working parts are isolated from the heat of the float chamber. As a result, scale and lime does not form at the feed valve.

(Covered by Patent No. 1,934,486.)

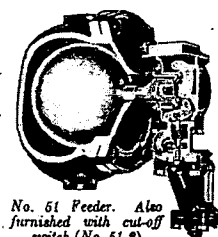
No. 47 Water Feeder and the No. 60-B Low Water Cut-off have "Quick-Hook-Up Feature" which permits installation in water glass tapings and assures correct reproduction of the water level. (Covered by Patent No. 1,997,785.)

All McDonnell Water Feeders have stainless steel valves, straight thrust valve action, double sylphon (packless) construction, large, easily removed integral strainer, and many other exclusive features. All cut-offs and feeder-cut-off combinations are equipped with the effective McDonnell quick-acting wiping-contact switch with low water alarm terminals as standard.

To determine type of equipment suited to specific conditions, consult the tables:



No. 47-2 Combined Feeder and Cut-off with Quick-Hook-Up Feature. Also furnished without cut-off switch (No. 47)



No. 61 Feeder. Also furnished with cut-off switch (No. 61-S)

(Table 1) For Hand Fired Jobs

Boiler Size in Square Feet	Steam Pressure	Product to Use
Up to 5,000	Under 25 lb	No. 47 Water Feeder
Above 5,000	Under 25 lb	No. 51 Water Feeder
Any Size	25 lb to 75 lb	No. 53 Water Feeder

(Table 2) For Automatic Jobs Where Completely Automatic Control is Desired*

Boiler Size in Square Feet	Steam Pressure	Products to Use
Up to 5,000	Under 25 lb	No. 47-2 Combined Water Feeder and Low Water Cut-off.
Above 5,000	Under 25 lb	No. 51-2 Combined Water Feeder and Low Water Cut-off.
Any Size	25 lb to 75 lb	No. 53-2 Combined Water Feeder and Cut-off.

(Table 3) For Automatic Jobs Where Only a Low Water Cut-off is Desired

Boiler Size in Square Feet	Steam Pressure	Product to Use
Any Size	Under 25 lb	No. 60-A—for 1 in. equalizing pipe hook-up. No. 60-B with quick hook-up feature for all round boilers and small sectional boilers.
Any Size	25 lb to 100 lb	No. 32 H. P. (Also used as a pump control on boilers of 100-lb maximum).

*The combined feeders and cut-offs (Nos. 47-2, 51-2 and 53-2) offer both protection and the convenience of completely automatic operation for oil or stoker-fired boilers. If, however, price is the dominant factor, the No. 60-A, No. 60-B, or No. 32-H. P. Cut-offs will dependably interrupt the current to the firing means when the water level falls to the danger point.

MCDONNELL & MILLER

Manufacturers of McDONNELL Boiler Water Level CONTROLS

General Offices: Wrigley Building, Chicago, Ill.

*"Doing one thing well"***TYPICAL SPECIFICATIONS FOR WATER FEEDER OR COMBINED FEEDER AND CUT-OFF**

Contractor to furnish and install complete in every essential detail for each boiler unit, Automatic Boiler Water Feeder—and Low Water Cut-off (if automatically fired)—equipment as manufactured by McDonnell & Miller, Wrigley Building, Chicago.

Boiler Water Feeder to be (Insert**) of the manufacturer's most improved design, with all working parts isolated from the steam and hot water zone by packless sylphon construction. Float power to be multiplied through a leverage mechanism which contains a self-centering roller directly above the valve stem. All bearing points, pivots, and levers, to be outside of the steam and water zone. Valve cone and seat to be stainless steel, removable and renewable. A strainer to be incorporated in the design and to have a solderless basket mounted on a flange for easy removal. Feeder to be installed complete with all piping, valves, fittings, and specialties, as indicated by descriptive diagram in the manufacturer's construction bulletin.

Upon completion of the installation, the contractor to place this automatic equipment in successful operation, subject to acceptance and approval of the architect's engineer.

**If job is hand fired, insert: "No. 47," "No. 51," or "No. 53." (See Table 1).

**If job is automatically fired, insert: "No. 47-2," "No. 51-2," or "No. 53-2." (See Table 2).

When combined units No. 47-2, No. 51-2, or No. 53-2 are specified, add the following paragraph:

"Make electrical connections for low water switch to control fuel supply equipment, so that the fuel is automatically shut off by the low boiler-water control. Control wiring to be of flexible armored cable. All wiring to meet the requirements of the City Electrical Inspection

Department and the National Board of Fire Underwriters."

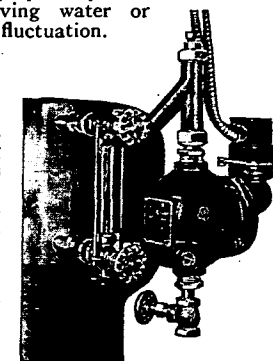
TYPICAL SPECIFICATIONS FOR LOW WATER FUEL CUT-OFF

Furnish and install on each boiler; in accordance with the manufacturer's instructions a McDONNELL & MILLER No. (insert†) Low Water Fuel Cut-off, to be of packless construction, with all working parts outside the steam and water zone. Switch to have snap, wiping-action contacts. Control wiring to be of flexible armored cable. All wiring to meet the requirements of the City Electrical Inspection Department and the National Board of Fire Underwriters.

†Insert at this point: "No. 60-A," "No. 60-B," or "No. 32-H.P." as indicated by service conditions per Table 3.

Ask for information covering special conditions such as application of water level control to higher pressure boilers in which steam pressure is in excess of water pressure. Our engineering department is qualified by widely diversified experience to solve any special problems which may arise involving water or liquid level fluctuation.

The McDonnell catalog contains complete descriptions, installation instructions, dimensions, wiring diagrams, capacity curves, and other valuable data.



No. 60-B Low Water Cut-off with Quick-Hook-Up Feature. Also furnished without Quick-Hook-Up. Parts for installation with 1-inch equalizing pipe. (No. 60-A)

Aquatic Chemical & Metallurgical Engineers

118 East 28th Street New York, N. Y.

An Organization of Water Treatment Experts

Manufacturers of Scientific
Products for Eliminating



Rust, Scale, Foaming and
Related Water Troubles

DEVELOPMENT—A-C-M-E laboratories, having surveyed many different waters and types of heating installations, are bringing to your attention a series of practical and accurate water treatments for house heating boilers.

Long technical experience with water problems in the marine and stationary high pressure boiler field, together with careful diagnosis and follow-up, enables us to bring the low pressure boiler plant within our scientific control.

TREATMENTS AND THEIR ACTION—Our materials are distributed in convenient quart containers ready for use. They act upon accumulated rust, scale and oils at a proper slow rate to *completely disintegrate and loosen them, without plugging and stopping steam and water flow—by holding the resulting sludge in suspension until blown out.* The treatments acting thru their colloidal, chemical and electro-chemical properties prevent the further formation of rust and scale, and stop frothing.

RESULTS—The heat retarding action of scale, rust and oil is eliminated; accumulations of mud and core-sand are swept from the system; foaming and dirty water prevented and corrosion overcome. As a result, steaming is easier, a surprisingly large amount of fuel is saved, repairs and replacements are minimized and the entire heating system made more efficient. **Because heating surfaces are actually cleaned, heat transfer is quicker.** Moreover, deposits of rust are removed from such places as hot water heating units, valves and other auxiliaries. Continued treatment prevents further accumulations. Since corrosion is arrested, boilers do not rust and leak.

UNIQUENESS OF A-C-M-E SERVICE AND TREATMENTS—For the first time, instead of buying a single preparation for all boilers and all troubles, you can purchase a different and scientifically prepared product depending upon whether yours is a cast-iron boiler or steel boiler, and whether it is coal, oil or gas-fired. Ask for No. 1L.P. folder.

Quantities Required for one complete cleaning (two treatments) of house heating steam boilers:

Cast-Iron: Coal or Oil-Fired—For each 6 sq. ft. or less of grate area, use two cans.

Steel: Coal or Oil-Fired—For each 150 sq. ft. or less of boiler heating surface, use two cans.

All Gas-Fired Steam Boilers—Two cans for the first and second 2000 sq. ft. of steam radiation, and two cans for each additional 1500 sq. ft. thereafter.

Hot Water Heating—For each 2500 sq. ft. or less of hot water radiation, two cans "Gas-Fired Treatment." Above quantities are based on actual extensive field experience under average conditions. Our laboratories will be glad to advise in unusual cases.

AIR CONDITIONING SYSTEMS, SPRAY TOWERS AND WATER SERVICE LINES can also be saved from the corrosive and destructive action of water and the impurities it collects. Recommendation without charge. Information upon request.

RECOMMENDED AND USED BY LEADING HEATING AND VENTILATING ENGINEERS

Scale which
acts as insula-
tion can be re-
moved safely.



Users are Impressed by Positive Results

Corrosion
deteriorates
equipment
quickly. Arrest
it promptly!

AMERICAN RADIATOR COMPANY

40 West 40th Street, New York, N. Y.

Division of AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

Products for Every Heating Requirement

On Automatic Heating

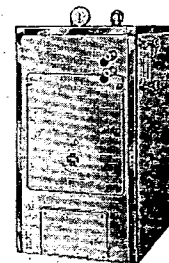
AMERICAN Radiator Company, the world's largest manufacturer of heating equipment, includes in its line every product necessary for complete automatic heating, excepting oil burners and mechanical stokers. This includes not only boilers specially designed for oil, gas or coal, but radiators and controls as well; and such accessories to the complete job as radiator enclosures, vents and valves, copper pipe and fittings, time and limit switches, and equipment for domestic hot water.

The use of this family of products assures you undivided responsibility for the satisfactory performance of the complete job. All of them are backed by the same dependability, the same experience, the same reputation for performance... and the name that for more than 43 years has stood for quality in heating—American Radiator.

An important feature of the boilers and the radiators listed here is that they are all cast-iron—the metal that scientific research and tests indicate is best for heating purposes. Especially for automatic heating—which is in most cases intermittent—cast-iron radiators have advantages possessed by no other. They will stand more than the normal abuse of installation and use without damage, and have the ability to hold heat longer between periods of firing, and are easier to keep clean.

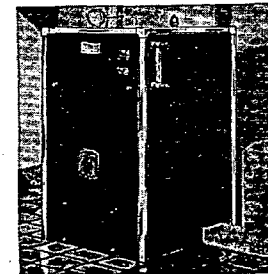
On the following pages are shown a few of the products of American Radiator Company. For detailed specifications or information on any of them, or on any other equipment for heating, see the Ideal Fitter, or write to American Radiator Company, 40 West 40th Street, New York, N. Y.

BOILERS for automatic heating with every fuel



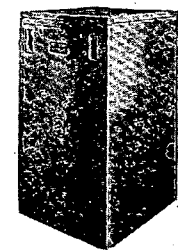
No. 12 OIL BURNING
BOILER

A deluxe model, exclusively for oil burning. Designed with the cooperation of the technical committee of the American Oil Burner Association. Jacket concealing Burner and controls can be furnished.



No. 21 AUTOMATIC
COAL-FIRED BOILER

Designed especially for use with mechanical stokers and recommended by leading stoker unit manufacturers, the No. 21 Boiler assures the best in coal fired automatic heat.



No. 11 OIL BURNING
BOILER

One of the first and finest exclusively oil burning boilers for the small home. Equipment includes controls for complete automatic requirements.

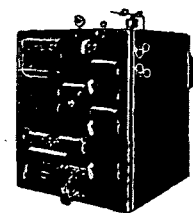
See also Page 861

MAKE IT AN "ALL AMERICAN" JOB!

AMERICAN RADIATOR COMPANY

PRODUCTS FOR EVERY HEATING REQUIREMENT

MAGAZINE BOILER No. 25

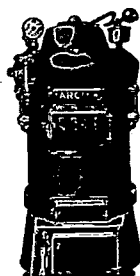


This coal boiler is self-feeding. As the fuel on the grate burns, additional fuel comes down from the large magazine. There are no moving parts or accessories. Fuel is fed by gravity

at a uniform rate and as needed. Unique by-pass directly to chimney provides for quick starting. **Ratings:** Steam, 300 to 2400 sq ft; Water, 480 to 3840 sq ft.

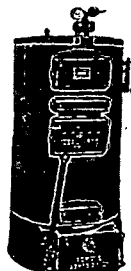
ARCO ROUND BOILER

A low priced boiler, well known for reliable and economical service. Optional front provides for oil burning. **Ratings:** Steam, 330 to 875 sq ft; Water, 485 to 1310 sq ft.

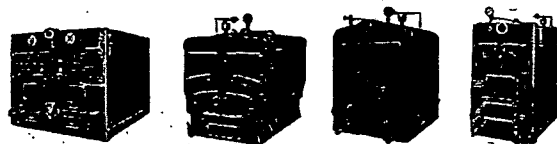


NEW ARCO ROUND BOILER

An old favorite with a new jacket and new features, including the Arco Circulator on certain sizes. **Ratings:** Steam, 240 to 690 sq ft; Water 380 to 1100 sq ft.



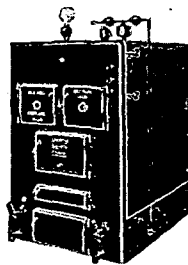
BOILERS FOR LARGER INSTALLATIONS



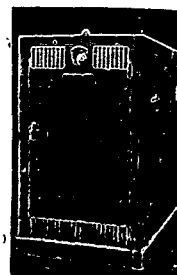
"Ideal" Water Tube Boilers—Five sizes from 23 in. to 79 in., for large buildings and commercial installations. Sectional construction permits easy installation in old buildings. **Ratings:** Steam, 880 to 15,530 sq ft; Water, 1400 to 24,870 sq ft.

MAKE IT AN "ALL AMERICAN" JOB!

"IDEAL" REDFLASH BOILERS



Regular

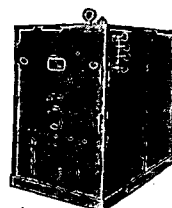


DeLuxe

An attractive boiler in regular or Deluxe Model for the modern home. 40 sizes for all fuels provide for every requirement. Special front for oil burning can be furnished. **Ratings:** Steam 365 to 7810 sq ft; Water, 585 to 12,500 sq ft.



ARCOLA—A simple, hot water radiator system for small homes up to 6 rooms, garages, outhouses, stores, etc. Generally sold complete with radiators. Needs no basement for installation.



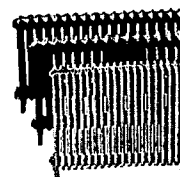
No. 92 Oil Burning Boiler — A deluxe model for larger than average homes, apartments, stores, churches, etc. **Ratings:** Steam: 1815 to 3675 sq ft; Water, 2910 to 5870 sq ft.

AMERICAN RADIATOR COMPANY

40 West 40th Street, New York, N. Y.

CAST IRON RADIATORS AND ENCLOSURES

ARCO RADIATORS



Small and trim, these radiators have an output equal to old types of larger radiators. They are handsome as free standing radiators, yet small enough to be recessed.

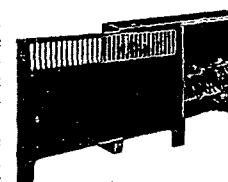
CORTO RADIATORS

A complete range of widths and heights permits an appropriate selection of this favorite radiator for any location. Highly efficient, easily cleaned and compact.



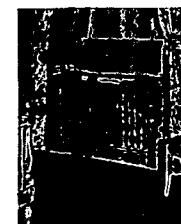
ARCO CONVECTOR

A cast-iron unit and enclosure designed together for most efficient concealed heating. A complete range of sizes in the Convector and in the enclosure, provides proper combination for every requirement.



FANTOM RADIATORS

The Fantom is a wall type radiator, hung on brackets preferably under a window, and fully or partially recessed with front exposed and grilled metal stool over top.



A COMPLETE LINE OF RADIATOR ENCLOSURES

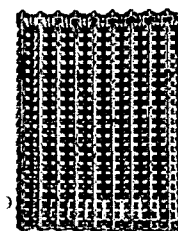
These radiator enclosures have been worked out to combine pleasing appearance with heating efficiency. They are made for every type of radiator, either

recessed, semi-recessed or free standing. They are sturdy, easily installed and reasonably priced. Special models may be had on special order.

Engineers are giving increasing recognition to the value of

ARCO CAST-IRON PRODUCTS FOR INDIRECT HEATING

VENTO CAST-IRON RADIATORS

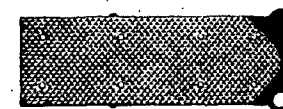


Vento Cast-Iron Heaters are used in many of the foremost schools, post offices, clubs, theatres and other public buildings for efficient indirect heating. They have become the standard for blower and ventilation systems.



PERFECTION PIN

For Gravity Indirect Heating with Steam or Water. Threaded Nipple or Flange and Bolt Connections.



MAKE IT AN "ALL AMERICAN" JOB!

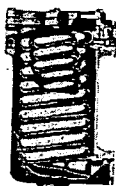
AMERICAN RADIATOR COMPANY

40 West 40th Street, New York, N. Y.

PRODUCTS FOR EVERY HEATING REQUIREMENT

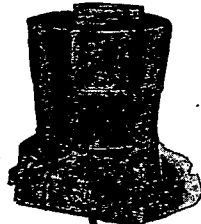
CONTROLS AND ACCESSORIES

EXCELISO WATER HEATER



Provides an easy and economical supply of domestic hot water from a steam or hot water boiler. Water from each section of the boiler passes into the metal shell of the Excelso, heating the domestic water in the copper coils.

SCUTTLE-A-DAY



A practical, coal burning water heater for small homes, stores, etc. Connected with a storage tank, it provides ample hot water for domestic use. 3 sizes for various requirements.

The Arco Incinerator—combines the features of the Scuttle-a-day water heater with an incinerator. Made in 40 to 90 gal. capacities.

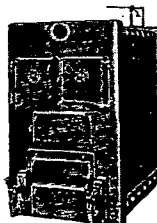
IDEAL DOME TYPE WATER HEATER

A new cast-iron heater for coal burning. Made in 7 sizes, in capacities from 65 to 250 gal. Plant tested to 315 lb. pressure and guaranteed for working pressure of 125 lb. Substantial in construction and moderately priced.

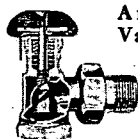


ARCO HIGHEST TANK HEATER

A water heater for large buildings. Low-cost installation and inexpensive operation but will give unsurpassed length and quality of service. For either coal or oil. Capacities: Coal, 1,500 to 3,000 gal; Oil, 1,520 to 2,520 gal.



A COMPLETE LINE OF PACKLESS VALVES, VENTS AND AIR VALVES



Arco Packless Valves—Provide absolute protection against leaks in steam, hot water, vapor or vacuum heating systems.



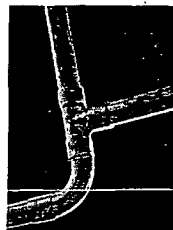
No. 515 Ideal Vacuum Air Valve—For one-pipe steam systems. Eliminates air rapidly and seals against its return.



No. 861 Arco Hurriver—Its unique construction allows for quick and great venting capacity assuring almost instant steam supply from mains to radiators.

ARCO WROUGHT COPPER FULL-FLOW FITTINGS AND ARCO WROUGHT COPPER PIPE

ARCO Full-Flow Fittings are the logical companions to copper pipe. They are wrought copper like the pipe they join, with the same coefficient of heating, cooling, contraction and expansion. Fittings and pipe available up to 2 in. Pipe is hard or soft temper, and in 3 weights.



ARCO METAL PIPE

Arco Super Alloy Pipe is strong, machinable and corrosion resisting. Made of high grade of cast-iron alloyed with copper and chromium. Furnished in 1½ in. to 6 in. sizes, inclusive. Threads and cuts with standard tools.

MAKE IT AN "ALL AMERICAN" JOB!

AMERICAN RADIATOR COMPANY

40 West 40th Street, New York, N. Y.

CONTROLLED WARMTH SYSTEMS

The Arco Model K Vapo-Orifice System is an engineered and coordinated unit which operates to insure equal distribution of warmth. On each radiator there is installed a special inlet valve with an adjustable orifice, by means of which flow of steam to each radiator can be accurately calibrated in accordance with the capacity of the radiator. The adjustment can be made while the system is in operation.

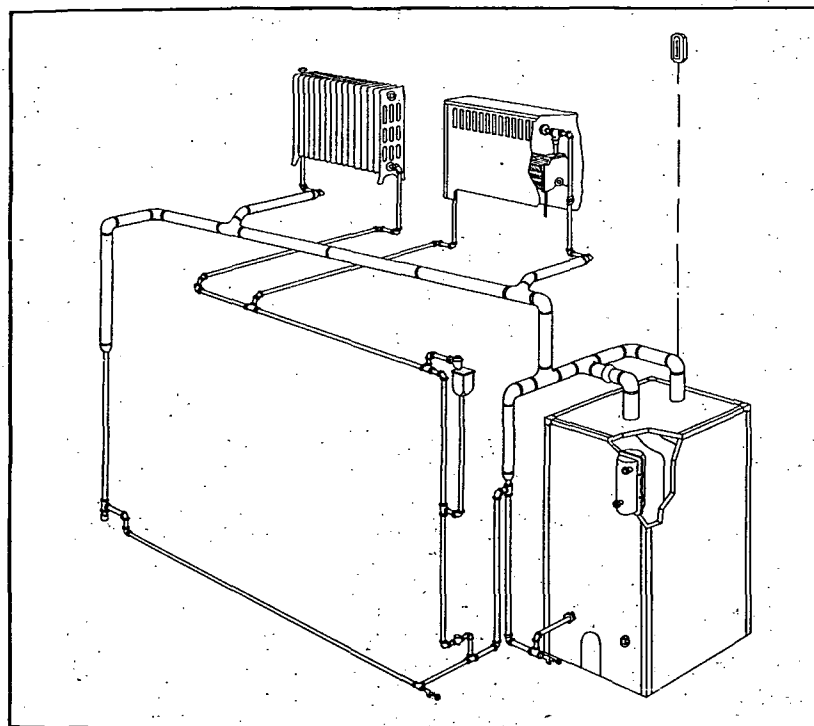
Pressure is so controlled that no more steam is admitted to the radiators than they are capable of condensing. Therefore no thermostatic traps are required on the return connections of the radiators. A thermostatic air eliminator equipped with a vacuum check is provided. If the pressure through inadvertence should exceed

the design pressure, steam will enter the return lines, close the vent port of the air eliminator, and the system will automatically equalize itself, insuring the return of condensate to the boiler.

When a zone system is used, the zone valves prevent the self-equalizing action and it is necessary to install an Arco Condensation Pump to insure the return of condensate to the boiler.

Simplicity of design and moderate cost makes possible the installation of this system for little more than the cost of a One Pipe Steam System.

"Standard Specifications" and "Design and Installation Guide" including all details are available at American Radiator Company branch offices.



MAKE IT AN "ALL AMERICAN" JOB!

Burnham Boiler Corporation

Manufacturers of Cast Iron and Welded Steel Boilers,
Cast Iron Radiators and Heating Accessories

Irvington-on-Hudson, N. Y.

Branch Offices

BOSTON; PHILADELPHIA; CHICAGO;
QUEENS VILLAGE, L. I.; LONG ISLAND
CITY, N. Y.; BALTIMORE; SPRINGFIELD;
LANCASTER; PITTSBURGH; ZANESVILLE;
ELIZABETH; GENEVA, N. Y.



Plants

ELIZABETH, N. J.; LANCASTER, PA.;
ZANESVILLE, OHIO; GENEVA, N. Y.

BURNHAM BUILT-IN-OIL BURNING BOILER

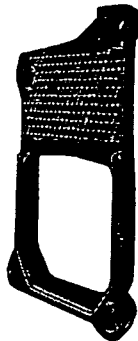
Design

Designed especially for securing greatest economy with oil fuel. Is not limited to use of any particular oil burner. Will operate satisfactorily with any gun-type burner. Equipped for all-year-round hot water supply. Provision made for six automatic controls insuring positive action and safety.



Construction

Cast iron square sectional, encased in heavy gauge steel jacket, attractively finished in spatterdash aluminum. Duco lacquered. Jacket of boiler heavily insulated against heat and sound. Every thing is built-in. Controls are in burner division, reached through door at back. Refractory furnished with boiler, also water heater, low water cut-off and draft adjuster.



Performance

Note all sides of fire pot or combustion chamber are water surrounded, even the bottom. This latter prevents heat from converting the moisture into steam below the usual concrete bottom, with danger of causing explosions. All fire travel is vertical and between a series of staggered teeth which literally eat up the heat. Between the crown sheet and smokebox, a distance of 19 in., approximately 2000 deg of heat is gathered by the water. The fact that when you open the peep door you can see for yourself, that practically none of the color flames of combustion go into the fire travel, proves conclusively how perfect is the combustion. It has a particularly low heat lag, due to the fact that the air supply is cut off soon as burner ceases. No cooling is therefore caused by chimney draft drawing air through the boiler. It is an economy boiler built as Burnham builds.

Boiler Ratings

No. of Boiler	Equivalent Direct Radiation	Minimum Gallons of Oil per Hour to Sustain Rating
Steam		
S. B.	725	1.5
S. C.	1075	2.25
Water		
W. B.	1160	1.5
W. C.	1720	2.25

Height.....	57 in.
Width.....	29 in.
Depth.....	36 in. for No. B. 42 in. for No. C.
Water line height.....	46 in.
Oil burner chamber.....	13 in. deep 27 1/2 in. wide 56 in. high
Supplies and returns.....	two 3-in. supplies two 3-in. returns
Smoke pipe size.....	No. S. B. 8 in. No. S. C. 9 in. Height to center, 50 3/8 in.

There's a Burnham for Every Heating Purpose

Burnham Boiler Corporation

Irvington-on-Hudson, New York

There's A
for Every



Burnham
Purpose



Catalogs Sent on Request

1—Welded Steel Boilers—Also Three Purpose Welded Steel Boilers.

For heating, hot water supply and incineration. Coal or oil. Completely welded for 15 lb working pressure. Multiple shaking grates. Sizes for commercial or domestic uses. Special folder sent on request.

2—Water Tube Boilers for Steam and Hot Water Heating.

17, 21, 27, and 36 in. Double shaking grates and long fire travel. Rating to 9,050 sq ft for steam and 15,085 sq ft for water.

3—Water Tube Boilers Jacketed in Color.

17, 21, and 27 in. Steel Jacket and 4 ply air cell asbestos insulation. Enameled rich red and black. Jacket goes on after all other set up work. Rating to 4,225 sq ft for steam and 6,800 sq ft for water.

4—Burnham Oil-Burning Boilers.

A specific-sized boiler for each specific heat job for use with any standard oil burner. Round Sectional Burnhams in 6 series and 24 sizes. Square Burnhams in 5 series and 39 sizes. For steam, vapor, or water.

5—Big Twin Sectional Boilers.

50 in. Grate, divided for easy shaking. Twin sections, divided down the middle. Ratings to 19,450 sq ft for steam 31,800 sq ft for water.

6—Tube Type Smokeless Boilers.

For burning soft coal efficiently and without smoke. Meet smoke ordinances everywhere. Similar to (2) above with addition of smokeless feature.

7—Round Sectional Boilers.

This boiler made the long fire travel famous. Handled easily. Very large steam dome. Ratings up to 1,550 sq ft for steam, 2,560 for water.

8—High Pressure Hot Water Supply Boilers.

Sectional construction Guaranteed to 120 lb working pressure. Supplies up to 3,800 gal.

9—Junior Hot Water Supply Boilers.

Will keep 175 to 700 gal tank always full of hot water. Guaranteed to 120 lb working pressure.

10—Burnham-Taco Tanks.

Combining water heater and storage tank in one unit for summer-winter use. Removable copper heating element. Tanks may be galvanized, evendur or copper.

11—Burnham-Cast-Iron Radiators.

Occupy 40% less space than ordinary cast-iron radiators of same rating. Shorter. Lower. Narrower. 3-tube type 3 1/4 in. wide. 4-tube type 4 7/8 in. wide. 5-tube type 5 1/2 in. wide. Can be recessed.

12—Fero Tube Radiators.

All heights—3, 4, 5, 6, and 7 tubes.

13—Burnham Air and Vacuum Valves.

Full line for radiators, risers and mains.

14—Burnham Radiator Valves.

Complete line of heating accessories. Including steel tanks of all kinds.

15—Burnham Flexible Headers.

16—Federal Unit Heaters.

Complete line in modern designs.

Crane Co.

Manufacturers of Valves, Fittings, Fabricated Piping, Steam
Specialties, Plumbing and Heating Materials

836 S. Michigan Avenue, Chicago, Ill.

Branches in All Principal Cities

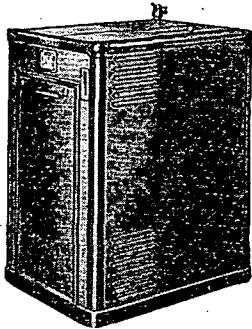
Write for Catalogs and full information on any materials

COMPLETE LINE OF RESIDENTIAL AND COMMERCIAL BUILDING BOILERS AND HEATING MATERIALS

Crane boilers are made in sectional or round cast-iron styles for hot water, steam, vacuum, and vapor heating with coal, coke, gas, and oil. Individual sizes and capacities up to 10,450 sq. ft. of total steam radiation or 16,720 total hot water; used in batteries, installations with a much larger capacity are available.

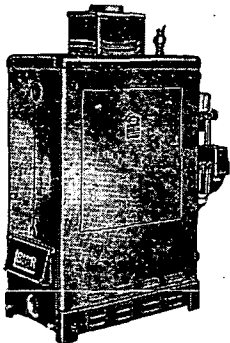
Crane Deluxe D-1 Oil Burning Boiler

The "Sustained Heat" principle of design incorporated in this boiler assures high combustion efficiency and low standby-loss. Available in regular and deluxe jacket models.



Basmor Gas Fired Boiler

With the famous Butterfly Bunsen-type burner, this boiler develops high efficiency with negligible standby-loss. All types of gas. Available in both regular and deluxe jacket models.

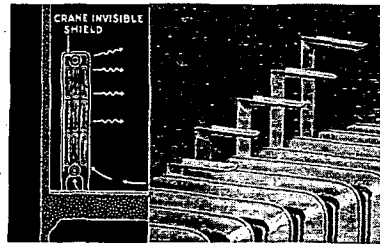
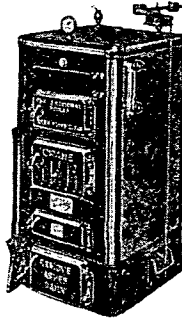


Crane Radiation includes exposed (legless, bathroom, hospital, wall), concealed, and shielded types of high efficiency and graceful design.

Unit heaters, humidifying radiators, and a complete assortment of valves, fittings and specialties complete the Crane heating line for residential and commercial use.

Crane Coal Fired Boilers

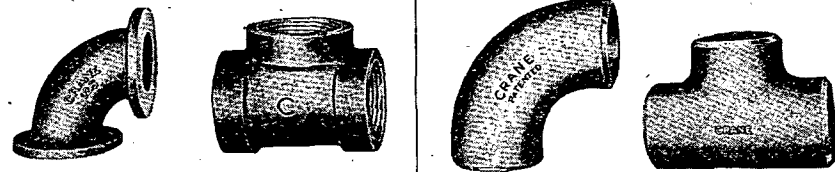
Extended "ceiling" surface, controlled water travel, and a score of operating refinements place this boiler on a plane of highest efficiency. Available in both regular and deluxe two-tone enamel jackets.



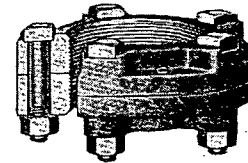
Crane Directed Radiation

Invisible low-cost section shields throw heat outward into room, equalizing temperatures, protecting walls and drapes.

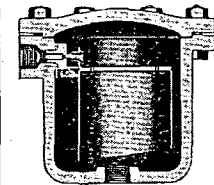
EVERY TYPE OF VALVE AND FITTING FOR HEATING INSTALLATIONS



Crane Welding Fittings—Uniform in cross-section and radius, with ends accurately beveled to fit pipe or tubing. Tangents facilitate welding in line.

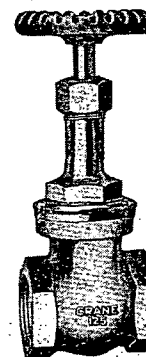
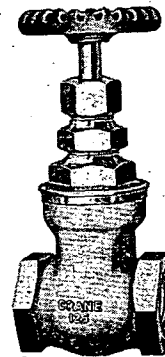


Crane Standard Cast Iron Fittings—Of dense, uniform metal, with faces accurately aligned and at exact right angle to flow. Smooth-cut threads, smoothly faced flanges.



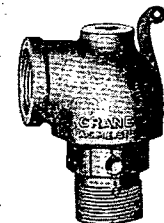
Crane No. 981 Trap—A cast-iron trap of high capacity. Simple working parts, removable without touching line. Expels condensate in instantaneous action.

Crane No. 512 Gem Union—A ground-joint union with brass to iron seat. No gasket required. Seat ring cannot loosen. Taper threads. Easy to install. 150 lb. steam.



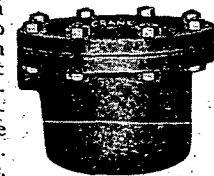
Crane Wedge Gate Valve (Left)—A brass valve with non-rising stem, stuffing box gland, and disc guides that prevent wear of seating surfaces. Can be packed while wide open under pressure. For steam working pressures up to 125 lb.

Crane Double Disc Gate Valve (Right)—A brass valve with rising stem, parallel seats, and wedging device assuring equal bearing on all parts of seat. Can be packed while wide open under pressure. For steam working pressures up to 125 lb.



Crane No. 1168 L. P. Pop-Safety Valve—A side-outlet valve of malleable iron, with brass trimmings, for steam heating boilers. Sturdy rigid base and ample clearance for tools. Regularly set at 5, 10 or 15 lb.; available at 3 to 25 lb.

Crane No. 984 Air Vent—An automatic vent to remove air from water mains, hot water heating systems, tanks. Operation similar to the No. 981 trap. Simple, low in price.



National Radiator Corporation

General Offices: Johnstown, Pa.

Sales and Service Through These Branch Offices and Warehouses

BALTIMORE	BOSTON	BUFFALO	CHICAGO	CINCINNATI
CLEVELAND	LEBANON	NEW YORK	MILWAUKEE	
PHILADELPHIA	PITTSBURGH	RICHMOND	WASHINGTON	

HEATING EQUIPMENT

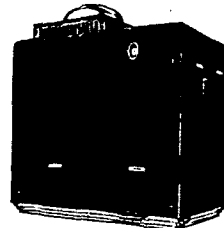


National Boiler Bond
—Issued and backed by a great independent surety company, a bond is given free with every National Bonded Boiler. It guarantees workmanship, materials, and official published ratings. Bond fully covers the purchase price of the boiler.



National Bonded Contento Boiler

For radiator systems in small homes or structures with or without basements. Hot Water only. Made in 5 sizes, with bonded ratings from 225 to 545 sq. ft. Burns hard coal, soft coal or coke.



National Bonded Gas Boiler

French gray enamel finish, chromium base and dome, fully enclosed. Made in 2 series and 16 sizes. Certified A.G.A. ratings from 300 to 2250 sq. ft. Steam, and 480 to 3600 sq. ft. Water.



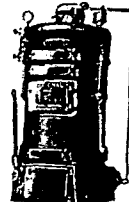
National Bonded Jacketed Square Boilers

Red baked-enamel jacket, black base and doors. Made in 3 series and 14 sizes, to meet the requirements of every modern home. Bonded ratings from 200 to 1400 sq. ft. Steam, and 330 to 2310 sq. ft. Water. Insulated with rock wool.



National Bonded Jacketed Oil-Burning Boilers

Red baked-enamel jacket, with black base and doors. Furnished with special front for either rotary or gun type burner. Made in 3 series and 14 sizes. Bonded ratings from 395 to 1570 sq. ft. Steam, and 632 to 2512 sq. ft. Water.



National Bonded Round Boilers

Business-like and efficient; an established favorite in many districts. For any fuel. Made in 6 series and 18 sizes. Bonded ratings from 175 to 925 sq. ft. Steam, and 285 to 1530 sq. ft. Water. Triangular or flat type grate bars are optional.

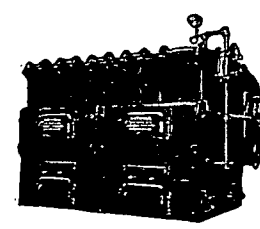
National Radiator Corporation

HEATING EQUIPMENT



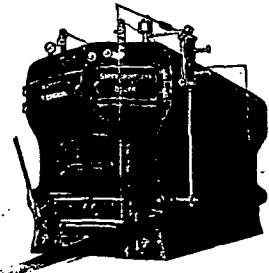
National Bonded Novus Sectional Boiler

(Including the Imperial Sectional Boiler.) Burn any fuel. Largest sizes have divided sections. Five series, 31 sizes. Bonded ratings from 480 to 7000 sq. ft. Steam, and 790 to 11500 sq. ft. Water. Imperial Sectional is of the Utica design.



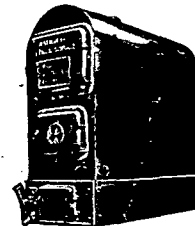
National Bonded Low Water Line Boiler

Eliminates high boiler room ceilings and boiler pits. Burn any fuel. Made in 2 series, 21 sizes, hard-coal or smokeless. Bonded ratings from 850 to 6600 sq. ft. Steam, and 1400 to 10,890 sq. ft. Water. Rear smoke outlet on larger sizes.



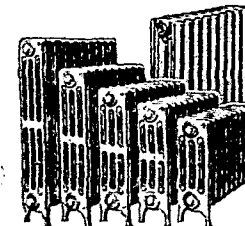
National Bonded Super-Smokeless Boiler

These smokeless boilers have demonstrated their ability to meet smoke ordinances in all parts of the country. Three series, and 24 sizes. Bonded ratings from 750 to 8300 sq. ft. Steam, and 1250 to 13,695 sq. ft. Water.



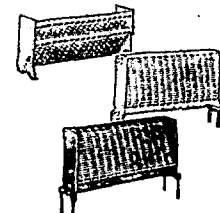
National Premier Steel Boilers

Have recessed front smoke chamber, water jacketed fivesides. Models for hand or stoker, oil, or gas firing. 11 grate widths, 29 sizes. Type C furnished in hard coal, or smokeless types. Ratings . . . 485 to 42,500 sq. ft. Steam; S.H.B.I. Code.



National Aero Panel and Tube Radiators

National Panel Radiation (shown at top of illustration) has iron-to-iron contact on front face. Made in 1, 2, 3, 4 and 5 tube depths in varying heights. Aero Tube Radiation is made in 3, 4, 5, 6 and 7-tube styles, in varying heights.



National Aero Cast-Iron Convectors

Proven dependability and permanence. Made of lifetime metal. 8 sizes and models, for recesses 3 3/4, 4 1/4, 5 3/4, 7 3/4 and 9 1/4 inches deep. Any length, in multiples of 2 inches. Fin spacing provides large volume of moderately warmed air.

Ask the Nearest Branch for complete technical information on any of these National Radiator Products.

Spencer Heater Company

Williamsport, Pa.

Branch Offices and Representatives

NEW YORK, N. Y.
PHILADELPHIA, PA.
BALTIMORE, MD.

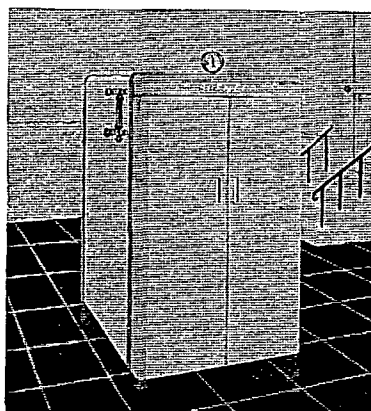
BOSTON, MASS.
SPRINGFIELD, MASS.
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ALBANY, N. Y.
BINGHAMTON, N. Y.
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SYRACUSE, N. Y.
ALLENTOWN, PA.
SCRANTON, PA.

Spencer Automatic Magazine Feed Heaters are made in cast-iron sectional types for steam, vapor and hot water heating. Also in steel tubular types for larger buildings. There are sizes and capacities to provide economical and convenient heat—safe and sure—for every type of building.

Spencer is the original magazine feed heater with a record of more than 37 years' successful operation. They are sold and installed by all good heating contractors.



Why Spencer Heaters perform so satisfactorily can best be explained by a brief inspection of their design and construction. The Spencer principle—as illustrated in the cross sectional view—is simple:

Once a day fuel is put into the magazine (A). It fills the sloping grate to the level of the magazine mouth (B). The fire bed always stays at (C) as far as fuel burns to ash (D) it shrinks and settles on the sloping grate (E) and more fuel feeds 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 Heater always gives the same uniform satisfying heat, and burns low cost coal economically. These exclusive Spencer advantages are available in all types, J and L series, cast-iron sectional, burning low cost No. 1 Buckwheat; the C-N series for larger

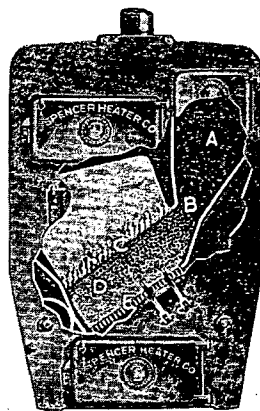
fuels; or the steel magazine feed heaters, also for No. 1 Buckwheat.

Coal - Coke - Gas - Oil—Spencer J and L series cast-iron and steel tubular heaters are primarily designed to burn low cost No. 1 Buckwheat anthracite and the C-N series Nut or Pea anthracite or coke.

If at any time a home owner desires to burn more expensive fuels—oil or gas—a Spencer Heater will show a higher efficiency than can be secured with any ordinary boiler.

Thermostats—A Minneapolis thermostat and electric damper motor are furnished as optional equipment.

Jacketed Covering—Attractive metallic jackets, as illustrated are available for Spencer Cast-Iron Heaters.



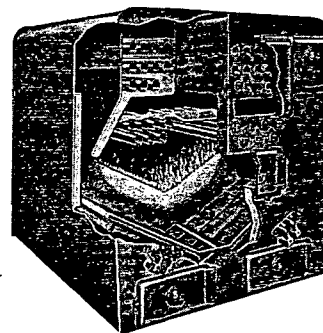
*The Spencer Heater Company guarantees that when installation is made in accordance with accepted standards, the heater will carry the number of feet of direct cast-iron column radiation listed. This allows ample provision for heat loss in covered mains, risers and returns, and for peak loads.

SPENCER STEEL TUBULAR MAGAZINE FEED BOILERS

For larger buildings we recommend Spencer Steel Tubular Boilers, also of the magazine feed type, burning low cost No. 1 Buckwheat anthracite or coke.

In the cross-section diagram illustrated here, 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 of its own weight to the fire. These boilers are built in two vertical sections for ease in handling and installation—a great advantage on remodeling or replacement jobs, eliminating necessity of costly tearing out of walls, etc.

Combination water tube and fire tube construction. Built to A.S.M.E. standards.



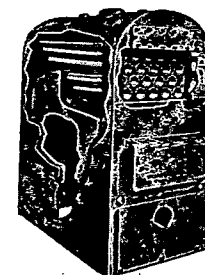
Sizes and Guaranteed Capacities

No.	STEAM	
	*Guaranteed Direct Cast-Iron Column Radiation Loads, Sq. Ft.	†E.D.R. Rating
M6-6	2300	2875
M6-7	2600	3250
M6-8	2900	3625
M6-9	3200	4000
M6-10	3500	4375
M7-6	4000	5000
M7-7	4700	5875
M7-8	5400	6750
M7-9	6100	7625
M7-10	6800	8500
M8-6	8000	10000
M8-7	9750	12185
M8-8	11500	14370
M8-9	13250	16555
M8-10	15000	18740

†E.D.R. ratings represent the attached net radiation load plus piping losses expressed in square feet of equivalent direct radiation (240 B.t.u. steam, 150 B.t.u. water) which may be placed on heater in accordance with accepted standards for economical operation.

SPENCER STEEL BOILERS, FOR OIL, STOKER OR GAS

For more than 37 years Spencer has been building efficient, economical and dependable coal burning boilers. With this background of experience, Spencer engineers have developed the Spencer Steel Boiler, for oil, gas or stoker.



The high sustained efficiency of these boilers means adequate heat for low fuel cost. Design is of the three-pass type. Combustion chamber is amply large. Built of open hearth steel boiler plate and electrone tubes—a quality product for dependable service.

Furnished with Taco indirect domestic water heating coils either of the storage tank or instantaneous type.

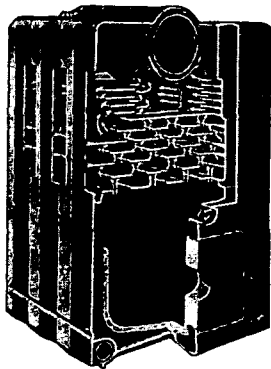
Sizes and Guaranteed Capacities

CAST-IRON SECTIONAL TYPE

No.	STEAM		WATER	
	*Guaranteed Direct Cast-Iron Column Radiation Loads, Sq. Ft.	†E.D.R. Rating	*Guaranteed Direct Cast-Iron Column Radiation Loads, Sq. Ft.	†E.D.R. Rating
J-3	175	220	290	360
J-4	265	330	440	550
J-5	355	440	590	740
L-105	390	490	645	800
L-106	510	640	845	1050
L-107	630	790	1045	1300
L-205	550	690	910	1140
L-206	725	910	1200	1500
L-207	900	1150	1490	1860
L-208	1075	1350	1780	2220
L-209	1250	1570	2070	2580
L-305	1150	1430	1900	2375
L-306	1500	1870	2475	3095
L-307	1850	2310	3050	3815
L-308	2200	2750	3625	4535
L-309	2550	3190	4200	5255
L-310	2900	3630	4775	5975
L-311	3250	4070	5350	6695
CN-502	280	350	460	575
CN-502½	360	450	590	735
CN-503	440	550	725	905
CN-5K3	520	650	860	1070
CN-504	600	750	995	1235
CN-5K4	680	850	1130	1400
CN-505	760	950	1265	1565
CN-5K5	840	1050	1400	1730
CN-506	920	1150	1535	1895

UNITED STATES RADIATOR CORPORATION

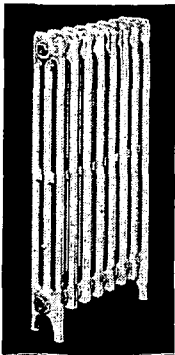
Detroit, Michigan, U. S. A.



Illustrated above is the "O" Series Boiler for horizontally fired "gun type" oil burners. The high efficiencies of this boiler are due to a new method of heat absorption through extended rib type heating surfaces. The boiler is of the wet base construction, which absorbs heat that would ordinarily be lost through the floor. Year 'round domestic hot water is provided.

Construction is of cast-iron. A heavy jacket and a thick blanket of rock wool completely insulate the boiler.

CAPITOL RADIATORS



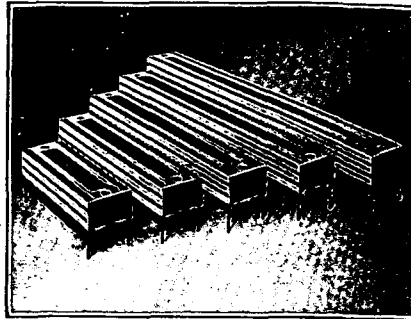
Capitol Radiators give mellow glowing warmth where you want it and when you want it.

Homes heated with Capitol Radiators are comfortable, clean, and healthful.

Capitol Radiators have the added advantage of the most approved type of assembly. Extra heavy malleable iron push nipples, machined with hair-breadth precision, form a tight, iron-to-iron joint. They need no

gaskets, have no threads to rust, are taken apart and assembled with the greatest ease.

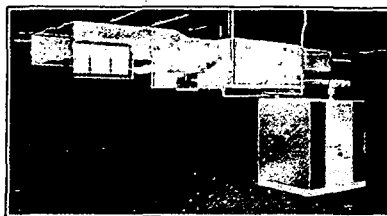
Capitol Fincast Convector



CAPITOL FINCAST CONVECTORS AND ENCLOSURES

Made entirely of cast-iron.
Made without joints.
Cast in one piece.
Many lengths and widths.
Tappings—top, bottom or ends.
Complete choice of enclosures.

THE CAPITOL AIR CONDITIONER



For use in residences and small buildings.

Flexibility to meet the individual requirements of the building is the keynote of design in the Air-Conditioner. The owner may have a completely air-conditioned building; he may have a partially air-conditioned building with modern steam or hot water heating in less frequented rooms; or he may have a system that conditions the air in the winter with a certain amount of cooling effect during the summer, due to circulation of air by a big quiet fan.

Weil-McLain Company

Manufacturing Division: Michigan City, Ind. and Erie, Pa.

General Offices: 641 W. Lake Street, Chicago

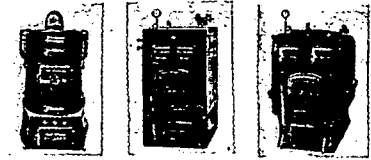
NEW YORK OFFICES: 501 Fifth Avenue

Prompt 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.

PRODUCTS—Cast-Iron Heating Boilers and Water Supply Heaters, Cast-Iron Radiators including standard tubular senior and junior types, "RAYDIANT" Concealed Cabinet and Wall Radiators, and Humidifiers.

Weil-McLain BOILERS

Weil-McLain Boilers are made in various types and in a wide range of sizes to satisfy every heating need and demand. The conventional line includes the Round, Jacketed, Square, Self-Feed and Smokeless types. A more recent addition is the Jacketed Type Round Boiler built especially for Automatic Heating with Oil, Gas or Stoker.



Round

Jacketed

Square



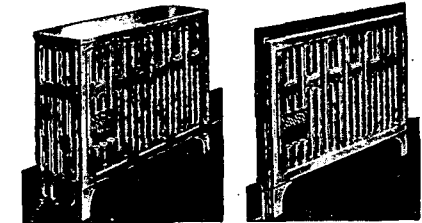
Self Feed

Jacketed All-Fuel

Jacketed Oil Burning

Raydiant with Heat Radiating LIVE Front

The Weil-McLain Raydiant Radiator is made in both concealed and cabinet enclosure types. It emits a scientific blend of radiant warmth and convected heat. Made entirely of cast-iron, it forms its own live metal front and grille. Has space-saving advantages, low roughing-in and setting cost. Vents equally well on all systems.

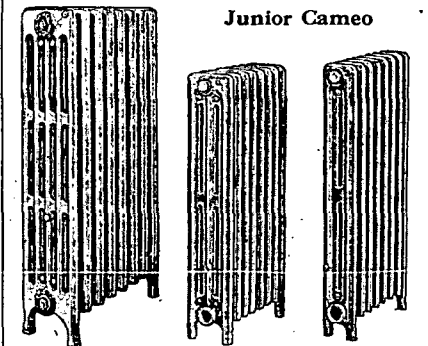


Cabinet Type

Concealed Type

CAMEO WEIL-McLAIN RADIATORS

Weil-McLain "Cameo" Radiators are made in a complete range of sizes, heights and widths, for every place and purpose in modern homes and buildings. Besides the regular floor type, Weil-McLain also make other types of "Cameo" such as the Junior floor type radiator, "Cameo" Wall and Bathroom types.



Delco Appliance Corporation

Subsidiary of General Motors

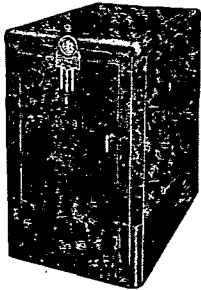
Rochester, N. Y.

DELCO-HEAT

Delco-Heat Harmonized Boiler

A Complete Unit with Boiler and Burner Designed for Each Other—To secure the greatest benefits with the use of oil as the heating medium, it is necessary to have a boiler which is designed to be used with oil heat. An oil burner operates at full volume as soon as it is turned on. It is therefore necessary that a very large heating surface be provided in the boiler to absorb this heat as rapidly as possible in order to prevent waste. The Delco-Heat Boiler is designed with the proper heating surface to insure this efficiency of absorption with the type of flame produced by the Delco-Heat Burner.

The Boiler is built for steam vapor or hot water heating systems. It has capacities of from 625 to 1725 sq ft of steam radiation and from 1000 to 2750 of hot water radiation. It can be passed through any average doorway in units not exceeding 420 lb in weight. The boiler section is insulated on top and all sides with heavy insulation. The bottom of the boiler is uninsulated to heat the air which passes under the boiler to the burner and is preheated before being used for combustion. The steel cabinet, finished in two-tone green and chromium, encloses the boiler and all equipment and controls.

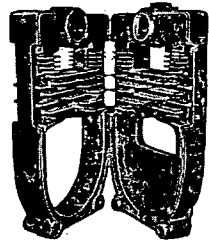


One of the advantages of the design of this boiler-burner unit is that the burner is located in the front of the cabinet, entirely outside the combustion chamber. It is therefore easily accessible if servicing is required. It is designed and built as part of this boiler but otherwise is similar to the standard Delco-Heat Burner. The Burner is designed to utilize low cost domestic fuel oil which requires no preheating.

Efficiency of Fins and Construction

The efficiency of cast iron fin type construction for heat transmission has been demonstrated for two decades in air-cooled Delco-Light electric plants where rapid heat liberation is desired. In the Delco-Heat Boiler, the process is reversed and cast iron fins rapidly absorb the heat from flame and gases in the boiler. The fins contain a thin sheet of water which quickly absorbs heat and circulates back into the main body of water, causing quick steaming. The position of fins and the staggered and broken-up flame and gas flow are clearly shown in the illustration. Progress of the flame is thus delayed until it finally enters a central duct at top of the boiler and then passes out the flue.

The upper nipple port is unusually large, permitting free circulation of water and steam within the boiler. Unrestricted circulation above the crown sheet is possible because of the unusual design. The boiler operates with a steady water line which is important, as a fluctuating water line operation might cause low water cut-off and stop burner.



Delco Heat Boiler Sections Showing Fin Arrangement

The boiler is used for steam, vapor or hot water heating systems, the only difference being in the accessories. The hot water heater consists of a special copper tube set into the rear section of the boiler and submerged in the boiler water. There are two capacities. On the DL-3, DL-4 and DL-5 boilers, the hot water heater will raise 33 gal of water 100 deg (when the boiler is 180 deg) each hour. On model DL-6 under the same circumstances, the heater has a capacity of 50 gal per hour.

Automatic Controls

Complete Automatic Control Service is standard equipment with the Delco-Heat Harmonized Boiler.

See also Pages 885 and 890

THE BABCOCK & WILCOX COMPANY

85 Liberty Street,

Manufacturers of

New York, N. Y.

Water-Tube Boilers

Oil Burners



Chain-Grate Stokers

Seamless Steel Tubing and Pipe

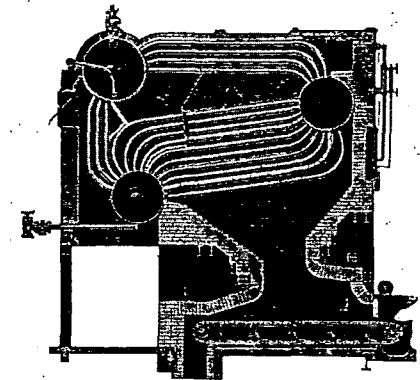
Branch Offices and Representatives in all Principal Cities

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 4980 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

The advantages of the Babcock & Wilcox Type H Stirling Boiler may be summarized as follows:

Unusual steaming capacity for the floor space and head-room required.

Boilers may be set singly or in battery. Setting heights can be varied to suit any condition of firing.

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.

Soot blowers can be readily installed to simplify thorough cleaning of all tubes.

A superheater can be furnished without any change in the standard design or construction.

The boiler is supported by a structural-steel framework entirely independent of the brickwork.

Ample provision is made for free movement of parts due to expansion and contraction.

A complete table of sizes and dimensions, together with pertinent installation data, is contained in a new 32 page bulletin which will be sent upon request. Simply ask for Bulletin G-8-A.

Class	Heating Surface Sq. Ft.	Depth of Setting, Ft., In.	Width of Setting		Floor to Center of Mud Drum, Ft., In.	Floor to Face of Steam Outlet, Ft., In.	Floor to Top of Boiler, Ft., In.	Size of Steam Outlet, In.
			Single Boiler, Ft., In.	Two Boilers in Battery, Ft., In.				
H-1	691	15-2	6-0	11-0	5-2	14-3/4	14-2/4	5
	921	"	7-0	13-0	"	"	"	"
	1152	"	8-0	15-0	"	"	"	"
	1382	"	9-0	17-0	"	"	"	"
	1612	"	10-0	19-0	"	"	"	"
	1843	"	11-0	21-0	"	"	"	"
	2073	"	12-0	23-0	"	"	"	"
H-2	2304	"	13-0	25-0	"	"	"	"
	877	17-8	6-0	11-0	4-9/4	14-5/2	14-2/4	5
	1169	"	7-0	13-0	"	"	"	"
	1462	"	8-0	15-0	"	"	"	"
	1754	"	9-0	17-0	"	"	"	"
	2046	"	10-0	19-0	"	"	"	"
	2339	"	11-0	21-0	"	"	"	"
H-3	2631	"	12-0	23-0	"	"	"	"
	2924	"	13-0	25-0	"	"	"	"
	1063	20-2	6-0	11-0	4-5/2	14-5/2	14-2/4	5
	1417	"	7-0	13-0	"	"	"	"
	1772	"	8-0	15-0	"	"	"	"
	2126	"	9-0	17-0	"	"	"	"
	2480	"	10-0	19-0	"	"	"	"
H-4	2835	"	11-0	21-0	"	"	"	"
	3189	"	12-0	23-0	"	"	"	"
	3544	"	13-0	25-0	"	"	"	"
	3898	"	14-0	27-0	"	"	"	6
	4252	"	15-0	29-0	"	"	"	6
	1245	22-8	6-0	11-0	4-11/4	14-11	14-5/2	5
	1660	"	7-0	13-0	"	"	"	"
H-4	2075	"	8-0	15-0	"	"	"	"
	2490	"	9-0	17-0	"	"	"	"
	2905	"	10-0	19-0	"	"	"	"
	3320	"	11-0	21-0	"	"	"	"
	3735	"	12-0	23-0	"	"	"	"
	4150	"	13-0	25-0	"	"	"	"
	4565	"	14-0	27-0	"	"	"	6
H-4	4980	"	15-0	29-0	"	"	"	6

The Bigelow Company

ESTABLISHED 1860

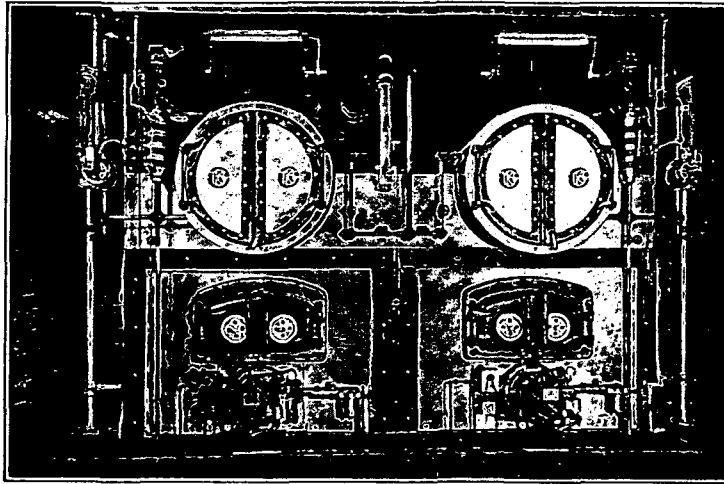
Main Office and Works

BOSTON, MASS.
NEW YORK, N. Y.

New Haven, Connecticut

PHILADELPHIA, PA.
SYRACUSE, N. Y.

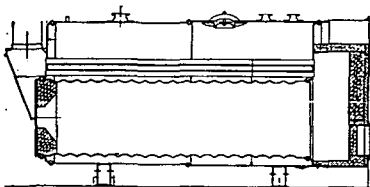
Manufacturers of Bigelow-Hornsby Water Tube Boilers, Bigelow Low Head Water Tube Boilers, Bigelow Three Drum Vertical Water Tube Boilers, Bigelow Horizontal Return Tubular Boilers, Bigelow Electric Steam Generators, Bigelow Scotch Type Boilers, Bigelow Two-Pass Boilers, Bigelow Manning Boilers, Bigelow Upright Boilers.



Two 60 hp Bigelow Two-Pass Boilers used for heating the new Home Office Building of The Hartford Steam Boiler Inspection and Insurance Co., the largest insurer of steam boilers in the United States

Since 1860 it has been the endeavor of The Bigelow Company to merit a reputation for constructing boilers of a high standard of design, workmanship and material. We are proud of the confidence shown in us by the use of Bigelow boilers in this building.

The Bigelow Two-Pass Boiler is designed to meet heating and power requirements, especially where space limitations prevail. It contains the recognized features of the well-known H.R.T. boiler, but due to the small amount of brickwork required it can be installed at a lower complete cost. The elimination of special brick shapes and staybolts in the furnace reduces the cost of maintenance to a minimum. Built in units from 25 to 250 Hp for power service, and the 15 lb. heating class in units from 3500 to 35000 sq ft of steam radiation.



BIGELOW SCOTCH TYPE BOILER

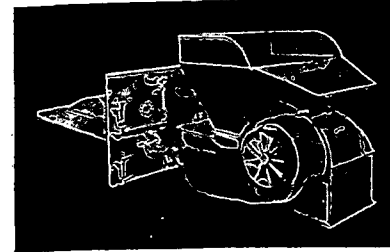
Modeled after the Scotch Marine type of boiler, this boiler has many advantages to warrant its use in the industrial field for power and heating. Self-contained—Compact—Low water-line. Built in units from 15 to 300 hp. The 15 lb. welded heating type built in units from 1800 to 35000 sq ft of steam radiation.

Combustion Engineering Company, Inc.

200 Madison Avenue



New York



C-E STOKER-UNIT

for burning bituminous coal under boilers ranging from small heating units to power boilers developing up to 400 hp. Fuel burning rates from 60 to 1600 lb per hour

Features

A Self-Contained Unit—Hopper, fuel feeding and distributing mechanism, grate, windbox, driving mechanism and forced-draft fan combined in a compact unit.

Installation—The small clearances required usually permit installation with only slight alterations to existing settings. Stoker body is shipped completely assembled ready for installation.

Hopper—Non-clogging and easily removable. Does not interfere with access to firedoors. Located at convenient height for filling with shovel.

Coal Feed—Screw conveyor, protected from heat, advances the coal from hopper to entrance of retort. Reciprocating pusher in retort continues the feeding and provides agitation of the fuel bed in the retort zone.

Grate Surface—Alternate fixed and moving grate bars. Designed for correct air distribution and made of a special heat-resisting iron assuring low maintenance.

Air Supply—Integral forced-draft fan, with inlet damper control, supplies air to windbox under the stoker. Volume of air may be regulated by control lever to suit the rate of coal burning.

Control—Stoker is operated by simple lever control for both coal feed and air supply.

Operation—Simple, easy and dependable. Variable-speed transmission permits sixteen rates of coal feed. Control levers conveniently located. Design of dump grates prevents avalanching of fuel bed when in dumping position.

C-E STOKERS

Combustion Engineering leads in the manufacture of stokers, other than domestic, in extent and variety of work done, nearly 14,000 C-E Stokers having been installed to date. Combustion Engineering's line of larger stokers includes the following, three of which (Type E, Coxe and Green) are the leaders in their respective fields.

Type K Stoker—A single retort under-feed stoker for burning bituminous coals under boilers in the upper size range of the C-E Stoker-Unit.

Type E Stoker—A single retort under-feed stoker for burning bituminous coal under boilers up to about 600 rated hp.

C-E Multiple Retort Stoker—For burning semi-bituminous and bituminous coals under boilers up to the largest sizes.

Coxe Stoker—A traveling grate stoker for burning small sizes of anthracite, coke breeze and lignite.

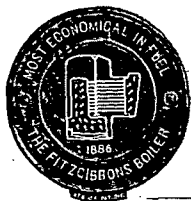
Green Stokers—A chain grate stoker available in both natural and forced draft types for burning non-caking or free-burning bituminous coals.

C-E BOILERS

Combustion Engineering Boilers include all fire tube and water tube types in sizes ranging from 25 hp up to the largest. Included are all designs formerly known by the trade names "Heine," "Walsh & Weidner," "Casey-Hedges" and "Ladd." Classified broadly, the various types of C-E Boilers are as follows: **BENT TUBE**—multi-drum, four-drum, three-drum, two-drum (complete steam generators). **STRAIGHT TUBE**—sectional header, box header (cross drum and long drum). **STEAM GENERATORS**—complete standardized units available in two types. **FIRE TUBE**—hrt, vertical, internally-fired, locomotive type. **MARINE**—sectional header, bent tube. **WASTE HEAT**—straight tube, bent tube.

C-E PULVERIZED FUEL SYSTEMS

C-E Pulverized Fuel Systems, formerly known by the tradename "Lopulco," are available in both direct fired and storage types for boilers ranging from 200 hp up to the largest. Combustion Engineering has recently placed on the market the Raymond Bowl Mill, the most advanced design of pulverizer now available.



Fitzgibbons Boiler Company, Inc.

Established 1886

General Offices: 570 Seventh Avenue, New York, N.Y.

Works: OSWEGO, N.Y.

Branches and Representatives in 55 Principal Cities

PRODUCTS—The Fitzgibbons line of Steel Heating and Power Boilers, manufactured under the Steel Boiler Manufacturers Code, meet the heating requirements of all buildings from the small home to the gigantic modern skyscraper. The line includes types for burning anthracite, bituminous (soft) coals or coke, and for oil, gas or stoker firing, in steam, vapor, vacuum or hot water systems. In other words, there is a Fitzgibbons Steel Boiler for every fuel, and every heating system from the smallest to the largest. **RATINGS:** All boilers of the line are rated in accordance with the S.H.B.I. Code.

FITZGIBBONS OIL-EIGHTY AUTOMATIC

Residence Steel Boiler for Oil Burning

Ratings, Steam—12 Sizes—425 to 2680 Sq. Ft.

Outstanding Features

Tanksaver (optional) supplies year 'round clean hot water without a separate storage tank. **Tankheater** (optional) a more efficient indirect water heater. **Combustrol**, automatically maintains balanced draft, diverts back drafts, prevents back firing. **Thermalizer**, makes every tube do its full share of heat absorbing. **Copper-Steel Plate Construction**, combines maximum strength with corrosion resistance. **Attractively Jacketed**. Teams up with any good rotary or gun type burner to form a unit of **Highest Efficiency**.

Models: Standard—for rotary burners and for exposed gun type burners.

Type B—for gun type burners where the burner is completely concealed but readily accessible through easily removable panels.

Catalog on Request



FITZGIBBONS COAL-EIGHTY

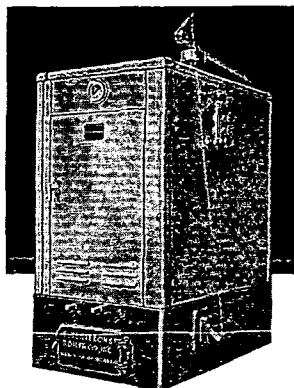
Jacketed Residence Steel Boiler for Coal

Ratings, Steam—400 to 1000 Sq. Ft.

Outstanding Features

Modernly Attractive—A tastefully colored, heavily insulated steel jacket puts this boiler far in advance of coal burning boilers from appearance standpoint. Sets new standards of coal burning performance because of **Thermalizer** and other efficiency increasing features. **Tankheater**, a completely submerged, more efficient indirect heater, is standard equipment. **Fitzgibbons Copper-Steel Plate Construction** provides maximum strength and durability. Ideally suited for **Mechanical Firing** with Stoker or Blower.

Catalog on Request



FITZGIBBONS R-Z-U JUNIOR

Multi-Service Steel Boiler

RATINGS, STEAM

Coal Burning Type..... 750 to 3200 sq. ft.

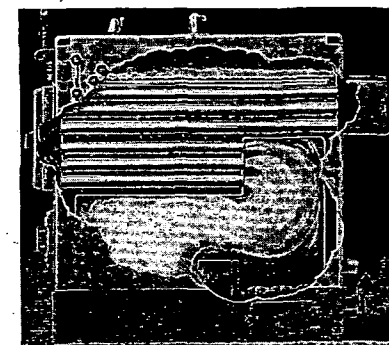
Oil Firing Type..... 1003 to 3893 sq. ft.

Stoker Firing Type..... 1003 to 3893 sq. ft.

Outstanding Features

Tanksaver (optional) supplies year 'round hot water without a separate storage tank. **Tankheater** (optional) a more efficient indirect water heater. **Auxiliary Grate** (optional), for refuse disposal and stand-by heating duty in oil fired installations. **Compact**, largest size will pass thru a 31 in. doorway. **Low Water Line**, eliminates need for a pit. **Jacket** (optional), on all types.

Descriptive Bulletin on Request



FITZGIBBONS Z-U Steel Firebox Boilers

Built for 15 lb. w.s.p.—A.S.M.E. Code.

Up-Draft Type..... 1800 to 35,000 sq. ft. steam

FITZGIBBONS R-Z-U Steel Firebox Boilers

The Z-U arranged for rear smoke outlet.

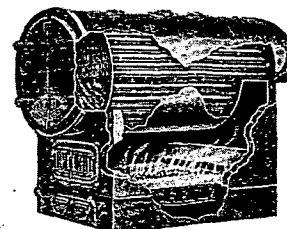
Built for 15 lb. w.s.p.—A.S.M.E. Code.

Up-Draft Type..... 1800 to 35,000 sq. ft. steam

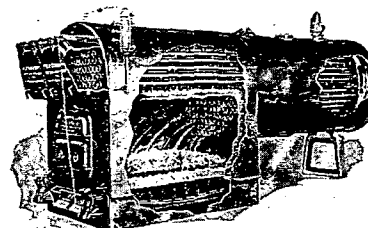
Smokeless Type..... 1800 to 35,000 sq. ft. steam

Oil, Gas, Stoker

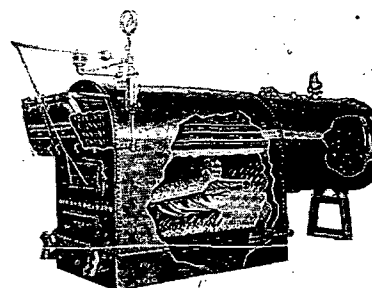
Type..... 2190 to 42,500 sq. ft. steam



R-Z-U



500 Series



600 and 800 Series

FITZGIBBONS "F" SERIES

Portable Riveted Firebox Boilers

Built for 100 lb. w.s.p.—A.S.M.E. Code.

Ratings, steam—1800 to 15,000 sq. ft.

FITZGIBBONS 500 SERIES

Portable Welded Firebox Boilers—
Return Tubular

Built for 15 lb. w.s.p.—A.S.M.E. Code.

Ratings, steam—3500 to 35,000 sq. ft.

FITZGIBBONS 700 AND "P" SERIES

Portable Riveted Firebox Boilers

700 Series built for 15 lb. w.s.p.—A.S.M.E. Code.

Ratings, steam—3500 to 35,000 sq. ft.

"P" Series built for 100 lb. w.s.p.—A.S.M.E. Code.

Ratings, horse power—25 to 250.

FITZGIBBONS 600 AND 800 SERIES Smokeless Down-Draft Riveted Firebox Boilers

Built for 15 to 100 lb. w.s.p.—A.S.M.E. Code.

Ratings, steam—3500 to 35,000 sq. ft.

*Descriptive Bulletins on any or all of above
boilers will be mailed on request.*

Farrar & Trefts

Incorporated

Buffalo, N. Y.

HEATING AND POWER BOILERS

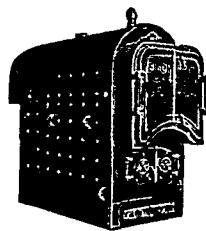
Bison Compact Boilers
 Bisonette Compact Boilers
 Firebox Return Tubular Boilers
 Firebox Locomotive Type Boilers
 Scotch Marine Type Boilers
 Vertical Boilers
 Horizontal Return Tubular Boilers
 Bison Two-Pass Return Tubular Boilers



Established 1864

STEEL PLATE CONSTRUCTION

Storage and Pressure Tanks
 Receivers, Welded or Riveted
 Steel Pipe, Welded or Riveted
 Buoys, Welded or Riveted
 Condensers and Kettles
 Smokestacks and Breechings
 Special Work in Stainless Steel,
 Everdur, Nickel, Aluminum
 or Monel Metal



The Bison Compact

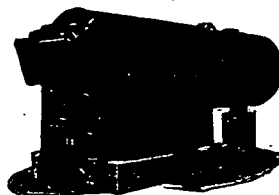
The F&T Bison Compact Welded Heating Boiler is more than just another boiler. It has been designed carefully so as to have a large furnace volume, the proper volume of water, just the right amount of steam liberating surface, the correct volume for steam storage and a balanced circulation. The result 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 1800 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 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 of riveted construction for power purposes at 100, 125 and 150 lb. working pressure in accordance with the A.S.M.E. Code. Sizes from 1800 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.

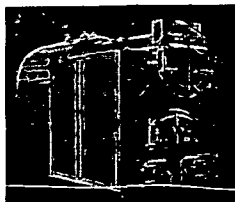


Firebox Return Tubular Boiler

The Bison Two-Pass Return Tubular Refractory Lined Boiler is favored by many Engineers because it Eliminates Water Legs, Flat Sides and Staybolts. This boiler is dependable for long years of continuous and economical operation. It stands up under heavy loads and provides that surplus of power so often needed.

This type of boiler is made of welded construction for 15 lb. working pressure and of riveted construction for 100, 125 and 150 lb. working pressure in accordance with the A.S.M.E. Code. Boilers are designed for ratings from 25 to 250 hp. and are inspected, tested, and stamped by a representative of a reliable insurance company before shipment.

They are furnished for hand firing with coal or for mechanical firing with oil, gas, or stoker.



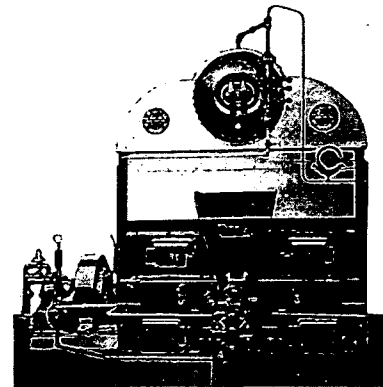
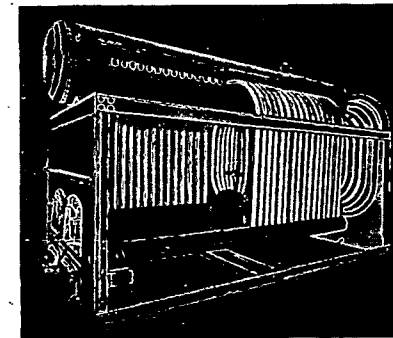
The Bison Two-Pass

E. Keeler Company

Williamsport, Pa.

ESTABLISHED 1864

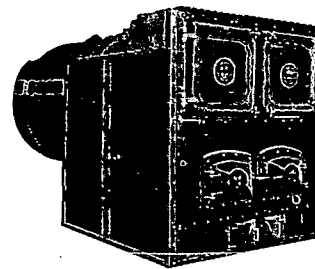
Riveted Steel Boilers For Heating and Power
 Steel Stacks, Breechings and Plate Fabrications

Keeler Type "CP" Water Tube Boiler
Equipped with Underfeed StokerKeeler Type "CP" Water Tube Boiler
Partly Assembled

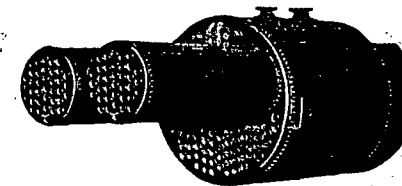
The Keeler Type "CP" Water Tube Boiler Embodies These Important Features

1. It is completely steel encased and insulated.
2. No brickwork required except for front wall and bridge wall.
3. The sides of the furnace are water cooled.
4. Clinkers cannot adhere to sides of furnace.
5. Furnace maintenance costs are reduced to a minimum.
6. Provides very large capacity in a given space.
7. Can be operated at high overloads without disturbing water level.
8. Has built in soot blower. Easy to clean interior or exterior.
9. Made in large range of sizes in pressures up to 450 lb.
10. Units as large as 250 hp can be shipped completely assembled.
11. Units larger than 250 hp are shipped knocked down.
12. Highly efficient with any method of firing.

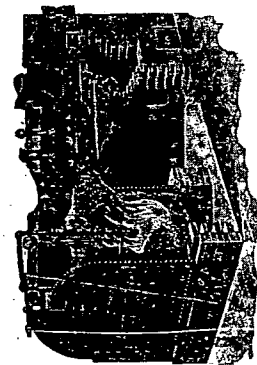
The E. Keeler Company manufactures Straight Tube Water Tube Boilers, both long drum and cross drum types, Curved Tube Boilers, both three drum and four drum types, Return Tubular and Double Duty Fire Tube Boilers for every power or heating requirement.



Keeler Double Duty Boiler

Keeler Double Duty Boiler
Before Encased

Bulletins of Any Type Sent on Request



KEWANEE FIREBOX BOILERS
Brick-set Type

KEWANEE BOILER CORPORATION

Kewanee, Illinois

BRANCHES IN 60 PRINCIPAL CITIES

Steel Heating and Power Boilers, Water Heating Garbage Burners, Tabasco Heaters and Tanks.

KEWANEE STEEL HEATING BOILERS

Kewanee offers the most complete line of Steel Boilers built today for heating any size or kind of building. There are more than 425 sizes in 18 distinct series, perfectly adapted for burning every fuel with the highest efficiencies.

Sixty-five years of intensive study and effort are back of the Kewanee Boiler designs. They are all constructed in the well-equipped factory at Kewanee, Illinois, conforming with the following Codes: A.S.M.E. for construction, S.H.B.I. for ratings.

The Kewanee series include:

RIVETED Firebox types, 510 ft. to 36,810 ft.

Brickset, Up-Draft and Smokeless

Portable, Up-Draft and Smokeless, two-pass tubes with front smoke outlet; or single-pass with rear smoke outlet.

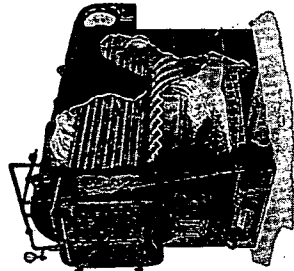
WELDED types, 410 ft. to 48,060 ft.

Type "C," Direct Draft, and Smokeless Arch. With corrugated crown sheet, Rear Smoke Outlet.

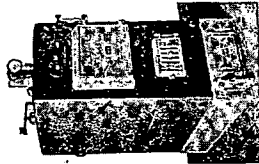
Residence Square Type "R," and Round Type "R."

SQUARE "R," Oil or Gas Jacketed Boiler with Steel Base. Nos. 88R1 to 88R9, Rated 901 ft. to 8984 ft.

KEWANEE SMOKELESS BOILERS
Portable Type



KEWANEE WELDED
TYPE C BOILERS



KEWANEE RESIDENCE
SQUARE TYPE R BOILERS

SPECIFICATIONS—BRICK-SET UP-DRAFT BOILER

Boiler No.	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Rated Capacity, Steam, sq. ft.	510	690	940	1240	1380	1830	2410	3340	4170	4300	5130	6070	7440	8770	10310	12080	12420	14670	16340	18610	
Burning Coal, Hand Fired	1440	1920	2230	2530	3060	3650	4230	5180	5950	6250	7180	8020	9720	11440	12840	12900	15160	16740	18840	20600	
Oil, Gas or Stoker Fired	2446	8306	43067	5308	53657	53659	36510	54210	42411	548410	54812	48313	54413	94416	460415	546418	60418	720418	814	720418	
Width and Length, in. x ft.	64	70	70	70	77	77	77	83	83	90	90	90	96	96	108	108	108	114	114	120	
Overall Height, in.	47	52	52	52	55	55	55	58 1/2	58 1/2	61	61	61	66	66	75	75	75	80	80	85 1/2	
Height of Water Line, in.																					
Approximate Weight:																					
Coal Burning	1900	2400	2700	2900	3300	3700	4200	5400	6000	6700	7300	8000	10600	11000	14400	16000	17600	19100	21700	23500	
Oil Burning	1800	2200	2400	2500	2900	3300	3700	4800	5300	6000	6500	7100	9400	10600	12900	14400	16000	17300	19800	21300	

Rated Capacity for Water Boiler is 60 per cent greater than Capacity for Steam Boiler.

SPECIFICATIONS—SMOKELESS DOWN-DRAFT BOILER

Boiler No.	307	308	309	310	311	312	313	314	315	316	317	318	319	320	321	322	323	324
Rated Capacity, Steam, sq. ft.	3390	4260	5020	6100	6950	7650	8290	9000	10100	10820	12270	14740	16860	18420	20410	21690	23040	24610
Burning Coal, Hand Fired	5050	5990	6650	7580	8430	9290	10070	10720	12260	13130	15500	17900	20460	22360	24800	26350	28000	29770
Oil, Gas or Stoker Fired	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9
Width and Length, in. x ft.	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9
Overall Height, in.	84	84	84	84	84	84	84	84	84	84	84	84	84	84	84	84	84	84
Height of Water Line, in.	71	71	71	71	71	71	71	71	71	71	71	71	71	71	71	71	71	71
Approximate Weight, lbs.	7800	8600	9200	10400	11100	11900	12600	13300	14000	14700	15400	16100	16800	17500	18200	18900	19600	20300
Oil Burning	7300	8000	8600	9200	9900	10600	11300	12000	12700	13400	14100	14800	15500	16200	16900	17600	18300	19000

SPECIFICATIONS—PORTABLE UP-DRAFT BOILER

Boiler No.	407	408	409	410	411	412	413	414	415	416	417	418	419	420	421	422	423	424
Rated Capacity, Steam, sq. ft.	2550	3340	4120	4300	5130	6070	7440	8770	10310	10820	12270	14740	16860	18420	20410	21690	23040	24610
Burning Coal, Hand Fired	4750	5270	6340	7040	7900	8760	9600	10640	11480	12260	13780	15800	17680	19850	22150	23570	25080	26770
Oil, Gas or Stoker Fired	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9
Width and Length, in. x ft.	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9	48-9
Overall Height, in.	84	84	84	84	84	84	84	84	84	84	84	84	84	84	84	84	84	84
Height of Water Line, in.	71	71	71	71	71	71	71	71	71	71	71	71	71	71	71	71	71	71
Approximate Weight, lbs.	6900	7400	8300	9000	9800	10600	11300	12000	12700	13400	14100	14800	15500	16200	16900	17600	18300	19000
Oil Burning	6400	6800	7600	8300	9000	9700	10400	11100	11800	12500	13200	13900	14600	15300	16000	16700	17400	18100

SPECIFICATIONS—TYPE "C" WELDED BOILER

STEEL BOILERS - 1500 C. WELDED DESIGN																						
*Boiler No.	749	750	751	752	753	754	755	756	757	758	759	760	761	762	763	764	765	766	767	768	769	770
Rated Cap., St. sq. ft.	2300	2690	3250	3630	4060	4550	5250	5950	6470	7000	8280	9610	10950	12110	13000	13880	14760	15640	16520	17400	18280	19160
Coal Hand Fired.....	3260	3940	4450	4950	5450	5950	6450	6950	7450	7950	9240	10580	11920	13000	13880	14760	15640	16520	17400	18280	19160	
Oil, Gas or Stoker.....	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	32-59	
Width and Length, in. x ft.	35 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	36 1/2	
Overall Height, in.	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	71 1/2	
Height of Water Line, in.	69	69	69	69	69	69	69	69	69	69	69	69	69	69	69	69	69	69	69	69	69	
Approximate Weight, lbs.	3300	3600	4200	4600	5000	5400	5800	6200	6600	7000	8200	9400	10600	11800	12600	13400	14200	15000	15800	16600	17400	
Oil Burning.....lbs.	4300	4700	5200	5500	6000	6500	7100	7500	7900	8300	10600	11700	12900	14000	15300	16600	17900	19200	20500	21800	23100	
Coal Burning.....lbs.	3800	4200	4600	5000	5400	5900	6300	6700	7100	7600	8900	9600	10600	11500	12700	13700	14800	15900	17000	18100	19200	

*Boiler Series 1749-1770 for oil, gas or stoker; Series 2749-2770 for anthracite; 7170 Series for stoker, 2890 ft. to 42060 ft.

SPECIFICATIONS—RESIDENCE TYPE "R" BOILER

Boiler No.	742	743	745	746	747	748
Rated Capacity, Steam, sq. ft.	790	1000	1350	1600	1780	1960
Burning Coal	840	1120	1470	1900	2160	2380
Oil, Gas or Stoker Fired	32-59	32-59	32-59	32-59	32-59	32-59
Width and Length, in. x ft.	32-59	32-59	32-59	32-59	32-59	32-59
Overall Height, in.	48	48	48	48	48	48
Height of Water Line, in.	35	35	35	35	35	35
Approximate Weight, lbs.	3300	3600	4200	4600	5000	5400
Oil Burning	3000	3300	3900	4300	4700	5100
Coal Burning	205	225	175	190	200	225
Jacket, Grated, Add.						

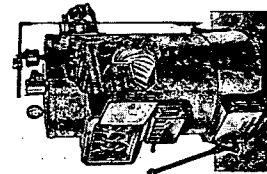
*Boiler Series 1742-1748 for oil, gas or stoker; Series 2742-2748 for anthracite.

Kewanee Indirect Hot Water Heating Coils for Type C, Square and Round R. Boilers; 125 sizes, 75 to 4500 gallons.

Kewanee Boiler Corp.

Boilers, Steel

Boilers, Steel



Kewanee Round
Type R Boiler

Choice of Round Jacket for any burner. Regular Jacket for enclosure with any gun type burner. Square Jacket for enclosure with any gun type burner outside jacket. Capacities of the Round "R," used as Direct Hot Water Heater are: 700 and 900 gallons per 3 hours 100 deg. raise.

Lookout Boiler & Mfg. Company



Successors to the
HEGGIE SIMPLEX BOILER CO.
Joliet, Illinois

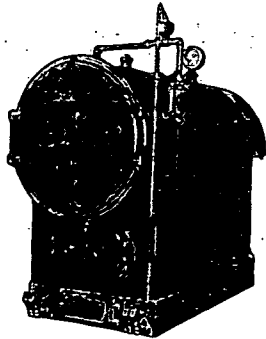
Chattanooga, Tennessee

Representatives in Principal Cities



REG. U. S. PAT. OFFICE

STANDARD SERIES LOOKOUT HEGGIE-SIMPLEX STEEL HEATING BOILERS



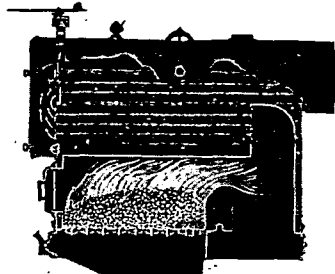
Steam Boiler Assembly

15-Lb. Steam
Pressure

Steel Plate
Welded Joints

A.S.M.E.
Construction

30-Lb. Water
Pressure



For Burning Any Solid Fuel

STANDARD SERIES SPECIFICATIONS

Smokeless Arch Type Direct Draft Type Mechanical Fired Type

Catalog Number Steam Boiler	Steam Rating Sq. Ft. (S.H.B.I.)	Catalog Number Steam Boiler	Steam Rating Sq. Ft. (S.H.B.I.)	Catalog Number Steam Boiler	Steam Rating Sq. Ft. (S.H.B.I.)	Heating Surface Sq. Ft. (S.H.B.I.)	Grate Area Square Feet	Width of Boiler, Inches	Length of Boiler, Overall, Inches	Height of Base, Inches	Height of Boiler on Base, Inches	Height of Water Line, on Base, Inches	Steam Liberating Area, Sq. Ft.	Water Capacity Full, Gallons	Size of Steam Outlet, Inches	Size of Return Opening, Inches	Diameter of Stack, Inches	Boiler Covering Required, Sq. Ft.
134A	1690	334A	1690	734A	2060	121	8.2	34	67 1/2	12	72	62 1/2	9.8	171	6	3	18	66
134C	2110	334C	2110	734C	2570	151	10.5	34	79 1/2	12	72	62 1/2	12.3	214	6	3	18	78
134E	2530	334E	2530	734E	3080	181	10.4	34	91 1/2	12	72	62 1/2	14.8	257	6	3	18	89
140A	3110	340A	3110	740A	3770	222	12.4	40	80 1/2	12	79	69 1/2	13.9	280	6	3	20	92
140B	3500	340B	3500	740B	4250	250	13.7	40	86 1/2	12	79	69 1/2	15.3	308	6	3	20	98
140D	4120	340D	4120	740D	5000	294	13.7	40	98 1/2	12	79	69 1/2	18.1	364	6	3	20	110
140E	4500	340E	4500	740E	5470	322	13.7	40	104 1/2	12	79	69 1/2	19.5	392	6	3	20	116
147A	5000	347A	5000	747A	6080	358	14.7	47	101 1/2	12	86	72 1/2	22.3	476	8	4	22	125
147B	5490	347B	5490	747B	6660	392	16.5	47	113 1/2	12	86	72 1/2	24.0	514	8	4	22	133
147C	5960	347C	5960	747C	7240	426	16.5	47	125 1/2	12	86	72 1/2	25.8	552	8	4	22	141
152A	6500	352A	6500	752A	7890	464	18.6	52	103 1/2	12	91	78 1/2	23.3	551	8	4	24	136
152B	7000	352B	7000	752B	8500	500	18.6	52	109 1/2	12	91	78 1/2	25.1	596	8	4	24	144
152C	7520	352C	7520	752C	9130	537	20.7	52	115 1/2	12	91	78 1/2	26.9	641	8	4	24	152
152D	8040	352D	8040	752D	9760	574	20.7	52	121 1/2	12	91	78 1/2	28.8	686	8	4	24	160
157A	9100	357A	9100	757A	11050	650	22.9	57	115 1/2	12	97	81 1/2	30.9	753	8	4	27	167
157C	10420	357C	10420	757C	12650	744	25.2	57	127 1/2	12	97	81 1/2	35.1	858	8	4	27	184
157D	11070	357D	11070	757D	13450	791	25.2	57	133 1/2	12	97	81 1/2	37.2	910	8	4	27	193
163A	12500	363A	12500	763A	15180	894	28.1	63	123 1/2	15	108	93	34.9	953	10	5	30	197
163B	13360	363B	13360	763B	16220	954	28.1	63	129 1/2	15	108	93	37.1	1014	10	5	30	206
163C	14200	363C	14200	763C	17240	1014	28.1	63	135 1/2	15	108	93	39.3	1076	10	5	30	215
163D	15000	363D	15000	763D	18220	1074	30.6	63	141 1/2	15	108	93	41.5	1137	10	5	33	224
170A	16020	370A	16020	770A	19450	1144	31.0	70	129 1/2	15	116	96 1/2	43.6	1269	10	5	33	230
170B	17020	370B	17020	770B	20670	1216	31.0	70	135 1/2	15	116	96 1/2	46.2	1347	10	5	33	240
170C	18030	370C	18030	770C	21900	1288	33.9	70	141 1/2	15	116	96 1/2	48.8	1423	10	5	36	250
178A	20000	378A	20000	778A	24290	1429	35.0	78	135 1/2	15	126	106 1/2	47.9	1524	10	5	36	258
178C	22390	378C	22390	778C	27180	1599	38.2	78	147 1/2	15	126	106 1/2	53.9	1751	10	5	36	281
184A	25000	384A	25000	784A	30360	1786	38.0	84	141 1/2	15	132	113	52.9	1831	10	5	40	291
184C	28010	384C	28010	784C	34020	2001	41.3	84	153 1/2	15	132	113	58.9	2042	10	5	40	314
184E	31010	384E	31010	784E	37660	2215	44.8	84	165 1/2	15	132	113	64.9	2255	10	5	40	337
184G	34010	384G	34010	784G	41290	2429	44.8	84	177 1/2	15	132	113	70.9	2467	10	5	42	361

To determine S.H.B.I. Ratings of Hot Water Heating Boilers, add 60 per cent to corresponding steam boiler rating.

RESIDENCE SERIES

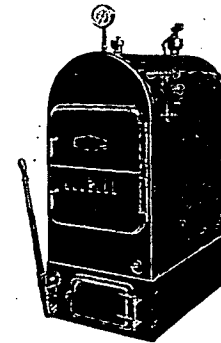
LOOKOUT HEGGIE-SIMPLEX STEEL HEATING BOILERS

Steel Plate
15-Lb. Steam Working Pressure

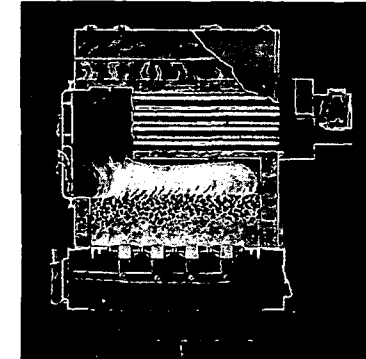
Welded Joints

A.S.M.E. Construction
30-Lb. Water Working Pressure

Direct Draft Type for Hand Firing any Solid Fuel
Mechanical Fired Type for Oil, Gas, or Stoker



Steam Boiler Assembly with Jacket



For Burning Any Solid Fuel

RESIDENCE SERIES (Double Pass—2 In. Tubes) SPECIFICATIONS

Jacketed and without Jacket

Direct Draft Type

Mechanical Fired Type

Catalog Number Steam Boiler	Steam Rating Sq. Ft. (S.H.B.I.)	Catalog Number Steam Boiler	Steam Rating Sq. Ft. (S.H.B.I.)	Heating Surface Sq. Ft. (S.H.B.I.)	Grate Area Square Feet	Width of Boiler, Inches	Length of Boiler, Overall, Inches	Height of Base, Inches	Height of Boiler on Base, Inches	Height of Water Line, on Base, Inches	Steam Liberating Area, Sq. Ft.	Water Capacity Full, Gallons	Steam Space Cu. Ft.	Size of Steam Outlet, Inches	Size of Return Opening, Inches	Size of Water Heater Opening Inches	Diameter of Stack, Inches	Height of Stack, Feet
2SJ1	370	2SCJ1	540	32	2.6	24	34 1/2	12	56	48	3.8	37	1.8	3	3	1 1/2	10	30
2SJ2	510	2SCJ2	680	40	3.5	24	40 1/2	12	56	48	4.8	48	2.3	3	3	1 1/2	10	30
2SJ3	670	2SCJ3	820	48	4.3	24	45 1/2	12	56	48	5.8	60	2.8	3	3	1 1/2	10	30
2SJ4	800	2SCJ4	970	57	5.1	24	53 1/2	12	56	48	6.8	71	3.3	3	3	1 1/2	10	30
2SJ5	950	2SCJ5	1160	68	5.4	27	47 1/2	12	65	55 1/2	6.6	103	3.8	4	4	1 1/2	12	40
2SJ6	1110	2SCJ6	1340	79	6.0	27	53 1/2	12	65	55 1/2	7.7	122	4.5	4	4	1 1/2	12	40
2SJ7	1300	2SCJ7	1580	93	7.0	27	60 1/2	12	65	55 1/2	8.9	143	5.1	4	4	1 1/2	12	40
2SJ8	1480	2SCJ8	1800	106	7.8	27	66 1/2	12	65	55 1/2	10.0	163	5.8	4	4	1 1/2	12	40
2SJ9	1620	2SCJ9	1970	116	7.9	30	60 1/2	12	72	61	10.1	175	6.7	5	5	1 1/2	14	50
2SJ10	1830	2SCJ10	2230	131	9.0	30	66 1/2	12	72	61	11.3	199	7.6	5	5	1 1/2	14	50

To determine S.H.B.I. Ratings of Hot Water Heating Boilers, add 60 per cent to corresponding steam boiler ratings.

In addition to the STANDARD SERIES and RESIDENCE SERIES of Steel Heating Boilers, we make the following:

HIGH FIREBOX SERIES—Mechanical Fired Type: Specifications given in Bulletin No. 112.

P. O. SERIES PORTABLE FIRE BOX BOILERS—(Single Pass)—Down-draft Water-Grate Type—Direct Draft Type—Mechanical Fired Type. Specifications given in Bulletin No. 117.

F.P.O. SERIES PORTABLE FIREBOX BOILERS—(Double Pass)—Down-draft Water-Grate Type—Direct Draft Type—Mechanical Fired Type. Specifications given in Bulletin No. 118.

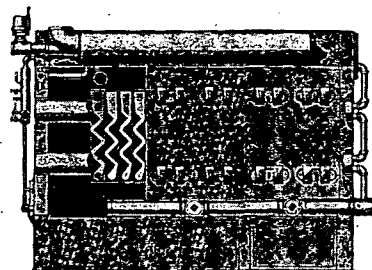
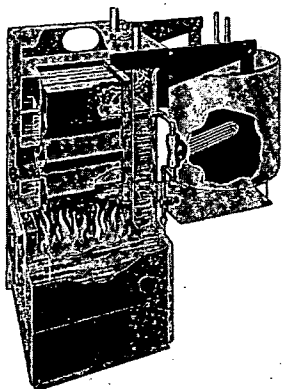
REFUSE BURNING WATER HEATERS—Direct Draft Type. Specifications given in Bulletin No. 119.

Waterfilm Boilers

Incorporated

154 Ogden Avenue, Jersey City, N. J., U. S. A.

BOILERS DESIGNED AND BUILT FOR AUTOMATIC FIRING



Send for full information today

DE LUXE SERIES

Boiler No.	2	3	4	5
Recommended Load Square Feet E. D. R. Pickup included in these ratings.	Steam 500' Water 800'	Steam 800' Water 1280'	Steam 1100' Water 1760'	Steam 1400' Water 2240'
Guaranteed B.T.U.	120,000	192,000	264,000	336,000
Maximum Firing Rate G.P.H.	1.65 gal to 2 gal	2 gal to 2 1/2 gal	2 1/2 gal to 3 gal	3 1/2 gal to 4 gal

DE LUXE SERIES completely jacketed with either short to cover boiler only, or long to cover boiler, burner and domestic hot water tank.

SECTIONAL SERIES

Single Series

No.	Recommended Load E.D.R.		Guaranteed B.T.U.	No.	Recommended Load E.D.R.		Guaranteed B.T.U.
	Steam	Water			Steam	Water	
1-5	1,250	2,000	330,000	1-12	3,000	4,800	750,000
1-6	1,500	2,400	390,000	1-13	3,250	5,200	810,000
1-7	1,750	2,800	450,000	1-14	3,500	5,600	870,000
1-8	2,000	3,200	510,000	1-15	3,750	6,000	930,000
1-9	2,250	3,600	570,000	1-16	4,000	6,400	990,000
1-10	2,500	4,000	630,000	1-17	4,250	6,800	1,050,000
1-11	2,750	4,400	690,000	1-18	4,500	7,200	1,110,000

Double Series

2-8	4,000	6,400	990,000	2-25	12,500	20,000	3,030,000
2-9	4,500	7,200	1,110,000	2-26	13,000	20,800	3,150,000
2-10	5,000	8,000	1,230,000	2-27	13,500	21,600	3,270,000
2-11	5,500	8,800	1,350,000	2-28	14,000	22,400	3,390,000
2-12	6,000	9,600	1,470,000	2-29	14,500	23,200	3,510,000
2-13	6,500	10,400	1,590,000	2-30	15,000	24,000	3,630,000
2-14	7,000	11,200	1,710,000	2-31	15,500	24,800	3,750,000
2-15	7,500	12,000	1,830,000	2-32	16,000	25,600	3,870,000
2-16	8,000	12,800	1,950,000	2-33	16,500	26,400	3,990,000
2-17	8,500	13,600	2,070,000	2-34	17,000	27,200	4,110,000
2-18	9,000	14,400	2,190,000	2-35	17,500	28,000	4,230,000
2-19	9,500	15,200	2,310,000	2-36	18,000	28,800	4,350,000
2-20	10,000	16,000	2,430,000	2-37	18,500	29,600	4,470,000
2-21	10,500	16,800	2,550,000	2-38	19,000	30,400	4,590,000
2-22	11,000	17,600	2,670,000	2-39	19,500	31,200	4,710,000
2-23	11,500	18,400	2,790,000	2-40	20,000	32,000	4,830,000
2-24	12,000	19,200	2,910,000	2-41	20,500	32,800	4,950,000

International Heating & Ventilating Exposition

THE AIR CONDITIONING EXPOSITION

Permanent Address—Grand Central Palace, New York, N. Y.

EXPOSITIONS HELD

The first in Philadelphia, 1930.

The second in Cleveland, 1932.

The third in New York, 1934.

The fourth Exposition will occur in Chicago, January 27-31, 1936.

Subsequent Expositions will be held on alternate, even numbered years.

These are held coincident with the Annual Meeting 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.

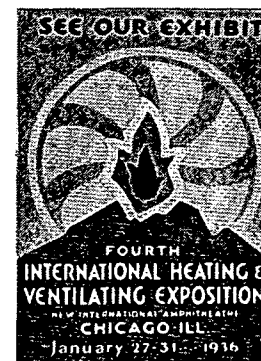
EXHIBITORS

Comprise leading firms in each phase of the industry: number has varied from 150 to 275 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.

- The COMBUSTION Group:** Furnaces, burners (oil and gas), grates, stokers, boilers, radiators (various types), refractories and auxiliaries.
- The HYDRAULIC Group:** Water feeders, water heaters, pumps, traps, valves, piping, fittings, expansion joints, pipe hangers, etc.
- The STEAM HEATING Group:** Vapor heating and steam specialties.
- 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.
- CONTROL Group:** Instruments of precision for indicating, controlling or recording temperature, pressure, volume, time, flow, draft or any other function to be measured.
- REFRIGERATING:** Compressors, condensers, cooling apparatus, contingent apparatus and refrigerants for homes, factories, railroads, etc.
- CENTRAL HEATING:** Apparatus and materials especially designed or adapted to the use of central heating and



central heating station supplies.

- INSULATING:** Structural insulators (refractory and cellulose materials), asbestos, magnesia clays and combinations thereof, pipe and conduit covering, etc., weather-stripping, etc.

- The MISCELLANEOUS Group:**

Electric Heaters, boiler and pipe repair alloys, liquids and compounds, etc., which are not included above.

- MACHINERY AND GENERAL EQUIPMENT.**

VISITOR ATTENDANCE

Comprises a registered attendance invited to the exposition and includes:

(Figures are 1934 analysis)

INDUSTRIES	
Governmental.....	186
Distributional Channels	
Contractors, Dealers, Jobbers, Supply Houses, 32 classifications.....	5998
Home Owners.....	414
Industrial Users, 49 classifications.....	3428
Professional and Service Organizations, 23 classifications.....	2017
Public Utilities.....	922
Real Estate Management and Operation, 10 classifications.....	581
Educational Institutions.....	303
Miscellaneous.....	1043
TOTAL.....	15,256

OCCUPATIONS

Executive (44 titles).....	6850
Construction (16 titles and trades).....	1022
Operation (44 titles and trades).....	2014
Technical (64 titles).....	2044
Not Classified including Educators, Publishers, Home Owners, etc.....	2222
TOTAL.....	15,256

These visitors came from 704 cities and towns in 37 States of the United States and 25 cities and towns in 11 foreign countries.

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 biennially brings together the research developments and improvements in equipment and materials.

Delco Appliance Corporation

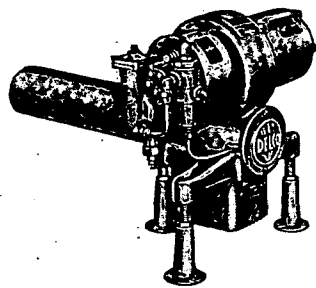
Subsidiary of General Motors

Rochester, N. Y.

DELCO-HEAT

Delco-Heat Conversion-Burners

Delco-Heat represents a distinct achievement in the automatic heating industry. It is backed by the years of study, research and experience gained in perfecting the combustion of liquid fuels. Delco-Heat incorporates many new, practical and economical features in design, operation and manufacture. Delco-Heat possesses all the essential requirements which are expected in modern heating apparatus. The Delco-Heat Models listed and illustrated, furnish a range of burner sizes which are adapted to the varied requirements of the small home, the large apartment house or commercial and industrial installations. The several sizes differ somewhat in design due to service requirements but each size is built of the same quality material and has the same skilled workmanship in its construction.



Delco-Heat Models DX and DX-12

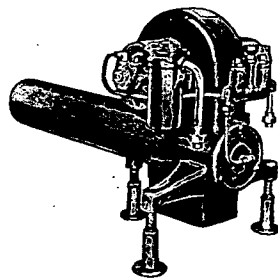
Delco-Heat Model DX: Capacity—Steam, 900 sq ft; Hot Water, 1440 sq ft. Efficient burner for small home or large residential use. It burns efficiently low cost domestic fuel oil. It is fully equipped with all necessary controls for complete automatic operation and carries the Underwriters' label. Ignition is by intermittent electric spark.

Delco-Heat Model D-12: Steam, 2330 sq ft; Hot Water, 3725 sq ft. Is intended

for larger installations and is appropriate for small apartments and stores and commercial buildings.

Delco-Heat Burners for Commercial Use

Delco-Heat equipment is economical to install and operate, simple in construction and design, quiet in use and has years of proven performance in commercial buildings of every character. If you have a special heating problem or require specific information not found here, there is a Delco-Heat representative nearby. The Engineering Department is also at your service and if you so prefer, complete specifications and recommendations will be made, if the plans and special information desired are sent to the Sales Engineering Division, Delco Appliance Corp., Rochester, N. Y.



Delco-Heat Models D-34 and D-44

Model D-34: Capacity—Steam, 5400 sq ft; Hot Water, 8600 sq ft. Burners adapted for commercial and industrial installations. Burns either No. 3 or No. 4 Fuel Oil with capacity of from 7.0 to 16.5 gals per hour. Ignition is by intermittent spark.

Model D-44: Capacity—Steam, 10,000 sq ft; Hot Water, 16,000 sq ft. It burns either No. 3 or No. 4 Fuel Oil with capacity of from 15.0 to 30.0 gals per hour. Ignition is by intermittent spark.

See also Pages 835 and 876

Electrol Incorporated

FINE OIL HEATING EQUIPMENT EXCLUSIVELY SINCE 1918

934 Main Avenue, Clifton, N. J.

Conversion Burners, Boiler-Burner Units, Air Conditioners



Electrol-Kewanee Heating Unit

A complete boiler-burner unit consisting of (1) exclusive design, horizontal, welded steel firetube boiler; (2) Electrol oil burner mechanism; and (3) domestic water heater. Induced draft. Guaranteed minimum 80 per cent over-all efficiency. Low stack temperature. Integral flame, low water and limit controls. Attractive sound-insulated jacket. Burns heavy furnace oil. Simple, automatic.

Unit	Cap. E.D.R. Steam	Gal. Oil per Hour	Gals. D.H.W. Three Hours	Over-all Dimensions, In.		
				L	W	H
E-K 5	750	1.6	50	47 1/2	34	55 1/4
E-K 10	1250	2.7	80	60 1/2	34	55 1/4
E-K 15	1750	3.75	100	75	34	55 1/4

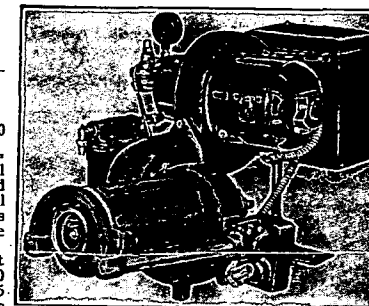
Electrol Oil Burner

Famous for nearly 20 years for dependable, economical operation in any boiler or furnace. Curved air chamber produces rotating air and ball-shaped flame. Exclusive Master Control safety. Continuous electric ignition. Cut-off valve at nozzle. New parts fit any existing models—eliminating obsolescence. Burns heavy furnace oil. Simple, fully automatic. Summer-Winter Dom. H. W. with indirect heater.

Model Sizes	Gal. Oil per Hour		Capacity E.D.R. Steam
	Min.	Max.	
TC	1.3	3 1/2	200-1100
TJ	1.3	2 1/2	200-800*
TU	2	8 1/2	600-2800
TM	12	24	2500-10,000

Electrol Conversion Oil Burners should be selected to heat a boiler to its full rated capacity, regardless of the load that may be connected to the boiler.

*Standard Equipment with the Model TJ is 1140 rpm motor. With 1725 rpm motor, capacity is same as model TC.



Electrol Direct-Fired Fan Furnace Unit

Provides either or both complete winter and summer air conditioning. Heating, humidification, air cleansing and circulation, plus cooling and dehumidification. Tested efficient design; competitively priced. Sizes to handle 112,000 to 254,000 Btu—1,000 to 2,500 cfm.

Electrol-American Heating Unit

A complete boiler-burner unit specially designed for small homes. 5- or 7-section, cast iron boiler of special construction, oil burning mechanism and domestic water heater all designed and built as one unit. Integral flame, low water and limit controls. Positive air adjustment at burner head. Burns heavy furnace oil. Simple, quiet, automatic. 80 per cent over-all efficiency.

Unit	Cap. E.D.R. Steam	Gal. Oil per Hour	Gals. D.H.W. Three Hours	Over-all Dimensions, In.		
				L	W	H
E-A 4	600	1.27	50	39 1/2	26	51 1/2
E-A 4A	825	1.74	50	47 1/2	26	51 1/2

Electrol Air Conditioner

Specially balanced for fully automatic operation with Electrol boiler-burner units or steam or hot water boilers with conversion burners. Provides heating, humidification, ventilation, air cleansing and circulation and, if required, summer cooling and dehumidification. Built in three sizes. Furnishes 6 to 7 air changes per hour. Washable cloth filters. Low velocity air circulation, eliminating drafts.

Norge Division

Borg-Warner Corporation

606-670 E. Woodbridge Street, Detroit, Michigan

Norge Whirlator Oil Burner

Service through franchised distributors and dealers throughout the United States.

Norge Whirlator Oil Burner is a "gun-type" burner built in six sizes, Model 8, handles 800 sq ft of steam radiation or its equivalent; Model N-18, handles 1800 sq ft; Model N-28, 2800 sq ft; Model N-38, handles 3800 sq ft; Model N-68, handles 6800 sq ft; Model N-88, handles 8800 sq ft.

The "Whirlator" is a patented design of multiple vanes, completely rotating entire mass of air in a smooth even movement, perfect ad-mixture with oil in correct proportion for high combustion efficiency.

Nozzle of latest improved design will atomize 28-30 Baume, known as No. 3 A.O.B.A. specifications, or lighter oils, with a fog-like spray.

Motors—(Model N-8) Split Phase, 110 volts, 60 cycle, alternating current, 1750 rpm. Small current consumption. Model N-8, $\frac{1}{10}$ hp motor—Model N-18, $\frac{1}{8}$ hp, Model N-28, $\frac{1}{4}$ hp.

Larger models are powered in proportion.

Ignition—Two-pole transformer, hot spark, designed for heavy duty, minimum

temperature rise, low input, shielded secondaries, built-in radio interference eliminator. Tested for maximum breakdown of 15,000 volts.

Controls—Minneapolis-Honeywell Series 10, low voltage room thermostat, type R-117-3 Protectoclay (stock mounted) or combustion safety device and boiler or limit control (either high or low voltage, as desired).

Pump—Rotary type, connected directly through flexible spring coupling to motor shaft. Simple and quiet.

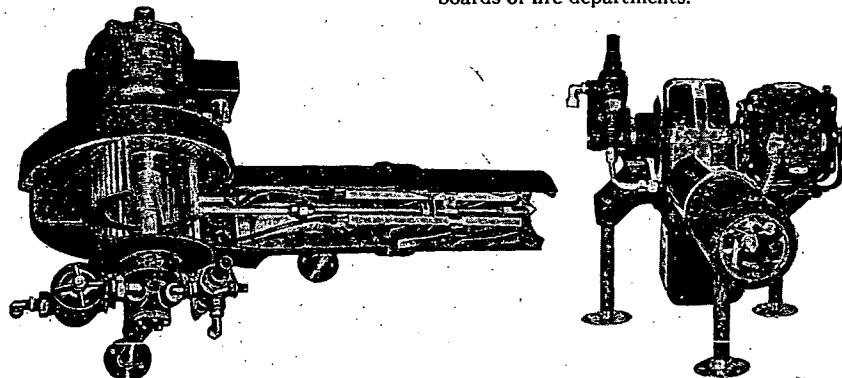
Fan—Fabricated light aluminum, strong and accurately balanced. Mounted on motor shaft with no intermediate connections.

Strainers—A series of two strainers used, with ample capacity to remove dirt and sediment from oil.

Accessibility—Fan or blower housing, and inspection cover in one light-weight casting. When removed exposes entire interior mechanism.

Legs—Three legs provide a sturdy mounting with easy adjustability.

Approvals—Listed as standard by Underwriters' Laboratories, and approved by all leading state and municipal safety boards of fire departments.



See also Pages 830 and 847

Williams Oil-O-Matic Heating Corporation
Bloomington, IllinoisManufacturers of Automatic and Manually Controlled Fuel Oil Burners
Service to Architects and Builders

CHICAGO, ILL., 641 N. Michigan Avenue

NEW YORK, N. Y., 1231 Graybar Building

For Williams Air Conditioning Equipment and
Ice-O-Matic Refrigeration Equipment, see File Index

A Complete Line

Williams Oil-O-Matic offers a complete line of oil heating equipment; five Oil-O-Matic burner models—a

genuine Williams Oil-O-Matic of ideal capacity for every size and type of house, for every apartment, public building or commercial structure.

New complete boiler-burner units; a new and revolutionary type of oil-burning automatic water heater; and a new oil-burning range burner for heavy duty ranges are available.

Heating Capacities of the Various Williams Oil-O-Matic Burners

Model	Domestic Shipping Weight	Length	Width	Height	H.P.	R.P.M.	Gals. Fuel Oil for Operating Hour	
							Min.	Max.
K-150	115	30"	14 1/2"	15 1/2"	1/10	1750	1/2	1 1/2
K-3	145	31 1/2"	15 1/2"	17"	1/10	1750	1	3
K-7	175	33"	22 1/2"	18 1/2"	1/5	1750	3	7
J-1800	255	45"	23"	25 1/2"	1/2	1750	7	15
JJ	295	50"	33"	22"	1	1750	15	25

Standard Draft Pipe 18 in. 12 in. length Draft Pipe optional.

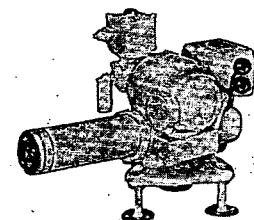
Standard electric current is 60-cycle, 110 volt. In case odd frequency motors are used, the maximum capacity of the burner will be reduced in proportion to the R.P.M. of the motor. The minimum capacity remains the same.

Water Heaters

WHA*	600	58"	23"	28"	1/10	1750		1/2
WHB†	885	75"	23"	28"	1/10	1750	1 1/2	1.4
WHC‡	1380	88"	29"	34"	1/10	1750	1	1 3/4

*Output: 90 F rise, 60 gal. per hour. †Output: 90 F rise, 120 gal. per hour. ‡Output: 90 F rise, 210 gal. per hour.

How to Decide Size of Burner



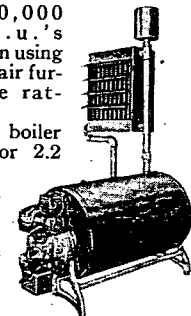
its equivalent.
480 sq. ft. of hot water radiation or
its equivalent.

For low-pressure Domestic Boilers, one gallon of fuel oil per hour (140,000 B.t.u.'s) is required for approximately:
300 sq. ft. of steam radiation or

See also Page 848

WILLIAMS
OIL-O-MATIC
HEATING

70,000
B.t.u.'s
when using
hot air fur-
nace rat-



ings.
24 sq. ft. steam boiler
heating surface (or 2.2
H.P.).

For exact detail
data, see Oil-O-
Matic Installation
and Service
Manual.

Oil-O-Matic
Hot Water Supply

A new efficient hot water supply unit made in 3 sizes covering ordinary home use, large home or commercial use, and heavy duty requirements. The entire apparatus—including Oil-O-Matic oil burner, combustion chamber, water reservoir, and all automatic controls—is combined in one compact unit. Tank being horizontal instead of vertical, permits the use of a unique triple flame travel.

Water heater is also available with one (or more) room heating radiators as illustrated.

Horizontal Boiler-Burner Unit

Horizontal boiler-burner unit design greatly increases efficiency. Built with A.S.M.E. code tanks, galvanized or not galvanized. Complete with full automatic controls. Suitable for hot water, vapor or vacuum heating plants, and the largest size unit suitable for steam plants.

Underwriters' Listing

Oil-O-Matic burners are listed as standard by the National Board of Fire Underwriters' to burn oils conforming to commercial, standard specifications for Nos. 1, 2 and 3 fuel oil having a viscosity of not more than 70 sec. (Saybolt-Universal at 100 F). Each burner carries the Underwriters' label.

Range Burners

New type Oil-O-Matic range burner burns low-cost fuel oil. Economical and efficient. Designed for heavy duty ranges.

Engineering Service

is available to architects. See A.I.A., File No. 30-6-1.

Iron Fireman Manufacturing Company

Automatic Coal Burners

Portland, Oregon

Factories: PORTLAND, ORE.; CLEVELAND, OHIO; TORONTO, CANADA

Retail Branches or Subsidiaries

CHICAGO, ILL. MILWAUKEE, WIS. ST. LOUIS, MO. NEW YORK, N. Y. MONTREAL, CAN.

Dealers in Principal Cities and Towns in the United States and Canada

Representation in numerous foreign countries.

IRON FIREMAN Automatic Coal Burners

"Forced Underfiring" Principle—Iron Fireman "Forced Underfiring" is based on the scientific principle of feeding fuel to the fire from below, under forced draft. From the conveyor screw coal enters the firebox under the fire and is gradually forced upward into the flame. As the coal approaches the fire, it is gradually heated. The volatile gases are distilled off in the presence of an excess of oxygen and are thoroughly ignited while passing through the incandescent fuel bed. This insures complete combustion. The ash is used into clinkers which are easily removed.

Advantages—Iron Fireman saves money and increases heating plant efficiency in four major ways: (1) Cuts fuel costs; (2) Reduces labor costs; (3) Provides steady, even heat or power; (4) Eliminates the smoke nuisance.

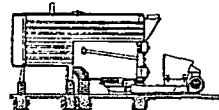
Installation and Sizes—Iron Fireman is made in a range of hopper and bin-feed sizes for commercial heating and power boilers up to 300 b.h.p. and also for homes. It can be installed quickly in practically any solid fuel boiler or furnace, old or new. Machines are shipped complete from the factory. All parts are standard and interchangeable.

Features of Design and Construction—Construction and operation of the Iron Fireman are characterized by sim-

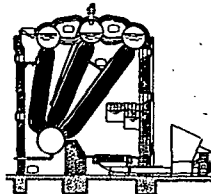


plicity throughout. Outstanding features of design and construction are: (1) Pressed steel construction. (2) Special patented transmission—three speeds and neutral. Gears run in bath of oil. (3) Electric motor—standard make. (4) V-belt drive. (5) Safety shear pin protects mechanism from damage. (6) Quiet ball bearing fan supplying forced draft to fire. (7) Automatic fire banking damper—conserves fuel and holds fire in proper condition when stoker is idle. (8) Positive pneumatic fume eliminator—an auxiliary air supply that insures positive movement of all gases through the fire. (9) Volumeter that supplies exactly the amount of air needed for perfect combustion regardless of fuel bed conditions or type of coal used. (10) Sectional retort especially designed to allow for heat expansion. (11) Dead plates of heavy iron and ribbed. (12) Sectional, self-cleaning tuyere blocks. (13) Conveyor screw cast of special Iron Fireman alloy steel from one-piece pattern. (14) Automatic electric controls designed for and used exclusively on Iron Fireman.

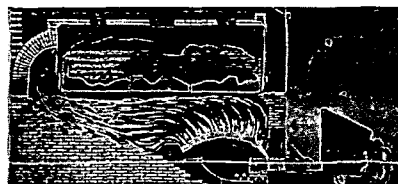
Automatic Controls—Iron Fireman starts and stops at the command of sensitive, accurate automatic controls. Directing controls govern stoker operation according to demands of time, tempera-



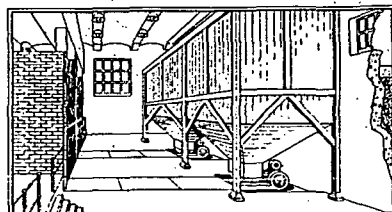
Typical Installation Down Draft Firebox Boiler



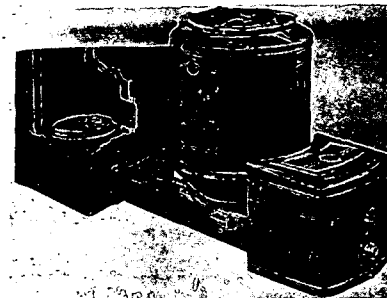
Typical Installation Four Drum Water Tube Boiler



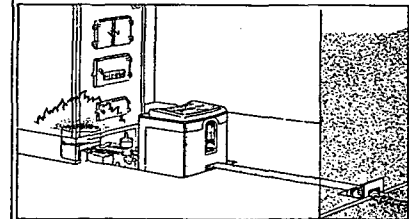
Iron Fireman in Operation in Horizontal Return Tubular Boiler, Low Bridge Wall



Multiple Installation of Bunker—Feed Stokers



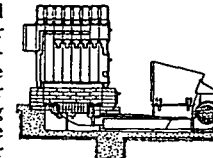
Domestic Installation—Hopper Model



Domestic Installation—Bin-Feed Model

ture, or pressure. An example of the superiority of Iron Fireman directing controls is the "Syncro-Stat" which provides automatic control of day and night temperature. Other directing controls include pressure regulators, hot water and furnace regulators, and the "Timetactor," a device which runs the Iron Fireman during predetermined intervals in order to keep the fire alive during mild weather.

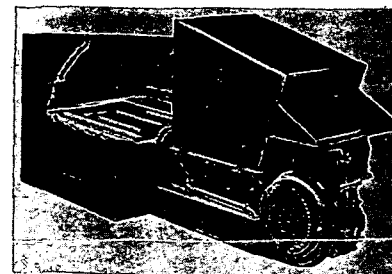
The most important unit of the operating control system is the motor-driven relay switch. This device starts and stops the stoker motor at the command of the Syncro-Stat or other directing controls. In the case of the larger stokers a magnetic operating switch works in conjunction with the relay switch.



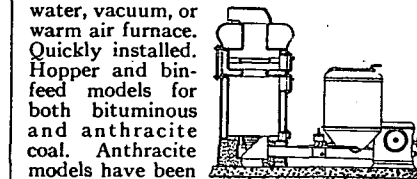
Typical Installation Cast Iron Boiler

Iron Fireman for Homes—The Iron Fireman residential model employs "Forced Underfiring" principle the same as larger machines, with simplified operation.

Can be recommended for any steam, hot



Commercial Model—For Heat or Power



Typical Installation Domestic Boiler

ENGINEERING SERVICE

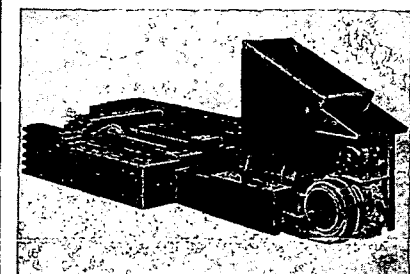
The Iron Fireman organization is nationwide. Trained men—backed by one of the largest manufacturing organizations in the field—are at your service to help you with the experience and practical heating information gained through servicing thousands of boiler rooms and heating plants in all parts of the country.

Any Iron Fireman engineer will gladly call and submit any additional information requested.

CATALOG AND INFORMATION

Catalogs give full information about the Iron Fireman. Descriptive folders give special data about installation in particular types of industries and in homes.

Secure them by addressing the factory or any Iron Fireman representative.



Industrial "Poweram" Model—For Heat or Power

Henry Furnace & Foundry Co.

Cleveland, Ohio Miles Furnace Fan Division

MILES FORCED DRAFT AND CONTROL UNIT

MILES COMBUSTION, PRESSURE AND TEMPERATURE CONTROL UNIT

Reduces costs, maintains uniform steam pressure and temperature and increases efficiency. Permits burning of cheaper grades of coal, slack, run of mine, buckwheat and rice.

Miles Combustion, Pressure and Temperature Control Unit is first, a forced draft unit, consisting of motor driven fan and pivoted damper actuated by steam pressure in the boiler, made in seven sizes to deliver air volumes as required and capable of overcoming various amounts of resistance. Second, it is an air pressure and control unit which effects an instant and minute control of the air volume and static pressure in response to pre-determined steam pressure or water temperature by means of the Miles Patented Air and Pressure Control Unit.

A pivoted shut-off damper yields to air pressure from the fan to an open position, and yields also to pressure due to steam or water temperature to a closed position. These counteracting pressures actuate the damper to the exact position which controls air volume and pressure under the grates and this in turn controls the fire.

So delicate is it in operation that the steam pressure will maintain a steady heating load with a variation in pressure of not to exceed two ounces over an extended period.



When a Miles Combustion, Pressure and Temperature Control Unit is installed, the chimney is eliminated as a draft producing factor. It becomes merely a vehicle for carrying off the products of combustion.

Automatic control and mechanical force supply the correct amount of air for any and every condition. The Miles Control Unit may be installed on either side of boiler or furnace or in the rear.

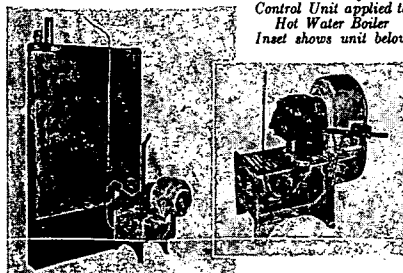
Only the Miles forced draft control has the patented graduated air supply feature which coordinates steam pressure and air volume to produce the required combustion. This unique device is in action at all times when heat is required, and supplies air constantly so that ashes are kept cooled below fusing point—hence no clinkers. Write for complete information.

DATA SHEET

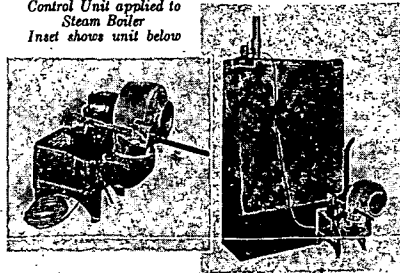
Miles Draft Control Unit for Steam and Hot Water Boilers and Warm Air Furnaces

No. of Unit	H.P. of Motor	Steam Radiation		Warm Air Furnace Btu Capacity
		Anthracite Buckwheat Bit. Nut	Anthracite Rice Bit. Slack	
Self-contained				
3-B	1/50	670	500	167,000
3-HW	1/15	1330	1000	332,000
4-A	1/20	1850	1380	462,000
4-B	1/10	3500	2625+	875,000
5-A	1/10	4150	3100	1,040,000
5-B	1/3	8000	6000+	2,000,000
6-A	1/4	9000	7250	2,250,000
6-B	1/2	13300	10,000+	3,320,000

Miles Combustion, Pressure and Temperature Control Unit applied to Hot Water Boiler
Inset shows unit below



Miles Combustion, Pressure and Temperature Control Unit applied to Steam Boiler
Inset shows unit below



See also Page 843

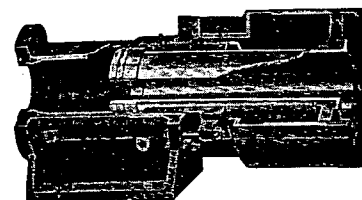
Over Fifty Years In Business

AMERICAN DISTRICT STEAM COMPANY
NORTH TONAWANDA, N.Y.

Branches and Agents in Principal Cities

STEAM DISTRIBUTION EQUIPMENT

ADSCO Piston-Ring Expansion Joint



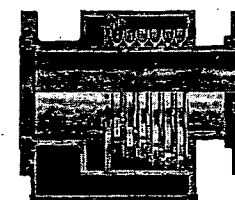
The ADSCO Piston-Ring Expansion Joint can be packed under full operating pressure without interruption to service. The piston rings hold the line pressure during the repacking operation. The ADSCO Piston-Ring Expansion Joint is available in all sizes in single and double slip design; semi-steel or cast steel bodies for all pressures up to 400 lb. and temperatures to 750 F., with flanged ends or ends beveled for welding. Write for Bulletin No. 35-15.

ADSCO Rotary Condensation Meter



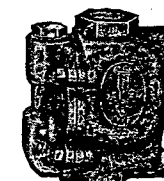
For measuring the steam condensation of a heating system or heating equipment. Used by District Heating Companies as a basis of charge for steam sold—institutional groups for steam cost distribution—and industrial firms for steam consumed in process work. Accurate within 1 per cent. Dependable. Compact. Reads directly in pounds of condensed steam. Furnished with semi-steel or aluminum body and cover in seven sizes; 250, 500, 750, 1500, 3000, 6000 and 12,000 lb. capacity per hour. Write for Bulletin No. 35-80.

ADSCO Packless Expansion Joint, U-Ring Type



The ADSCO Packless Expansion Joint, U-Ring Type is based on an entirely new principle for controlling pipe expansion. The expansion element is a series of die-formed "U-Rings" of stainless alloy steel, welded without transverse seams. The joint is installed with the element compressed. Under full operating conditions, the element is in a neutral position, free from strain. Available with all steel body, fully enclosed element, for pressures to 400 lb. and 800 F., with flanged ends or ends beveled for welding. Write for Bulletin No. 35-50.

ADSCO Vertical Steam Trap



Vacuum Service
To 16 lb. pressure
Gravity Service
0 to 125 lb. pressure

The ADSCO Vertical Steam Trap is a continuous flow, float type, trap with direct vertical inlet and outlet connections and an optional 90 deg. inlet connection. All working parts are mounted on the cover, all piping connections on the body. It has a reversible valve and reversible seat, giving a double life trap at no extra cost. The cover can be removed quickly and the mechanism serviced easily without disturbing the piping. Write for Bulletin No. 35-85A.

E. B. Badger & Sons Co.

63-75 Pitts Street,
Engineers and Manufacturers



Boston, Mass.

N. Y. OFFICE, 271 Madison Avenue
OFFICES IN PRINCIPAL CITIES

PRODUCTS and SERVICES—Corrugated Copper Expansion Joints, Pipe Bends, Chemical Apparatus, Copper and Sheet Metal Work, Copper Boilers and Hot Water Tanks; Engineers on Process Work

BADGER DIRECTED FLEXING EXPANSION JOINTS

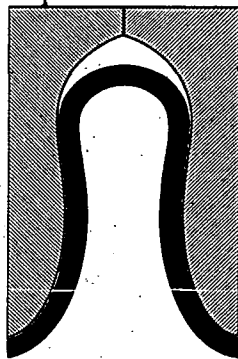
Since the Badger Corrugated Type of Expansion Joint was first placed on the market more than 40 years ago, several important improvements have been incorporated in this joint by E. B. Badger & Sons Co., Engineers, each adding to its life and utility.

1. Adoption of equalizing rings.
2. Machine method of making uniform corrugations.
3. Use of seamless copper tubes.
4. Improved machine method of manufacture resulting in minimum forming stresses.
5. Monel sleeves for protection against superheated steam.
6. Welding end joints.
7. Adoption of scientific heat treatment.
8. Use of special deoxidized copper.
9. Directed Flexing.

Advantages of the Corrugated Joint
Eliminates packing; avoids steam losses. No Servicing, Compact, Accessible, No Manholes needed.

Directed Flexing

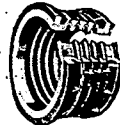
This feature, exclusive with joints made by E. B. BADGER & SONS Co., has brought about a marked increase in the life of the corrugated type of joint. Corrugations are "all-curved" and when flexing actually wrap and un-wrap themselves in an undulating movement about the equalizing rings. The movement is progressively controlled as well as limited. Original tests and actual service have proved thoroughly the soundness of this new design of corrugated expansion joint.



Different Types Available

Both flanged end and welding end joints are available from 3 in. (welding) and 4 in. (flanged) up. Single or double joints with or without service outlets. All joints can be equipped with telescoping monel metal sleeves to protect against superheated steam. We can furnish flanged joints less than 4 in. in size by fitting the 4-in. units with companion flanges, bolts and gaskets tapped to the desired size.

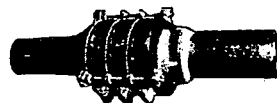
Flanged Types



Fitted with alignment bars on the 4 and 5 in. sizes. Choice of standard or extra heavy flanges. From 1 to 3-in. expansion, inclusive.

On 6 in. pipe and larger, no alignment bar is used. Standard 125 lb or extra heavy 250 lb flanges as required. From 1 to 4-in. expansion, inclusive.

Welding Types



For use with both saturated and superheated steam. Single type from 3 in. up, to take care of 1 in. expansion or more. Pipe nipples welded directly into pipe line.

Double units for 2 in. expansion or more. Equipped with service outlets if desired. Complete units furnished, mounted on base plate, with anchor and guides.

Single and Multiple Corrugated Expansion Joints for Low Pressure

For use between turbine or engine exhausts and condenser or on low pressure lines. Excellent for absorbing shock and vibration. Flanges in round, oval or rectangular shapes. Guaranteed up to 30 lb pressure.

American Machine and Metals Manufacturing Corp.

DeBothezat Division

Executive and Sales Offices: 100 Sixth Ave., New York, N. Y.

Factory: TROY, N. Y.

Branch Offices and Representatives in All Principal Cities



Type H of the "Selective Series."
30 in., 5 hp. 1140 r.p.m. 8200
c.f.m. against 1 in. static pressure

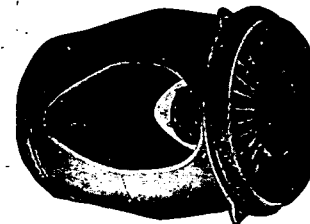
Fans and Blowers are guaranteed to have non-overloading power characteristics. Complete operating safety under varying working conditions is assured.

Disc Pressure Fans "Selective Series"—Meet all pressure requirements efficiently—made in various sizes from 8 in. to 10 ft. Bulletin SS-101 contains complete technical data on L-Type for low pressures and HL and H Types for high pressures.

Giant Fans—Are made in 5 ft. to 10 ft. diameters and are directly connected to motor, chain or Tex-rope drive. Technical data concerning these fans is contained in Bulletin SS-101.

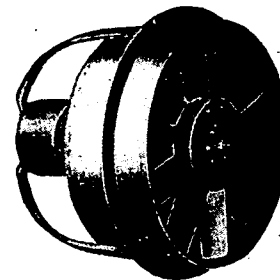
Bifurcators—Can be placed in a straight duct without introducing right angle turns in the ducts or requiring extended shafts and bearings. As motor is located outside of air stream, safe, care-free operation is assured.

Vari-Speed Fans—Speed selection is secured through variable pitch motor pulley. Quiet. Compact. Reliable.

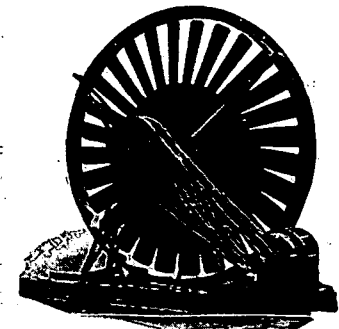


Bifurcator. For special ventilating jobs where fumes of excessive temperature, corrosive or explosive character, are to be removed

Duplex-Rotation Impeller Blowers—Consist of disc pressure fan wheels rotating in opposite senses so designed that the rotational losses are completely eliminated. The unit is directly driven by a DUPLEX-ROTATION motor, which eliminates belts and gears. It is equipped with ball bearings throughout and is suitable for horizontal or vertical mounting. This unit can replace any fan of same diameter now producing objectionable noise, without alterations. Impeller Blowers are also supplied with belt or gear drives. Write for Bulletin DR-101.



2-Stage Duplex-Rotation Impeller Blower. Large air volumes against high static pressures at low peripheral velocities. Extremely high efficiency over entire range of operation



Giant Fan. Powers from 5 hp. to 100 hp. Volumes to 310,000 c.f.m. static pressures to 5 in.

Bayley Blower Company

1817 S. Sixty-Sixth Street Branches in Principal Cities Milwaukee, 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

Bayley Plexiform Fan:

Is a multi-blade fan for supplying air for heating and ventilating systems, manufacturing processes, drying systems, forced and induced draft systems. It is suitable for handling high or low temperature gases at medium or low pressure. Will deliver maximum quantities requiring minimum space with great economy.

This is a distinct Bayley product, high class material and workmanship, properly designed to avoid excessive vibration and overstraining of parts. Inlets and outlets are properly sized for maximum delivery and maximum efficiency. Fans are furnished in single or double width of any required arrangement and with sleeve or anti-friction bearings.

Aeroplex Fan:

Is of high speed design with self limiting power characteristics. Application parallel to the Plexiform Fan. Highly efficient and quiet in operation.

Bayley Exhausters and Pressure Blowers:

Type "B" exhaust fan is for heavy duty, handling refuse from industrial and textile plants. Type "SE" is used in handling smoke, fumes and dust-laden gases. Type "H" for high-pressure work. These units are highly efficient and of high class design and workmanship.

Bayley Turbo Air Washers, Humidifiers and De-Humidifiers:

The Turbo Atomizer used in the Bayley Washer produces a steady, fine spray. Water at low pressure is delivered to the center of a rapidly revolving cone-shaped rotor provided with atomizing pins set in its periphery. This



atomizer requires very little attention, and will operate successfully under low water pressure. The orifices are large and this atomizer, unlike high pressure nozzles, cannot clog.

Bayley Chinook Heating Sections:

The Chinook section is used with blast heating, ventilating and drying systems, and is suitable for high or low pressure steam circulation. The base is divided into two chambers. Steam enters (see cut) the lower chamber, rising through $\frac{3}{8}$ -in. pipes located within the $1\frac{1}{4}$ -in. pipes leading from the upper chamber. Condensation takes place in the larger pipes, the water falling into the upper chamber and draining away through the return outlet. The Chinook can be repaired in the middle of the bank without breaking steam connections or taking down a section.

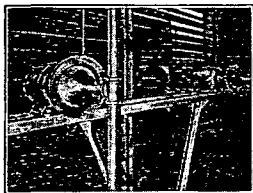
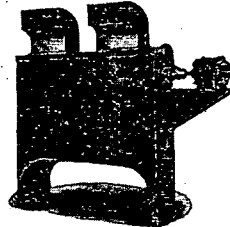
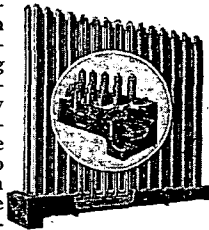
Shipped assembled in smaller sizes, and knocked down in the larger units. May be installed in horizontal or vertical position.

Bayley Chinookfin Heating Sections:

Are the same design as the Chinook Heaters, using heavy gauge copper fin tubes. As compared with Chinook it is much lighter and occupies less space.

Bayley Plexin Unit Heaters:

This unit incorporates Chinookfin radiation and Plexiform or Aeroplex fans. The fan assembly including top plate and motor is removable as a unit for maintenance and inspection. The heating element is a removable unit. Casing all welded extra heavy gauge. This is an exceptionally high grade unit at a moderate price.



The Bayley Turbo Air Washer Showing Turbo Atomizer and Eliminator

Buffalo Forge Company

484 Broadway, Buffalo, N. Y.

Branch Offices

ALBANY.....	414 Standard Bldg.	KNOXVILLE—C. F. Sexton.....	P. O. Box 2224
ATLANTA.....	724 First Natl. Bk. Bldg.	LOS ANGELES.....	708 Pershing Sq. Bldg.
BALTIMORE.....	303 Morris Bldg.	MEMPHIS—A. W. Shelby.....	134 S. Second St.
BOSTON.....	Station Melrose, Mass.	MINNEAPOLIS—E. F. Bell.....	619 Foshay Tower
CHARLOTTE—J. W. Fraser & Co.,		NASHVILLE—Southern Sales Co.....	117 Fifth Ave. N.
P. O. Box 1436—124 Brevard Court		NEW ORLEANS—A. C. Hayes.....	1415 Carondelet Bldg.
CHATTANOOGA—H. T. Purcell Co.....	502 Pound Bldg.	NEW YORK CITY.....	39 Cortlandt St.
CHICAGO.....	20 N. Wacker Drive	PHILADELPHIA.....	703 Culton Bldg.
CINCINNATI.....	622 Broadway	PITTSBURGH.....	912 Fulton Bldg.
CLEVELAND.....	418 Rockefeller Bldg.	SAN FRANCISCO—Herbert-Moore Mchry. Co.....	550 Fifth St.
DALLAS—Buffalo Engineering Co., Inc.....	315 South Harwood	ST. LOUIS.....	1596 Arcade Bldg.
DENVER—Hendrie & Bolthoff Mfg. & Supply Co.,	1621-17th Street	SALT LAKE CITY—Salt Lake Hardware Co.....	
DETROIT—Coon-DeVisser Co.....	2051 W. Lafayette Blvd.	SEATTLE.....	2434 First Ave. S.
GREENVILLE—Geo. R. Morgan,		TOLEDO—Carl M. Eyster Co.....	1922 Linwood Ave.
P. O. Box 563, 316 Walker Bldg.		WASHINGTON.....	820 Woodward Bldg.
KANSAS CITY.....	315 Dwight Bldg.	WILKES-BARRE—Power Engineering Corp.,	Coal Exchange Bldg.
KITCHENER, ONT.....	Canadian Blower & Forge Co., Ltd.		

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 Unit Coolers

Suspended Type: SC Cooler has copper coils suitable for cold water, brine, methyl chloride or Freon. Type SS units have steel coils, hot galvanized after assembly, for use with ammonia. Compact, simple, inexpensive.

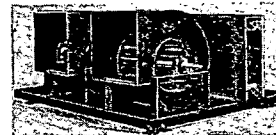
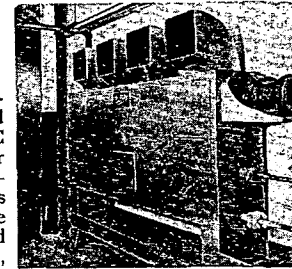
Floor and Flat Suspended: Available for same kinds of refrigerants, but with capacities from 2130 to 6480 cfm.

Central Conditioning Cabinets:

Type PC—for cooling and heating. Heating coils are two-row copper fin type capable of heating from 70 F to 135 F with 2 lb steam. Cooling capacities— from 3 to 24 tons.

Buffalo Air Washers

The outstanding features of Buffalo Air Washers are: eliminators in one piece and demountable; spray nozzles, non-clogging; suction screen extends entire width of tank; maximum contact between air and water spray. A complete line of self-contained unit air washers is also available.



Buffalo Unit Heaters

Gas Units: Made in suspended and floor types. Total capacity 75,000 to 450,000 Btu input per hour. All models equipped with full automatic safety features. Eliminates boiler room, fuel storage, and necessity for fireman or engineer; is easy to install; clean and automatically controlled. Write for bulletin.

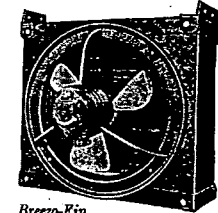
Steam Units: Both suspended and floor type units in a large range of capacities and with centrifugal or disk type fans.

Buffalo "Limit-Load" Conoidal Fans with Silent Floating Bases

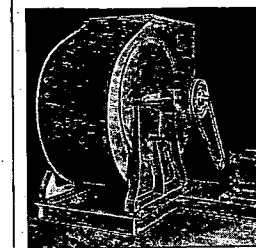
The Buffalo "Limit-Load," high-efficiency, non-overloading ventilating fan mounted on the silent, insulated floating fan base eliminates all motor and fan vibration, making an almost noiseless installation.

Breezo Ventilating Fans

Breezo fans provide efficient, inexpensive ventilation on any job where they may be used to exhaust into the open. Made in sizes from 8 in. to 36 in. diameter.



Breezo-Fin Steam Unit

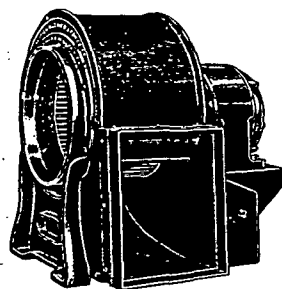


Champion Blower & Forge Co.

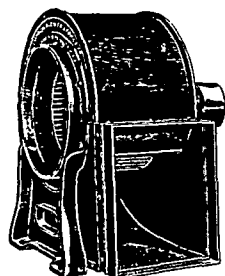
Manufacturers and Engineers

Plant and General Offices: Lancaster, Pa.

Manufacturers of Blowers, Ventilating Fans, and Exhaust Fans for Air and Material; and Blast Gates.



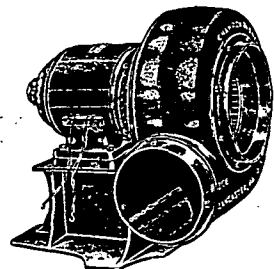
Type "SE" Electric Driven Fans



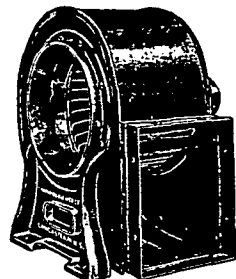
Type "S" Manivane Belt Driven Fans

Sizes—6 to 60 in. Wheels

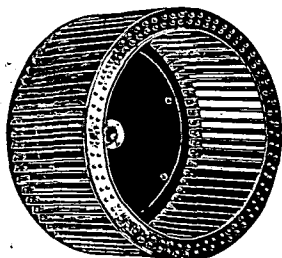
For Heating, Ventilating, Cooling, Drying or Forced Draft



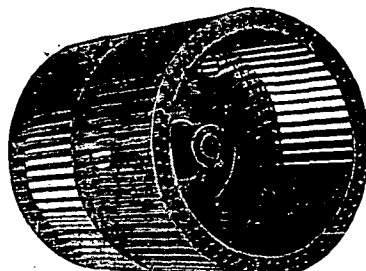
Type "CE" Manivane Fans
For Duct Work and Quiet Operation.



Type "BC" Backward Curve Fan
Sizes—12 to 60 in. Wheels



Single Width, Single Inlet.



Double Width, Double Inlet.

Type "C" Manivane Blast Wheels

Built strong and substantial to withstand air resistance. Free from vibration and noise. Especially adapted for oil burner, stoker, general heating and ventilating purposes; where noise is objectionable.

ILG Electric Ventilating Company

Propeller Fans, Blowers, Unit Heaters, Air Conditioning Equipment

2880 North Crawford Avenue, Chicago, Ill.

Sales Representatives in all Principal Cities

ILG SELF-COOLED MOTOR PROPELLER FANS

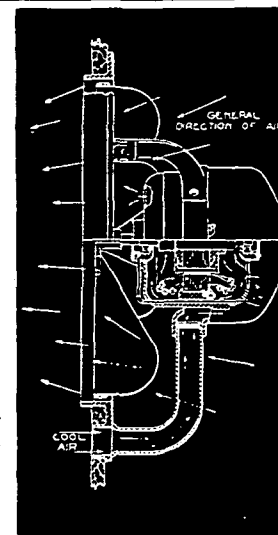


The green Ilg Self-Cooled Motor Propeller Fan is used everywhere for removal of foul air, fumes, heat, etc. Features include: the patented Ilg self-cooled motor, dynamically balanced, vibration-free, bucket type wheel, and a strong one-nameplate responsibility. Motor, wheel and frame are all Ilg-built.

The ingenious self-cooled motor design of the Ilg fan combines the low operating cost of the open motor with the protection of the fully enclosed motor.

The black square tells the story. The fan action draws clean air through the vent pipe from the outside; circulates it through the motor (follow the arrows) and exhausts it. The Ilg motor stays clean, stays cool; no foul air can reach it.

There is a complete range of Ilg fan sizes from 12 to 72 in. in A.C. and D.C. Each fan is thoroughly tested before shipment. Ratings are guaranteed to be in accordance with the Standard Test Code of the A.S.H.V.E. and the National Association of Fan Manufacturers.



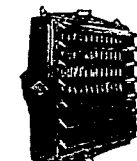
The Motor that Breathes

OTHER ILG PRODUCTS

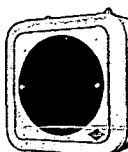


Ilg Universal Multiblade Blowers—Ilg Type B Universal Blowers are designed to combine compactness, quietness and efficiency. Motor is recessed in side of blower, requiring no separate base. Multiblade wheel is mounted on motor shaft. There is no inlet bearing. Available also for belt drive. Capacities 1750 cfm to 70,000 cfm, single and double width.

Ilg Type "B" Volume Blowers—The latest development in small volume, low pressure blowers. Light weight, quiet running. Dynamically balanced multiblade wheel is mounted on motor shaft. Motor and steel housing are supported by cast-iron base. Universal discharge. Available in 11 capacities, 180 cfm to 2100 cfm.



Ilg Unit Heaters—Steam and Electric—Copper tube and fin construction and enclosed self-cooled motor. Ilg-built throughout. For steam or hot water. Tested with 500 lb hydrostatic pressure. Available in 14 capacities; also, Ilg electric unit heaters for all electric operation, available in 20 sizes.



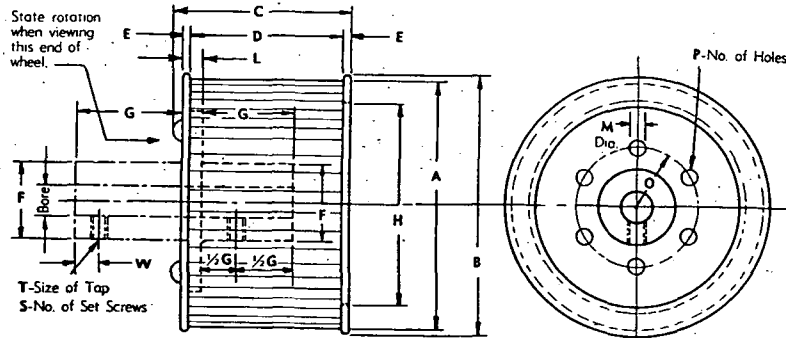
Ilg Cooling and Air-Conditioning Units—Self-contained Ilg Spot Coolers in 1/2 ton and 1 ton cooling capacities with water or air cooled compressors. Also central unit systems using floor cabinet or ceiling suspension units with remotely located compressor. For cooling, dehumidifying, and recirculating. Also, for heating and humidifying.

The Torrington Mfg. Co.

50 Franklin Street, Torrington, Conn.

PRODUCTS—All Aluminum Blower Wheels and Disc Propeller Type Fans

SINGLE WIDTH SINGLE INLET BLOWER WHEELS

Pat. 1,700,017
1-22-29

Hub can be furnished for inside or outside of wheel.

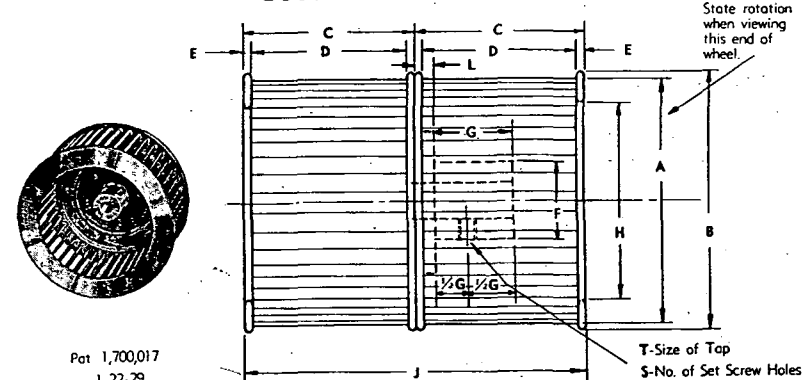
No.	Nominal Size	No. of Blades	Actual Blade Dia. A	Actual O.D. B	Actual Blade Width C	E	F	G	H	L	Bore	O	M	P	T	S	W	Weight With Hub
00A	3 x 1 1/2	27	3	3 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	3 oz.
00	3 x 1 1/2	27	3	3 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	3 1/2 oz.
1/4	3 1/2 x 1 1/2	45	3 1/2	3 3/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	4 1/2 oz.
0A	4 1/2 x 2 1/2	34	4 1/2	4 3/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	8 1/2 oz.
0A	4 1/2 x 2 1/2	45	4 1/2	4 3/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	9 1/2 oz.
0B	4 1/2 x 3	45	4 1/2	4 3/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	11 1/2 oz.
1/4B	5 x 2 1/2	45	5	5 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	13 oz.
1/4A	5 x 2 1/2	45	5	5 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	13 oz.
1/4	5 x 3	38	5	5 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	12 1/2 oz.
1D	6 x 1	31	6	6 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	13 oz.
1A	6 x 2	32	6	6 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	14 oz.
1E	6 x 2	45	6	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	15 oz.
1	6 x 3	31	6 1/4	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	15 1/2 oz.
1C	6 x 3	45	6 1/4	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	15 1/2 oz.
1B	6 x 3 1/2	38	6 1/4	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	15 1/2 oz.
1 1/4A	7 1/2 x 2	47	7 1/2	7 1/2	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # 1 1/2 oz.
1 1/4B	7 1/2 x 3	37	7 1/2	7 1/2	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	1 # 4 1/2 oz.
1 1/4	7 1/2 x 3 1/2	47	7 1/2	7 1/2	4	4	4	4	4	4	4	4	4	4	4	4	4	1 # - 8 1/2 oz.
1 1/4C	7 1/2 x 4 1/2	36	7 1/2	7 1/2	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	1 # - 9 oz.
1 1/4A	9 x 2	47	9	9 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 # - 1 oz.
1 1/4D	9 x 2 1/2	48	9	9 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 # - 2 oz.
1 1/4B	9 x 3	47	9	9 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 # - 4 oz.
1 1/4C	9 x 3 1/2	59	9	9 1/4	4	4	4	4	4	4	4	4	4	4	4	4	4	2 # - 5 oz.
1 1/4	9 x 4 1/2	47	9	9 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	2 # - 8 oz.
1 1/4	10 1/2 x 5 1/2	47	10 1/2	10 1/2	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	3 # - 12 oz.
2	12 x 6	64	11 1/4	12 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	5 # - 4 oz.
2 1/2	15 x 7 1/2	64	15 1/4	15 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	10 # - 0 oz.

1. Attractive appearance.
2. Light weight minimizes power consumption.
3. No protective finish necessary.
4. Lower transportation costs because of light weight.
5. Aluminum construction minimizes resonance and wheel operates more quietly.
6. No rivet head or other projections on inlet face of wheel.
7. Patented construction reduces manufacturing cost.
8. Rugged design—tested to speeds far in excess of normal.
9. Made from a special aluminum alloy having great strength.
10. Approximately one-third weight of steel wheels.
11. Low starting torque.

The Torrington Mfg. Co.

50 Franklin Street, Torrington, Conn.

DOUBLE WIDTH DOUBLE INLET BLOWER WHEELS

Pat. 1,700,017
1-22-29

No.	Nominal Size	No. of Blades	Actual Blade Dia. A	Actual O.D. B	C	D	E	F	G	H	J	L	Bore	T	S	W	Weight With Hub
00A	3 x 2 1/2	27	3	3 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	4 1/2 oz.
00	3 x 3	27	3	3 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	5 1/2 oz.
1/4	3 1/2 x 3	45	3 1/2	3 3/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	7 1/2 oz.
0	4 1/2 x 4 1/2	34	4 1/2	4 3/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	14 oz.
0A	4 1/2 x 4 1/2	45	4 1/2	4 3/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 #
0B	4 1/2 x 6	45	4 1/2	4 3/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 3 1/2 oz.
1/4B	5 x 4 1/2	45	5	5 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 5 oz.
1/4A	5 x 5 1/2	45	5	5 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 6 oz.
1/4	5 x 6	38	5	5 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 5 oz.
1D	6 x 2	31	6	6 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 1/4	1 # - 5 1/2 oz.
1A	6 x 4	32	6	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 8 oz.
1E	6 x 4	45	6	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 7 1/2 oz.
1	6 x 6	31	6 1/4	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 12 oz.
1C	6 x 6	45	6 1/4	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 10 1/2 oz.
1B	6 x 7 1/2	38	6 1/4	6 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 15 oz.
1 1/4A	7 1/2 x 4	47	7 1/2	7 1/2	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	1 # - 3 oz.
1 1/4B	7 1/2 x 6	37	7 1/2	7 1/2	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	3 1/4	1 # - 4 1/2 oz.
1 1/4	7 1/2 x 7 1/2	47	7 1/2	7 1/2	4	4	4	4	4	4	4	4	4	4	4	4	1 # - 8 1/2 oz.
1 1/4C	7 1/2 x 9	36	7 1/2	7 1/2	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	4 3/4	1 # - 11 1/2 oz.
1 1/4A	9 x 4	47	9	9 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 # - 4 oz.
1 1/4D	9 x 5 1/2	48	9	9 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 # - 7 oz.
1 1/4B	9 x 6	47	9	9 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 1/4	2 # - 8 1/2 oz.
1 1/4C	9 x 7 1/2	59	9	9 1/4	4	4	4	4	4	4	4	4	4	4	4	4	2 # - 13 1/2 oz.
1 1/4	9 x 9	47	9	9 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	4 1/4	2 # - 2 oz.
1 1/4	10 1/2 x 10 1/2	47	10 1/2	10 1/2	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	5 1/4	2 # - 5 1/2 oz.
2	12 x 12	64	11 1/4	12 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	6 1/4	2 # - 11 oz.
2 1/2	15 x 15	64	15 1/4	15 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	7 1/4	2 # - 8 oz.

HOUSING SCROLLS. We do not manufacture housings but have prepared drawing with tables of dimensions for Scrolls for each size wheel. Copy will be mailed upon request.

CAPACITY TABLES. Capacity rating and curve sheets showing static pressure, air deliveries and brake horsepower at various speeds are available for the majority of sizes and will be sent upon request.

L. J. Wing Mfg. Co.

Branch Offices in 59 Seventh Avenue, New York, N. Y.
Principal Cities

PHONE: CHelsea-3-0026

Factory:
NEWARK, N. J.

Wing Featherweight Unit Heaters and Process Heating Units; Utility Unit Heaters; Wing Scruplex Safety Ventilating Fans, Fog Eliminators and Exhausters; Wing Forced Draft Blowers, Turbine or Motor-Driven; Steam Turbines; Man-Coolers; Wing Garage Heaters; Wing Door Heaters.



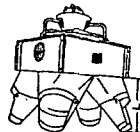
Wing Featherweight Unit Heater—with the Wing Featherfin Heating Element

Wing Featherweight Unit Heaters

Light weight and vertical downward discharge of the heated air through high velocity multiple discharge outlets are original and unique features of Wing Featherweight Unit Heaters. Thus downward circulation of large volumes of warmed air to the floor and uniform distribution of it over the entire area is assured, with resultant economy in plant heating.

They produce a pleasing sense of warmth at the floor level because the warmed air from each of the several heater discharges actually reaches the floor. There is a design of discharge outlet to meet every condition. *Bulletin H-5.*

Standard Discharge Outlets for Type HC



Wing Featherfin Heating Element

Extremely light in weight. By simple variations of the heating surface any desired final air temperature may be obtained with any given steam pressure.

Fin and tube extended surface type—Hairpin or return bend design—

Headers of steel, tubular design—Tested at 1000 lb. pressure.

Bulletin H-5.



Detail of Wing Featherfin Heating Element showing Compression Union Tube Connection

Wing Utility Heaters



A general purpose heater. Delivers heat-air in one general direction. With adjustable vane diffusers and safety guard for fan. *Bulletin U-3.*

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.*

Wing Featherfin Heater Sections for general blast heating. *Bulletin HS-1.*



Wing Industrial Fog Eliminators

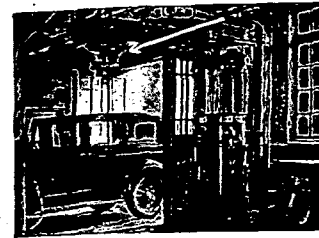
Eliminate fog, odor and fumes in dye-houses, bleacheries, pasteurizing plants, chemical works, paper mills, etc. Easy to install. *Bulletin FE-11.*

Wing Garage Heaters

Especially designed for heating garages. They are located above the open aisles, and deliver a high-velocity conical column of air downward in such a manner that it rolls along the floor in every direction, putting the heat on the floor and under the cars just where it is needed. Cuts yearly cost of heating in half. *Bulletin G-1.*



Type G Heater with Cone Discharge Outlet

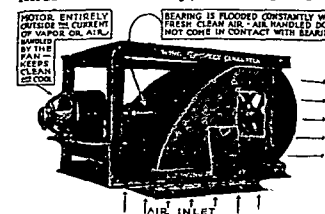


Wing-Scruplex Safety Ventilating Fan

A propeller type fan that delivers air against static pressure, quietly, efficiently and safely.

The screw design propeller moves the air forward in straight lines without eddy, assuring high static efficiency.

Sizes 10 to 60 in., capacities range from .950 to 52,000 c.f.m. *Bulletin F-5.*



Wing-Scruplex Exhauster with bottom inlet

Wing Variable Temperature Featherfin Heater Sections

Obtainable separately for general heating and process work, or in combination with any of the Wing Unit Heaters.

Will deliver air at any temperature from the maximum obtainable, with prevailing steam pressure, down to the temperature of entering air.

The temperature of discharged air is controllable manually or automatically without throttling the steam supply—therefore proof against freeze-ups. *Bulletin VT-1.*



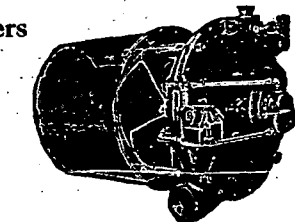
Wing EM Blower

Wing Motor-Driven Blowers for Forced Draft

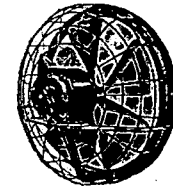
The installation of Wing Motor-Driven Type EM Blowers and Combustion Controls on boilers of all sizes makes possible the use of Rice and Barley coal, Screenings and other inexpensive fuels with great savings in fuel cost, which usually pay for the blowers in the first year of operation. Help maintain steam pressure and give increased capacity with minimum attendance. *Bulletin M-66.*

Wing Turbine Blowers

Forced draft for hand-fired or stoker-fired high pressure boilers is economically supplied by the Wing Turbine Blower. Greatly increased boiler capacity, flexibility of control and thorough combustion of low cost fuels are obtained. These units occupy very little floor space, and hence can be readily installed in any boiler room. Great savings can be effected in plant operation by the installation of Wing Turbine Blowers. Exhaust steam from turbine can be used for heating buildings or feed water or for processing purposes.



Bulletin T-97.



Wing Door Heaters

A combined electric and steam directional heater, very compact in size and simple in construction. Suspended from ceiling over each opening. Used for instantaneously heating inrush of cold air at doorways of garages, bus terminal stations, freight houses, docks, piers, shipping and receiving departments. *Bulletin D-1.*

B. F. Sturtevant Co.
Hyde Park, Boston, Mass.

Sturtevant
REG. U.S. PAT. OFF.



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The Cooling and Air Conditioning Corporation
Division of B. F. STURTEVANT COMPANY

A. M. Lockett & Co.
New Orleans, La.
Houston, Texas
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ATLANTA CAMDEN CHICAGO LOS ANGELES NEW YORK
HYDE PARK BOSTON, MASS.

**DATA ON HEATING, VENTILATING,
AIR CONDITIONING AND VACUUM
CLEANING EQUIPMENT FOR
ARCHITECTS, ENGINEERS,
CONTRACTORS**

The publications listed below have been prepared to aid the architect, engineer and contractor in the selection of proper equipment for industrial, public, and private buildings of all types and sizes. If you do not have all of these publications in your file we will gladly send copies upon request.

COOPERATION

Sturtevant Engineers, located at each of the offices listed are always ready to cooperate with architects, engineers and contractors in the selection of equipment suitable for any prospective installation.

VENTILATING FANS

CATALOGUE No.

- 271-2 Multivane Fans (Forwardly curved blade type).
- 381-2 Silentvane Fans (Backwardly curved blade type).
- 414 Rexvane Fans (Radial blade type).
- 385 Propeller Fans.
- 332-1 Ventilating Sets (direct motor-driven centrifugal type fans for ventilating small rooms. Capacities: 80 to 1600 c.f.m.).
- 400-5 Direct-connected Fans and Blowers (Propeller Fans; Window Fans for kitchens and offices; Centrifugal Fans from 80 to 6460 c. f. m.; Portable Gas-Engine-Driven Fans; Coal Burning Blowers; Forge Blowers; Dust Blowers).

CATALOGUE No.

- 345 Carbon Monoxide Asphyxiation and Its Prevention.

**MISCELLANEOUS HEATING AND
VENTILATING EQUIPMENT**

CATALOGUE No.

- 377-1 Unit Ventilators.
- 395-1 Rexvane Speed Heaters (Floor type unit heaters).
- 396-2 Speed Heaters (Suspended Type Unit Heaters).
- 382 Coal Burning Blowers.

AIR CONDITIONING EQUIPMENT

CATALOGUE No.

- 295-1 Air Washers.
- AC 101 Industrial Air Conditioning.
- 398 Comfort Air Conditioning.
- 389 Household Humidifiers.
- 378 Filtracooler (Compact, high efficiency air washer. For filtering, washing, humidifying, cooling, dehumidifying. Used principally for public buildings and factories).
- 383 Humidifier (A humidifying and filtering unit primarily adapted to central systems. Used for both comfort and industrial processing work).
- 384 Suspended Type Air Conditioning Units.
- 401-1 Railway Air Conditioning.

VACUUM CLEANERS

CATALOGUE No.

- 397-1 Central System Vacuum Cleaners.
- 413 "Vortex" Portable Vacuum Cleaners.

E. K. Campbell Heating Co.

2441-3-5 Charlotte Street

Factory Branch
St. Louis, Mo.

Kansas City, Mo.

Sales Agents in
Many Cities

**MANUFACTURERS OF HEATING, VENTILATING
AND AIR CONDITIONING EQUIPMENT**

PRODUCTS

**E. K. Campbell Furnace Fan System of Heating and Cooling.
Thermidaire Fans and Blowers.
Thermidaire Steam Unit Heaters.
Thermidaire Blast Coils.
Thermidaire Gas-Fired Unit Heaters, propeller and centrifugal fan type.
Diffusers and grilles.**

The **E. K. CAMPBELL FURNACE FAN SYSTEM** is installed in more than a thousand buildings—churches, schools, theatres, auditoriums, factories, garages—in twenty-three states and Canada. The system is in use in 17 churches in Indianapolis; also in the Butler University Field House, reputed to be the largest building in the world heated by a furnace fan system—in forty-five churches and over a hundred buildings in greater Kansas City; twenty-eight churches, and other buildings in St. Louis including one printing plant completely air conditioned.

This system is especially adaptable to buildings with large spaces that are intermittently heated. Large volumes of air give low final temperature and even distribution, with low temperature gradient from floor to ceiling. Butler Field House, with 40,000 sq ft floor area in one room, has maximum temperature difference at any level of 3° F and a temperature gradient from floor to ceiling of 12° F in 85 ft.

An intermittently heated building has a large amount of cold material, which must be warmed before comfort is obtainable. The conservative ratings of the **E. K. Campbell Furnace** take into account this heating-up load, the furnace being able with hand-fired coal to put out double its rated capacity.

With large air movement, ample furnace capacity, and a furnace which permits heavy firing with low stack temperatures, fuel economy naturally follows. In addition this plant gives ample humidity, ventilation without drafts or open windows, and summer cooling, either with air movement alone, or, in conjunction with other apparatus, positive cooling.

The firebox of the furnace is built of heavy flange steel, reinforced above the firebrick lining with an additional thickness of steel, giving greater strength and longer life. From the firebox the gases pass into an economizer, which utilizes the plunge draft and counterflow principles, the gases being taken out at the bottom. This increases heat extraction, makes the furnace economical to operate, and the heavy construction makes it capable of standing the strain of heavy firing easily.

Twenty-six years of pioneering experience in developing the system and manufacturing the equipment make our engineering service valuable. When our equipment is used as directed, we relieve the architect, engineer, contractor and owner of any expense needed under our guaranty of results.

Thermidaire fans and blowers, for heating, ventilating, and cooling are sturdy, efficient and quiet operating.

Thermidaire steam unit heaters are built in all types. The heating element is all copper, with a special brazing which has equal mechanical strength to the copper, insuring permanent tightness. A wide range of types and capacities makes these units adaptable for any condition.

Thermidaire blast coils have the same heating element construction as the unit heaters, all copper, brazed for permanent tightness. Encased in heavy steel, they are easily installed, and make a permanent installation.

Thermidaire gas-fired unit heaters have the vertical tube, plunge draft design, with large gas passages, which make the units self-cleaning and maintain the high and constant efficiency.

E. K. Campbell Heating Co. also manufacture diffusers, and grilles of various designs. Neat, inconspicuous, finished in baked enamel with choice of several colors, they are suitable for most conditions.

Further information gladly given on request.

Electric Air Heater Company

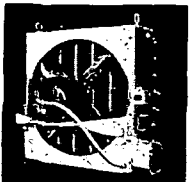
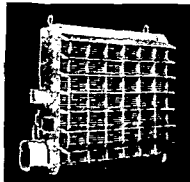
Division of The American Foundry Equipment Co.

677 Byrkit Street, Mishawaka, Indiana, U. S. A.

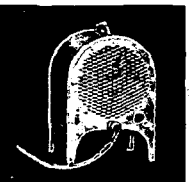
Manufacturers of Industrial and Domestic Electric Air Heaters

Industrial Electromode Heaters

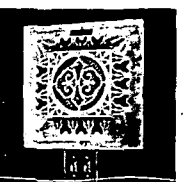
This type electric air heater design includes all the principles of steam unit heaters. The difference is that the source of heat supply is a set of wires instead of a Boiler House, piping and steam traps. Controlled by thermostats, electric air heaters are economical, easily installed and heat is available within one minute at any time, and cut off instantly when no longer required.



Industrial Heater



Portable Heater



Bilt-In Heater

Model	Ratings			60 F Temp. Rise		45 F Temp. Rise		Fan Diam.	Price List
	KW	BTU	EDR	CFM	FPM	CFM	FPM		
8-3	3.0	10245	42.7	155	515	207	690	6 1/2	60.00
11-4 1/2	4.5	15367	64.0	233	382	310	596	9	90.00
11-6	6.0	20490	85.0	310	508	412	660		
14-10	10.0	34150	142.0	517	488	684	645	12	135.00
14-15	15.0	51225	213.0	776	732	1034	975		
20-25	25.0	85375	356.0	1290	533	1720	712	18	240.00
20-35	35.0	119525	498.0	1800	750	2400	1060		
27-45	45.0	153675	640.0	2325	570	3066	748	24	330.00
27-60	60.0	204900	854.0	3100	760	4133	1000		
32-90	90.0	307350	1280.0	4656	727	6108	954	30	450.00
32-120	120.0	409800	1708.0	6208	968	8280	1294		

Portable Electromode Heaters

This portable electric air heater meets the demand for portable heaters as well as for permanent installations. This type may be equipped with a built-in thermostat for temperature control. A special switch can be added which will permit running the fan without heat. A waterpan can be added so that 1 to 3 pints of water may be evaporated per hour to give humidity control.

Model	KW	BTU	EDR	60 F Temp. Rise		45 F Temp. Rise		Face Area Sq Ft	Diam. Fan	Net Wt.	Price List
				CFM	FPM	CFM	FPM				
A-15	1.5	5122	21.3	75	220	103	310	0.327	7 3/4	20	50.00
A-20	2.0	6830	28.4	100	305	138	420				
B-30	3.0	10245	42.7	150	340	207	468	0.442	9	25	60.00
B-40	4.0	13660	56.9	200	450	276	625				
C-50	5.0	17075	71.1	260	435	345	575	0.60	11 1/2	32	70.00
C-60	6.0	20490	85.3	310	515	415	692				

Prices f. o. b. Mishawaka, Ind.

For 25, 40, 60 cycle and D.C. current—add 10 per cent to list. 1-Kwhr = 3415 Btu = 14.23 sq ft Equiv. in Dir. Radiation. 60 cycle A.C. is Standard—other cycles and D.C. at extra cost for motor. Important: Specify Voltage and kind of current when ordering.

Bilt-In-Wall Type Electromodes

This unit is designed to fit the heating requirements of the home, apartment, office or summer (winter) home. The grille is of an attractive design and may be finished with any color to harmonize with interior decoration scheme. This heater is installed on the standard spaced stud just above the baseboard and controlled by a thermostat it gives automatic controlled temperature in each or all rooms.

Send for Data Book No. 236

Fedders Manufacturing Co.

HEAT TRANSFER SPECIALISTS SINCE 1896

57 Tonawanda Street, Buffalo, N. Y., U. S. A.

Manufacturers of Unit Heaters, Household and Commercial Electric Refrigeration Appliances and Air Conditioning Equipment

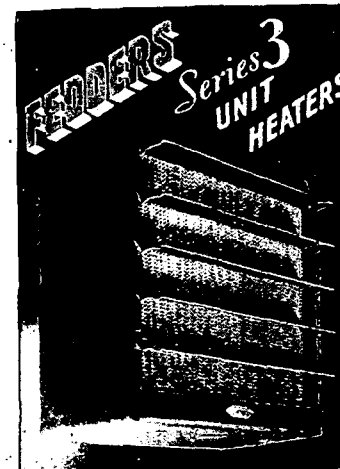
FEDDERS SERIES 3 UNIT HEATERS

Handsome external appearance is an indication of advanced engineering design inside as well as outside. Sturdy Series 3 Unit Heater Cabinets reinforced with integral channel members protect the heating element from torsional stresses and twist due to piping strains.

Full Floating Mountings allow for expansion and maintain element alignment in cabinet. Copper tubes having a 1 to 3 streamline ratio and patented individual convoluted fin design eliminate differential expansion stresses between neighboring tubes. Curved headers provide extreme strength and eliminate diaphragmatic action thus assuring a stable heating element.

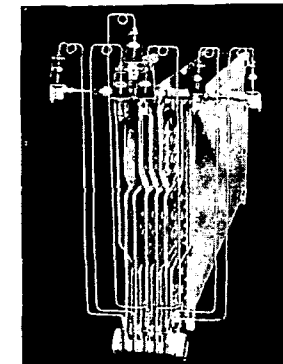
This combination of handsome appearance, rugged construction and quiet operation makes them ideal for commercial as well as industrial applications. Twenty standard models in well graduated capacities up to 1300 sq. ft. EDR. Write for Catalog.

Pat. 1,970,105; other Pats. Pending



FEDDERS STANDARD AIR CONDITIONING SURFACE

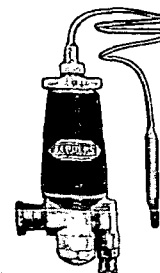
Fedders complete line of Air Conditioning Surface, built in capacities up to 100 tons I.M.E., is cataloged and rated on a package basis. It simplifies engineering of the job. All copper and brass construction for high heat transfer efficiency. Fedders manifolding provides proper distribution throughout the surface and reduces internal friction to a minimum. Approved by Underwriters' Laboratories. Fedders High Capacity Thermostatic Expansion Valves are easily adjusted and provide sensitive, accurate refrigerant control. Write for catalog giving complete specifications and capacities. (Pat. pending).



MODEL 33 HIGH CAPACITY THERMOSTATIC EXPANSION VALVE

Fedders Model 33-HC Valves help get the greatest effectiveness out of the coils. On multiple section coils, the settings of the valves controlling each section can be easily adjusted in accordance with the varying temperature differentials as the warm inlet air becomes cooler as it progresses through each section of the coil.

Orifice: 0.218 in. Diameter. Capacity: 4 tons with Freon. 8 1/2 tons with Methyl Chloride.



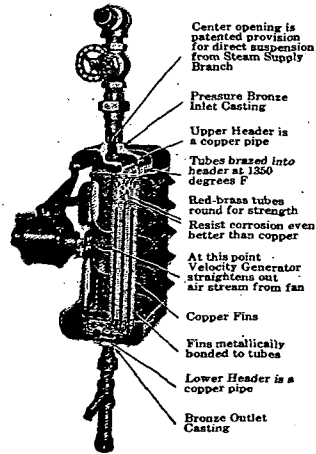
Pat. Nos. 1,987,948, 1,974,631

Modine Manufacturing Co.

17th and Holburn Streets, Racine, Wisconsin
Branches in All Principal Cities

MODINE UNIT HEATERS

Application—Not only successfully used for industrial, garage and similar space-heating applications, but ideally suited also to show-rooms, stores, oil stations, churches, school rooms, resi-



Each tube of condenser, before entering lower header, bends in a way which provides for free expansion.

dential recreation rooms, etc. They have also effected important economies in the drying of paint, leather, enamels, paper, wood, etc., by speeding up and bettering the drying process.

Description—Exclusive features that not only make the Modine a better unit

MODINE UNIT HEATERS CAPACITIES AND DIMENSIONS (In Inches)

Model No.	Over-all Height	Width	Depth Less Motor	E.D.R.	C.F.M.	Motor R.P.M.
76	10 1/2	9 3/4	5 3/4	76	187	1550
126	16	13	8	126	456	1590
152	18	15	8	152	540	1590
181	18	15	8	181	770	1140
204	18	15	8	204	735	1140
238	18	15	8	238	731	1140
275	19 1/2	18	9	275	1052	1140
352	22 1/2	18	9	352	1425	1140
440	22 1/2	18	9	440	1320	1140
542	23	22	9	542	1710	1125
710	26 1/2	22	9	710	2250	1120
903	27 1/2	26	10 1/2	903	3000	1125
1163	30 3/8	26	10 1/2	1163	4050	1110
1545	30 3/8	32 1/2	11	1545	5400	1125
2015	30 3/8	36 1/2	11	2015	6540	1110

All above models are available with variable speed motors. Units for hot water application also available.

structurally but assure far more effective, economical distribution of heat: (1) The **Expansion Bend** given each tube of the condenser before entering lower header, provides for free expansion, thus eliminating expansion strain from being transferred to header tanks. (2) **Velocity Generator** for redirecting and controlling the heated air stream, assures greater possible throw consistent with comfortable heating. (3) **Direct Pipe Suspension** greatly facilitates horizontal redirection of the heated air stream, permitting full 360 deg rotatability. Also means far easier installation, at less cost—no brackets, pipe rods or straps being necessary. Write for Unit Heater Catalog 635.

MODINE COPPER CONVECTORS

A convector comparatively light yet unusually sturdy in construction. Available in a wide range of sizes and four general types—Concealed, Recessed, Floor Cabinet and Wall Cabinet. For steam, vapor or vacuum systems; line also includes a hot water type copper radiator designed specifically for hot water systems. Write for Catalog 135.

MODINE COPPER-CAST RADIATORS

A combination of copper and cast-iron that combines convected and radiated heat for greatest comfort and economy. Modine Copper-Cast heats faster than cast-iron because of the copper convection unit, and cools much slower than unassisted convectors, thus maintaining a more even temperature than either.

Write for Catalog 135.

MODINE HUMIDITIONER

The Humiditioner, a low cost domestic air conditioner used in conjunction with a radiator heating system. It humidifies, cleans, circulates heated air; can be furnished with cooling coil for living quarter cooling. The Humiditioner is ordinarily installed in the basement, suspended from the ceiling where it takes up no floor space; gives benefits of air conditioning without sacrificing virtues of radiator heating.

Write for Catalog 135.

OTHER MODINE PRODUCTS

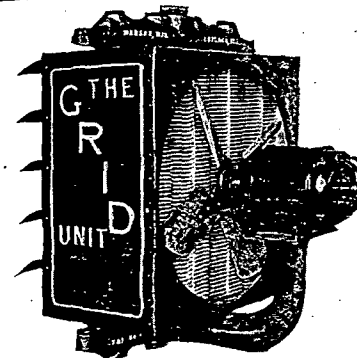
Modine also offers complete lines of Blast Heaters, Cooling Coils and Unit Coolers. Data gladly furnished on request.

The Unit Heater and Cooler Co.

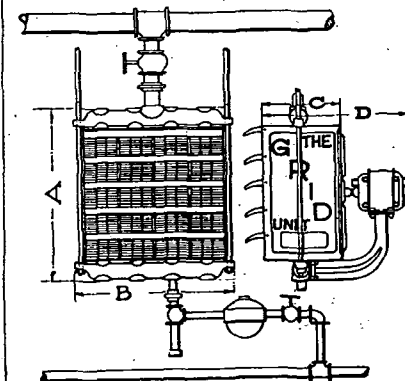
Wausau, Wisconsin

Offices in Principal Cities

MANUFACTURERS OF THE GRID UNIT



Sturdy—Permanent
The Grid Unit Heater
(A product of the D. J. Murray Mfg. Co.)



Typical Installation of Grid Unit Heater

GRID UNIT HEATER DATA

Delivers More Warm Air to Floor and Working Zone *Varies with type of motor.

Model No.	Dimensions				Face Area Sq. Ft.	Motor		Volume at Fan	Capacities 5 Lb. Press. 60° Air		Approx. Shipping Weight	Pipe Sizes	
	A	B	C	D*		Hp.	R.P.M.		B.T.U.	Final Temp.		Supply	Outlet
1015	11 3/4	15 1/2	9	14 1/2	.52	1/30	1600	220	19450		90	3/4"	3/4"
1020	11 3/4	20 1/2	9	14 1/2	.69	1/30	1500	285	24200		100	3/4"	3/4"
1200	15	12 1/2	11 1/4	17 1/2	1.04	1/20 1/30	1700 1150	711 515	46000 34900	119 122	120	1 1/4"	1 1/4"
1500	22	18	11 1/2	20	1.67	1/10 1/20	1750 1150	1450 1015	77500 65500	109 118	210	1 1/2"	1 1/4"
1520	27	18	11 1/2	20	2.2	1/8 1/20	1750 1150	1700 1288	104000 83500	113 118	250	1 1/2"	1 1/4"
2000	27	23 1/4	11 1/2	21 1/2	2.8	1/6 1/10	1150 850	2500 1835	148000 113000	114 117	320	2"	1 1/4"
2025	32	23 1/4	11 1/2	21 1/2	3.6	1/6 1/10	1150 850	2875 2380	177000 158500	115 120	370	2"	1 1/4"
2500	32	28 1/2	11 1/2	28	4.5	1/2 1/4	1150 850	4200 3375	225000 197000	108 113	440	2"	1 1/4"
2530	36	28 1/2	11 1/2	28	5.3	1/2 1/4	1150 850	4650 3350	282000 226000	115 121	500	2"	1 1/4"
3000	38	33	13 3/4	29	6.5	1 1/2 1/2	1150 850	8100 6350	394000 341000	104 109	725	2 1/2"	1 1/4"

Grid Unit Heaters—Reduce fuel cost and maintenance expense—no damage possible to Grid due to electrolytic action as it cannot take place in Grid Units.

Long Life—Efficient Service—Good for 250 lb. working pressure.

Manifolds or Headers and Steam Chamber—Cast High Test Iron.

Fin Surface—Aluminum alloy cast metal to metal bond with steam chamber.

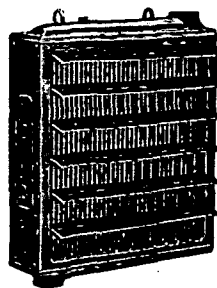
Lower outlet temperatures and greater volume delivers warm air to working zone of room, thus eliminating heat losses at ceiling and avoiding stratification.

YOUNG RADIATOR Company

Racine, Wis.

Representatives in All Principal Cities

Manufacturers of Young Unit Heaters—Convection Heaters—Blast Units—Unit Coolers—Evaporators—Commercial Heat Transfer Units



Model "SH" Unit Heater

The Young line of heating and cooling equipment is complete. Thus the architect, heating contractor or engineer may select Young products to meet his most exacting specifications.

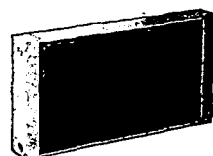
There are 24 different models of **Suspended Unit Heaters** equipped with constant or variable speed motors. **Floor Type Unit Heaters** have capacities from 120,000 to 800,000 Btu per hour.

Young Blast Units and **Commercial Heat Transfer Surfaces** may be used for various applications operating in conjunction with heating, cooling, or air conditioning systems.

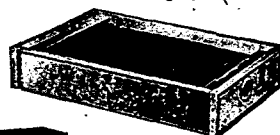
Young Unit Coolers are used with cold water, brine, or any common refrigerant.

Young Evaporators for use with Freon or Methyl chloride are scientifically designed to distribute the liquid with one or a group of expansion valves.

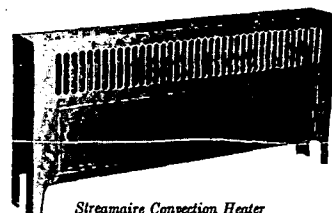
There are four distinct types of **Convection Heaters**—free standing cabinets, recess cabinets, wall hung cabinets, and plastered-in enclosures. All models incorporate the **Streamaire** design with large elliptical tubes which make Young convectors the outstanding installation for use with one-pipe or two-pipe steam, vapor, vacuum, or hot water heating systems.



Commercial Heat Transfer Unit

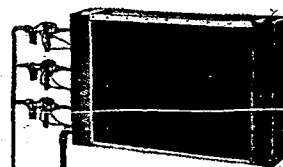


Blast Unit

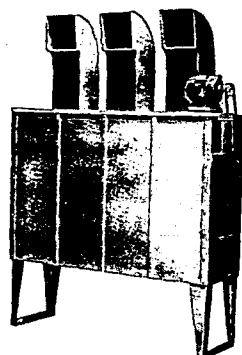


Streamaire Convection Heater

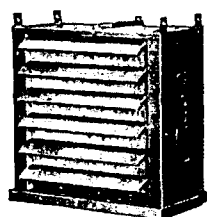
Additional Information
Furnished
on Request.



Evaporator



Model "FH" Unit Heater



Unit Cooler

AEROFIN CORPORATION

850 Frelinghuysen Avenue

Newark, N.J.

Manufacturers of **AEROFIN**

The Standardized Light-Weight Encased
Fan System Heating and Cooling Surface

11 West 42nd Street, NEW YORK

Land Title Building
PHILADELPHIA

United Artists Building
DETROIT

Burnham Building
CHICAGO

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.

Flexitube AEROFIN

FLEXITUBE AEROFIN: (Fig. 1) supplants the original non-corrodible Low Pressure AEROFIN.

Flexitube AEROFIN is distinguished from all other developments by its off-set tubes.

This flexible single-pass, header-to-header design so perfectly relieves or absorbs expansion and contraction strains that header and header-joint construction adequate for high pressure service may be employed, and the permanency of the construction assured thruout the practically unlimited life of the surface.

Single-pass tube construction is essential for low pressure service and is desirable for intermediate pressures (up to 200 lb.).

The headers are one-piece bronze castings, machined for tube joints and pipe tappings.

The joints between the copper tubes and the headers are of a patented type made by using a brazing material applied at high temperature and have proved stronger than the tube itself under all operating conditions for which AEROFIN is sold. We believe this joint to be the most practical and dependable yet developed.

The arrangement of Supply and Return connections readily lends itself to every installation requirement and permits installation in any position, tubes vertical or horizontal, or Units "laid flat" for vertical air flow.

Uniform steam or cold water distribution thru every tube is assured by proper orifice restriction at supply ends of tubes.

Design and Construction: The heat-transfer surface in AEROFIN is a plurality of seamless copper tubes about which is wound a helix of copper ribbon, approximately smooth.

The extended fin surface is applied and tinned while held in position, by highly developed automatic machines. The tinning of the tube and the extended surface makes them metallicly integral, affording maximum heat transmission and permanent effectiveness.

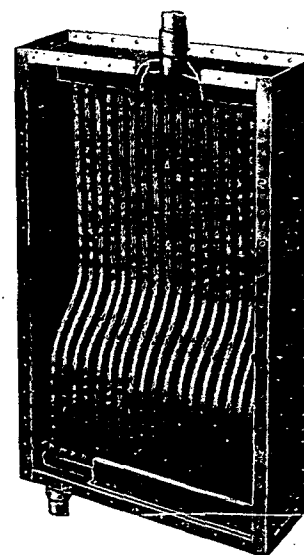


Fig. 1

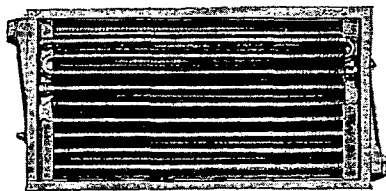


Fig. 2

Flexitube AEROFIN can also be made all aluminum—or admiralty tubes can be furnished.

Flexitube AEROFIN is constructed as briefly described above. In Universal AEROFIN and High Pressure AEROFIN the seamless tubes, with their extended fin surface, are continuous. (See Figs. 2 and 3.)

Standard Casings: The casings of all AEROFIN Units are 29 in. wide (except 6-tube Universal AEROFIN which is 21 $\frac{1}{2}$ in.) across tubes, and 12 tubes wide Flexitube which is 20 $\frac{3}{8}$ in. across tubes, from outer edge one flange to outer edge opposite flange, and 10 in. deep in direction of air flow. Length of casing is nominal tube length plus 8 $\frac{1}{2}$ in.

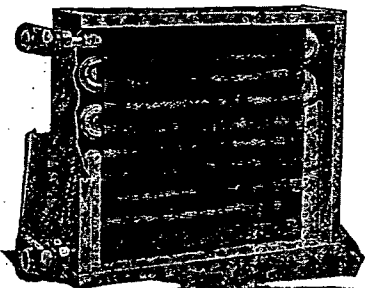


Fig. 3

Aerofin Sizes: FLEXITUBE AEROFIN: Made in thirteen standard tube lengths, and in 2 widths, either one or two staggered rows of tubes per Unit, in 5 series (as explained in Bulletin).

Universal Aerofin: (Fig. 2). Available in one-row or two-row Units, 6-tubes or 9-tubes across face, in seventeen standard tube lengths, between 2 and 10 ft., inclusive.

High Pressure Aerofin: (Fig. 3). Made in thirteen standard tube lengths 2 ft.; to 8 ft., either one, two or three staggered rows of tubes per Unit.

Aerofin Encased Booster Units: (Fig. 4). For horizontal or vertical air flow.

Six sizes 150 to 1624 c.f.m., 200 lb. working steam pressure.

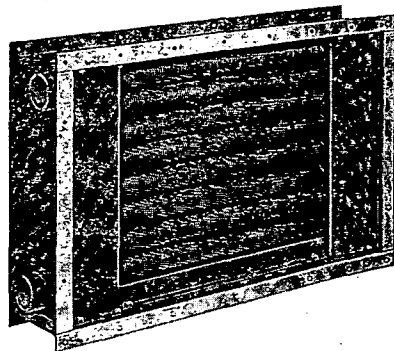


Fig. 4

Aerofin Continuous Tube Water Coils: Are designed for air cooling by circulating cold water through the AEROFIN and air over the extended fin surface. These units can be made for either horizontal or vertical air flow.

Tubes and fins are made of copper, completely tinned with a permanent metallic bond between fin and tubes. Headers are made of one-piece cast bronze, and casings of heavy galvanized iron.

Every unit is tested to 1000 lb. hydrostatic pressure.

For details see AEROFIN Continuous Tube Water Bulletin.

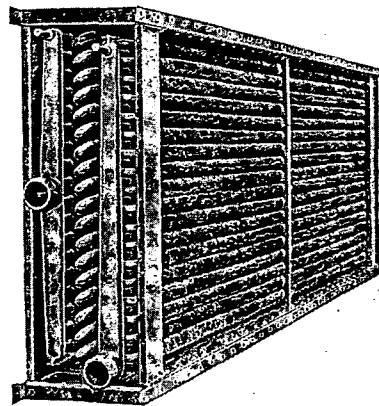


Fig. 5

Size (6) Row Continuous Tube Water Coil with front end cover removed, showing supply and return header with connections for air vents. This is a left-hand unit (supply header on left side) for air flow from right to left.

Narrow Width Aerofin: (Fig. 6), is to be used for Cooling only. Depth of casing is 5 in. instead of 10 in. This saves half the space required, in direction of air flow. Straight tubes. Made in one or two rows deep only.

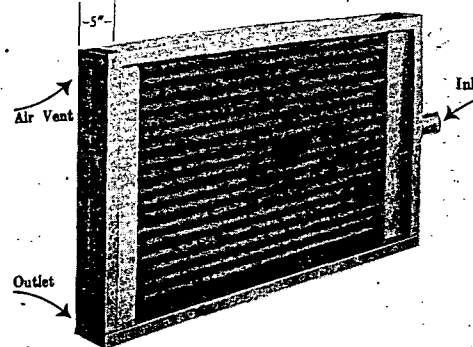


Fig. 6

Aerofin Direct Expansion Cooling Units

Aerofin Direct Expansion Cooling Units constitute a new development in Surface Cooling. (See Fig. 7.) Available for use with Freon or Methyl Chloride, these units are designed for the maximum cooling effect in a given space. The novel and unusual design provides for a maximum of operating efficiency by supplying refrigerant for each row of tubes from a header to several passes in the face of the unit, thus maintaining full cooling effect with wet gas throughout all the tubing, except for a necessary slight length at the end of each pass to provide a sufficient degree of superheat to insure dry gas entering the suction header.

Each Row is entirely separate. Units are built with either one, two or three rows of tubes and by bolting two units face to face, 4 rows, 5 rows or 6 rows may be had two units in depth.

Units are made in eleven lengths and two widths, which with 1 row, 2 rows, 3 rows, 4 rows, 5 rows and 6 rows deep, makes available 110 combinations.

Then several units may be installed in height and two units in width (since supply and return headers are on same end of each unit), so that all varieties of installation may be furnished.

Units are designed so that each row of tubes may be controlled separately.

All headers are equipped with standard Mueller fittings, which may be reduced to size desired by using standard sweated

reducing fittings, or adaptors to compression fittings.

The design of Aerofin (Direct Expansion) allows a wide selection of independent circuit arrangements for the use of one or more compressors of the same or of different capacities operating entirely independent of each other.

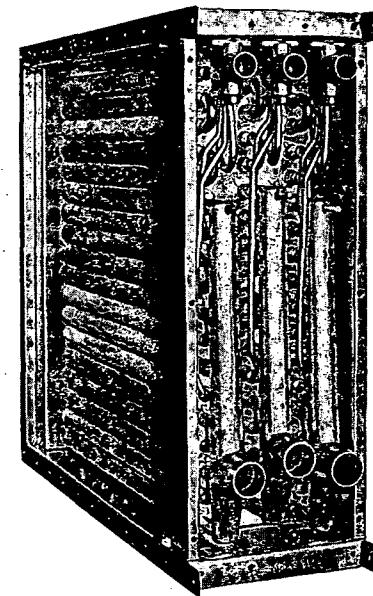


Fig. 7

End plate removed showing distributing and suction headers

Steel Supporting Legs: Standard steel supporting legs, 18 or 24 in. high, template punched to same bolt hole centers as standard casing, are furnished when ordered. These legs may be attached quickly and obviate necessity of any other foundation.

Advantages: AEROFIN weighs but 9 to 16 per cent as much as equivalent cast-iron and occupies about $\frac{1}{3}$ of the space. Expensive foundations are unnecessary, building re-enforcement is not required and the Units may readily be suspended from beams or roof trusses, or installed snugly in any out-of-the-way corner.

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus. List upon request. Ask Newark for Heating Bulletin G32 at once, or Direct Expansion Bulletin D-E 34 or for Bulletin, C. T. 35. Water Coils.

The G & O Manufacturing Company

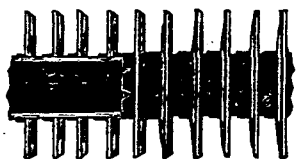
138 Winchester Avenue

New Haven, Connecticut

G&O

INDIVIDUAL FIN TUBING

RADIATING ELEMENTS FOR UNIT HEATERS (HIGH AND
LOW PRESSURE) AND CONVECTION HEATERS



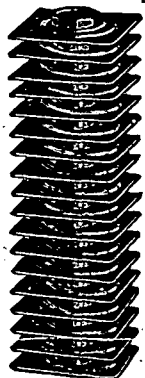
INDIVIDUAL G & O FINS—The use of individual fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface, because all G & O fins have ample collars around the tube opening insuring liberal contact with tubing.

ANY SIZE OR SHAPE—Fins of any size or shape may be obtained giving any desired proportion of primary and secondary radiating surface.

SQUARE FINS—A square fin has about 30% greater surface than a round fin of a diameter equal to a side of the square.

INDIVIDUAL FINS—Individual fins permit of any fin spacing; also, of using fins in groups at intervals along tubes. Fins placed at practically any angle on tubes.

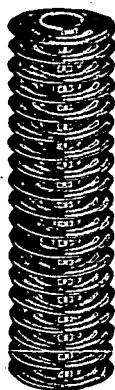
VARIOUS SHAPES—G & O individual fin tubing is furnished in straight lengths; in U bends with short radii; in continuous return bend coils, or other shapes. Ends of tubes may be left free of alloy for mechanical joints.



G & O Individual Fin Tube Data

Standard Sizes

O.D. of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8"	3/4" sq.	6	0.59 sq. ft.
3/8"	7/8" sq.	6	0.81 sq. ft.
3/8"	7/8" r.d.	6	0.64 sq. ft.
3/4"	1 1/2" r.d.	6	1.61 sq. ft.
3/4"	1 1/2" sq.	6	2.50 sq. ft.
1"	2 1/2" sq.	6	4.00 sq. ft.



GRINNELL COMPANY, INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers,
Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: Providence, R. I.

Offices, Plants and Branches

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ATLANTA, GA. (Plant and Foundry)	COLUMBIA, PENNA. (Plant)	NEW YORK, N. Y.
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GRINNELL COMPANY OF THE PACIFIC

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GRINNELL COMPANY OF CANADA, LTD.

MONTREAL, QUE. (Branch) VANCOUVER, B. C. (Branch) TORONTO, ONT. (Plant and Foundry) WINNIPEG, MAN. (Branch)
OSHAWA, ONT. (Foundry)

PRODUCTS AND SERVICES—

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 Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoflex (Unit Heaters); Grinnell Thermofin (Convectors); Grinnell Unit Coolers (for refrigerating service); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Constant Level Size Circulating 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.

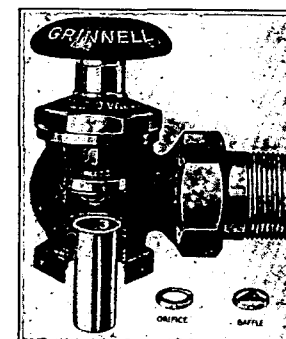
Valves: Check, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell "Junior" Automatic Sprinkler Systems for Basements and other hazardous areas of Dwellings, Small Apartment Buildings, Schools, Churches, Stores, etc.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

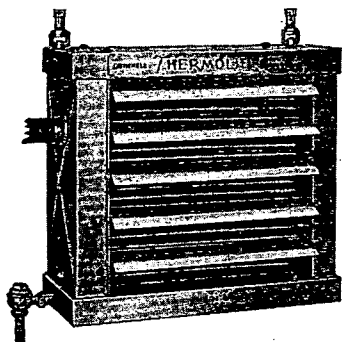
The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems. This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices 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 1015

THERMOLIER

Patented

THE GRINNELL UNIT HEATER**Industrial and Factory Types—125 Lbs. W.S.P.**

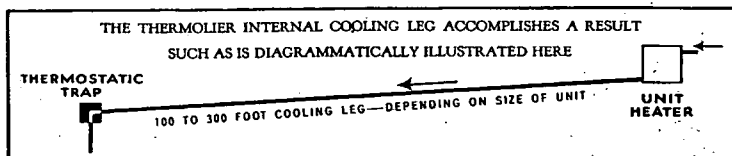
Industrial Type

Thermolier is a ruggedly built unit heater whose efficiency and dependability have been proved by actual performance in field service. Thousands of them are installed in industrial buildings and commercial structures of all types of occupancy.

Thermolier has 14 points of superiority, the most outstanding of which is the internal cooling leg built right into the unit, an exclusive Thermolier feature. See drawing below.

Radiation is from brass-finned seamless copper U-tubes rolled into a cast-iron tube sheet. No solder is used for strengthening joints and there are no flat horizontal surfaces to catch dirt.

Units may be controlled manually or automatically, singly or in groups. Installation and piping are extremely simple and inexpensive, hence the unit may be moved from one



location to another at small cost if found desirable on account of changes in building or occupancy. The complete line includes 22 Models in Two Types.

Thermoliers provide maximum distribution of heat without objectionable drafts.

Specifications

Fan—Grinnell special of rugged construction. **Motor**—heavy duty, oversize, enclosed, moisture-proof. **Housing**—Copper on Industrial Type with rubbed lacquer finish; steel on Factory Type finished in gray duco. **Frame**—Heavy pressed steel, providing rugged support for motor and fan. **Special Features**—Adjustable swivel hanger rod couplings; louvers rigid, but easily adjustable; integral cooling leg insuring perfect drainage through one thermostatic trap for pressures up to 25 lb.

For pressures not exceeding 125 lb., a thermostatic trap of proper construction can be used and should be attached directly to the unit.

CAPACITIES

60° F. Entering Air Temperature—2 lbs. Steam Pressure

Model Nos.	B.t.u. per Hour	Model Nos.	B.t.u. per Hour	Model Nos.	B.t.u. per Hour
20	22,700	60	104,800	100	234,000
20L	17,200	60L	77,700	100L	196,000
30	31,200	70	117,000	140	320,000
30L	23,200	70L	100,000	140L	271,000
40	69,400	80	164,600	180	368,000
40L	53,300	80L	139,300	180L	294,000
50	90,700	90	189,200		
50L	67,100	90L	151,600		

Data Book covering other pressures and temperatures, dimensions and complete installation information on application. Address GRINNELL COMPANY, INC., 277 West Exchange Street, Providence, R. I.

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.

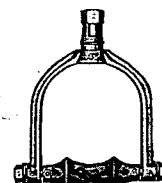
Adjustable Swivel Rings (Patented)Fig. No. 101
Solid Ring

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)**

An adjustable type of pipe roll using a single hanger rod. Swivel Shank allows vertical adjustment and automatically locks, preventing loosening from vibration.

Fig. No. 174
Swivel Pipe Roll**Universal Concrete Inserts (Patented)**

Made of air furnace malleable iron, in one body size, to take a special removable nut, tapped for ¾ in., ½ in., ⅝ in. or ¼ in. rod as required. Nuts automatically lock by means of V-type teeth on both insert and nuts.

Fig. No. 232
Universal Insert**GRINNELL WELDING FITTINGS**

Grinnell Welding Fittings are made from Seamless Steel Pipe 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.

Welding faces for all plain circumferential Butt Welds are scarfed or beveled to the regulation 45 deg. angle with ⅛ in. square end on inside of fitting. Angles of bevel other than 45 deg. can be furnished on special orders.



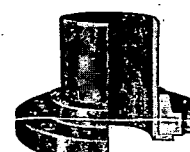
90° Elbow, Long Turn



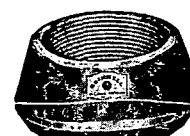
Welding Outlet



Welding Tee



Lap Flanged Welding Neck



Threaded Outlet

John J. Nesbitt, Inc. AND Buckeye Blower Company

Manufacturers of Heating, Ventilating
and Air Conditioning Equipment

EXECUTIVE OFFICES

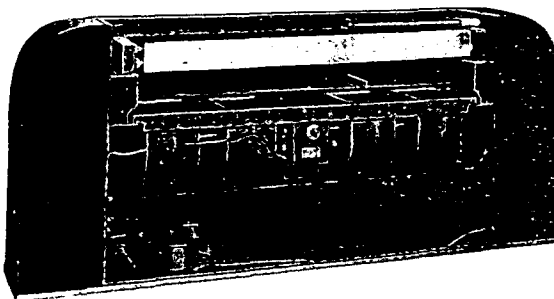
State Road and Rhawn Street
Holmesburg, Philadelphia, Pa.

Buckeye Sales and Service Offices in Principal Cities of U. S. A.

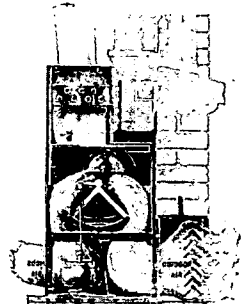
Sales and Service on Nesbitt School Room Unit Ventilators
through Offices of American Blower Corp.

FACTORIES

Holmesburg, Philadelphia, Pa.
Columbus, Ohio



Below — Cross-section view
showing the Nesbitt Syncr-
etizer mixing outdoor and
room air



Nesbitt Series 400 Syncretizer

Called TOMORROW'S Heating and Ventilating Unit TODAY, because of its easy adjustment to present or future ventilation requirements. Made in three types or cycles of control—F, A, and O. All three types recirculate all room air during the heating-up period, before room occupancy. Type F, during room occupancy, introduces all outdoor air. Type A, during room occupancy, introduces any desired minimum quantity of outdoor air when heating is required, and an increasing quantity, up to all outdoor air, when cooling is required. Type F may be readily converted to Type A cycle of control at any time by a simple adjustment within the unit.

Type O, during room occupancy, introduces a variable quantity of outdoor air without a fixed minimum. The quantity supplied is governed by the indoor and outdoor temperatures, and is that volume required to maintain a minimum air-stream temperature—usually 60 deg—at the fan discharge, before the air passes through the unit radiator.

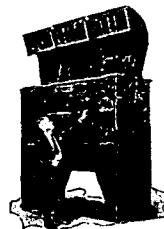
All three types are made in the same size and style of casing for a given capacity, and all have the same arrangement of component parts. The differences are in the cycles of control. All three have the Nesbitt Airstream Minimum Temperature Control to prevent cold drafts. Ask for Publication 225 (engineering data), and Publication 226 (The Story of Syncretized Air).

Nesbitt Series B Thermovent—for Large Interiors

A large-volume unit ventilator incorporating three distinct features which have been proved essential to the proper heating and ventilating of auditoriums, gymnasiums, assembly halls, and like gathering places: (1) *Quietness of operation*, due, in part, to large-diameter, forward-curved fans operated at low speeds, with V-belt drives; (2) *Modulated Steam Valve Control*, with uniform discharge temperatures over the entire radiator assured by Nesbitt internal steam-distributing tubes; and (3) *Air-Stream Minimum Temperature Control*, the Nesbitt development which prevents cold drafts.

Furnished, like the Nesbitt Series 400 Syncretizer, in three types or cycles of control—F, A, and O—to deliver all outdoor air, or to partially recirculate, with or without a fixed minimum quantity of outdoor air. Available in capacities ranging from 2000 to 6000 cfm, and in two types of casing, for exposed or hidden location. Ask for Publication 227.

Buckeye Giant Unit Heater



A blower, draw-through type unit heater for the economical heating of large areas in industrial plants, garages, airplane hangars, etc. Made in a variety of capacities, ranging from 1880 cfm, 139,500 Btu, to 19,600 cfm, 1,290,000 Btu, with 2 lb steam, 60 deg entering air.

Giant Heaters are equipped with Nesbitt Copper Finned Tube Radiating Surface for operating steam pressures up to 150 lb gage. Also applicable for use on Forced Circulation Hot Water Systems.

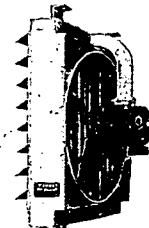
Giant Heaters are built for Floor Mounting—Wall Mounting (both Standard Vertical and Inverted Vertical) and Horizontal Suspended.

Wall boxes and outside air dampers can be furnished with all types.

Giant Heaters are furnished with or without the Thermostat Damper. This efficient bypass damper with automatic or hand control prevents overheating and stratification and effects substantial fuel savings by controlling heat output in definite relation to demand. Giant Heaters may be had with either V-belt drive or direct-connected motor. Ask for Buckeye Publication 145.

Giant Unit Heaters are tested and rated in accordance with the Standard Test Code of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Industrial Unit Heater Association.

Buckeye Unit Heater



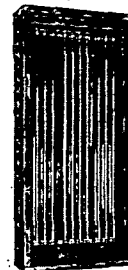
A rugged, compact, suspended type, disc fan unit heater, built in capacities ranging from 810 cfm, 35,800 Btu, to 5030 cfm, 313,500 Btu, with 2 lb steam, 60 deg entering air, and for operating steam pressures up to 150 lb. Also adaptable for use on Forced Circulation Hot Water Systems. Available in either single- or multi-speed.

Buckeye Unit Heaters may be conveniently located to supply heat exactly where it is needed, and the louvers are individually adjustable to direct the air flow to the proper level. Heavy brackets are provided at the top of each heater for attachment to hanger rods.

Recirculating ducts or fresh air intake ducts, together with the necessary wall boxes and louvers, will be furnished with these heaters when specified. The propeller fans used have been specially developed for these heaters and they combine lightness and durability with exceptionally high aerodynamic efficiency. They are perfectly balanced and are securely attached to the motor shaft. Ask for Buckeye Publication 144.

Buckeye Unit Heaters are tested and rated in accordance with the Standard Test Code of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Industrial Unit Heater Association.

Nesbitt Heating Surface with Steam-Distributing Tubes



The increasing application of automatic temperature control to heating, ventilating and air conditioning systems has created the need for this new type of blast steam heating surface. Nesbitt Copper Heating Surface, through the use of all-copper, fin-and-tube construction, retains the rapid steam condensing capacity and the permanence proved for light-weight copper radiation, and gains unique *controllability* by inserting, within the usual steam tubes, additional or steam-distributing tubes. These carry the steam equally to all parts of the radiator section, regardless of the quantity supplied. Thus, whether the section is operating at full

capacity or merely tempering the incoming air, a uniform discharge temperature is assured. Possibility of freezing is eliminated under normal operating conditions. Nesbitt Copper Heating Surface is made in a wide range of capacities, to meet all conditions. Fins are made fast to tubes through mechanical construction processes, thus eliminating the use of solder or other bonding materials. Ask for Publication 229.

Bell and Gossett Company

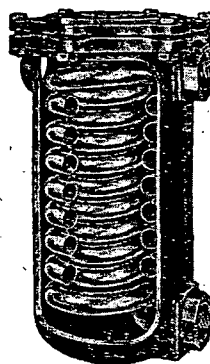
3000 Wallace Street

Chicago, Ill.

HOT WATER SYSTEMS AND SPECIALTIES

B & G INDIRECT WATER HEATERS FOR STEAM AND VAPOR SYSTEMS

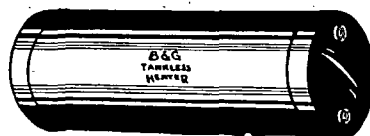
Ratings below are based on 100 degree rise in three hours with boiler water temperature of 180 degrees or more



Description	No.	Capacity Gallons	Max. Length Inches	Max. Width Inches	Shell Openings Inches	Coil Openings Inches	Shipping Weights Pounds
SINGLE COIL For residences of all sizes. Duplex apartments and small buildings.	30	30	11 1/4	5 1/2	1	3/4	12
	40	40	12 1/4	5 1/2	1	3/4	13
	52	52	14 1/4	7 1/4	1 1/2	1	15
	66	66	15 1/2	7 1/4	1 1/2	1	29
	82	82	18 1/4	7 1/4	1 1/2	1	39
	100	100	20 1/4	7 1/4	1 1/2	1	40
DOUBLE COIL For larger apartments, garages, medium sized factories and office buildings.	120	120	22 1/4	7 1/4	1 1/2	1	42
	144	144	25 1/4	7 1/4	1 1/2	1	46
	160	160	12	11 1/2	2	1 1/2	53
	192	192	14 1/4	11 1/2	2	1 1/2	59
TRIPLE COIL For heavier requirements.	300	300	19 1/4	11 1/2	2	1 1/2	78
	400	400	23 1/4	11 1/2	2	1 1/2	86
	600	600	21 3/4	15 1/4	3	2 1/2	230
DOUBLE TRIPLE COILS	800	800	25 3/4	15 1/4	3	2 1/2	290
	1000	1000	29 1/4	15 1/4	3	2 1/2	290
	1200	1200	23 1/4	29	4	3 1/2	568
DOUBLE TRIPLE COILS	1600	1600	27 1/4	29	4	3 1/2	642
	2000	2000	30 1/4	29	4	3 1/2	716

B & G TANKLESS HEATER

This B & G Tankless Type Heater has been designed to fulfill a need for a heater of unusual capacity that could be installed in boiler rooms lacking space for storage tanks.



For all Steam, Vapor and Hot Water Heating Boilers

Size No.	Heating Surface in Square Feet	Capacities in Gallons in 1 Hour Heated from 50 F to 130 F		
		Boiler Water at 212 F	Boiler Water at 200 F	Boiler Water at 180 F
12	12	207	168	100
20	20	345	288	168
30	30	516	430	250

B & G THERMOCHEK

Controls automatically the temperature of hot water storage tanks heated by indirect heaters. A simple, completely automatic device that eliminates lime and sediment formation in the heater coils. Sizes 1 1/2 in. and 2 in.



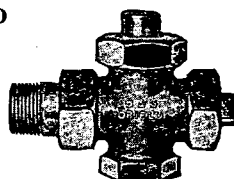
B & G MONOFLO FITTING

This fitting is used on mechanically - circulated one-pipe hot water systems. When installed in the main at each radiator connection, it assures a balanced system with short circuits eliminated and all radiators receiving their full quota of hot water.

No.	Size, Inches
1 1/4	1 1/4
1 1/2	1 1/2
2	2

B & G ORIFLO FITTING

The Oriflo fitting is for use on hot water radiators. New or old gravity or forced circulation jobs can use this device for accurately regulating the temperature drop through the radiators, thereby balancing the heating system quickly with minimum expense and without shutting down the heating plant.



B & G TRIPLE DUTY SYSTEMS FOR HOT WATER SYSTEMS

Size No.	Recommended Tank Sizes	Capacities Gals. per Hour	*Capacities Sq. Ft. E.D.R.	Shipping Weight Pounds
15	30 gals.	30	1,500	152
16	45 gals.	35	1,500	153
17	66 gals.	40	1,500	155
18	88 gals.	50	1,500	159
19	30 gals.	30	3,000	158
20	42-66 gals.	40	3,000	161
21	66-88 gals.	50	3,000	165
22	66-120 gals.	70	3,000	183
23	120-180 gals.	100	3,000	198
24	180-300 gals.	140	3,000	213
25	300 to 400 gals.	200	6,000	445
26	400 to 500 gals.	266	6,000	465
27	500 to 600 gals.	333	6,000	505

*Capacities in square foot of direct radiation are based on 200 Btu radiator emission with a 20° F. drop.

B & G BOOSTER

Size No.	Face to Face Dimensions	Flange Size	Motor 110 Volt 60 Cycle	Normal Delivery Gallons per Minute	Shipping Weight
1"	7 1/4"	1 1/2" Screwed	1/8 hp.	15 at 2 ft. head	70 lbs.
1 1/2"	8 1/2"	1 1/2" Flanged	1/8 hp.	30 at 2 ft. head	80 lbs.
2"	8 1/2"	2" Flanged	1/6 hp.	60 at 4 1/2 ft. head	83 lbs.
3"	12"	3" Flanged	1/3 hp.	100 at 5 1/2 ft. head	135 lbs.

B & G MOTORIZED VALVES

These valves are guaranteed to stop the flow of water in the heating mains of hot water systems. They are used for zone control and in year around, indirectly heated domestic hot water systems. Operation is by a 60 cycle, A.C. 25-volt induction type motor.

Series No.	Description	Shipping Weight Pounds
1 1/4	Single 1 1/4 in. Main	19
1 1/2	Single 1 1/2 in. Main	20
2	Single 2 in. Main	23
2 1/2	Single 2 1/2 in. Main	35
3	Single 3 in. Main	40
4	Single 4 in. Main	78
5	Single 5 in. Main	140
6	Single 6 in. Main	190

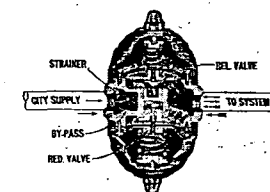
D & T SELF-FILLING, AIR-CUSHIONED TANK EQUIPMENT

Self-Filling Air-Cushioned Tank Equipment consists of a D & T Air Tank and a D & T No. 500 "All In One Unit" which is a combination relief and pressure reducing valve, performing the double duty of relieving excess pressure and keeping the system filled with water.

System No.	Capacity Sq Ft Radiation	System No.	Capacity Sq Ft Radiation
518	Up to 500 ft	540	1600 to 2000 ft
524	500 to 800 ft	560	2000 to 3000 ft
530	800 to 1200 ft	580	3000 to 4000 ft
535	1200 to 1600 ft	600	4000 to 5000 ft

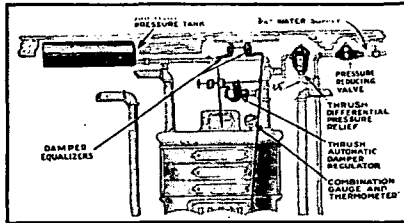
D & T SIMPLEX TANK-IN-BASEMENT SYSTEM

This system consists of an air-tight Compression Tank, Relief Valve (illustrated), Thermometer, Gage and Vacuum Breaking Valve. Made in 14 different sizes with capacities up to 5000 sq ft of radiation.



H. A. Thrush & Co.
Peru, Indiana

THRUSH TANK IN BASEMENT SYSTEM



Illustrating Class AA System

Thrush System of Hot Water Heating provides an automatically controlled heating plant, operating with high efficiency at low fuel cost. Circulation is increased, resulting in even and quick heat distribution. Installation cost is reduced because small pipe and valve sizes are required when Thrush System is used.

Thrush System may be installed with either old or new Hot Water Heating plants. It converts a gravity type job into a Closed System with accelerated circulation and increased heat transmission.

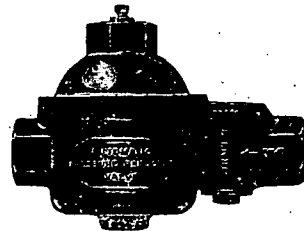
Thrush Tank in Basement Systems are made in different types to care for hot water heating jobs of every conceivable kind, all providing the advantages of the closed system with controlled circulation. Each system is made in sizes to care for the needs of any job. It is possible to provide varying degrees of automatic operation, damper control, automatic system filling, automatic pressure relief, etc., to meet the demands of any job or to offset any competition.

In all Thrush Systems the Thrush Pressure Tank takes the place of the ordinary gravity or open expansion tank. It is near the boiler where it cannot freeze. It provides a reservoir to take care of water expansion and to avoid waste of heated water. Being under pressure, the system circulates more positively and rapidly, saving fuel. Easy boiling is prevented and a wide range of boiler temperatures from 120 to 220 F. may be carried without noise or boiling. This gives the job extreme flexibility of performance and the highest thermal circulation efficiency possible to secure.



No. 4—Differential Pressure Relief Valve
Tapped 1/4 in.

No. 4. Differential Pressure Relief Valve for Hot Water Heating System is safe and dependable. It has a large metal diaphragm which actuates movement of a free working valve member—has no tight-fitting guide. It is fully guaranteed when properly installed. Set to open at 28 lb. pressure, suitable for 1, 2, or 3 story buildings. Special adjustment up to 35 lb. Weight 12 lb.



No. 12—Pressure Reducing Valve
Tapped 1/4 in.

No. 12. Pressure Reducing Valve provides automatic filling and maintenance of proper water supply in a Hot Water Heating plant. Large waterways. Working parts brass. Fully guaranteed against defects in workmanship and materials. Pressure setting suitable for 1, 2, or 3 story buildings. Weight 7 lb.

H. A. Thrush & Co.
Peru, Indiana

THRUSH FLOW CONTROL SYSTEM—For Automatic Hot Water Heating

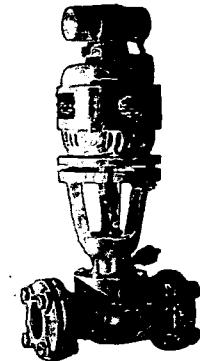
Every Boiler Burner Unit Hot Water Heating Plant should be equipped with Thrush Flow Control System because it makes the job heat quickly, maintains healthful temperatures automatically, increases the efficiency of the burner, permits economical year round use for Summer-Winter domestic hot water supply and saves fuel. The water in the boiler is maintained at a constant temperature and cannot pass through the Flow Control Valve to the radiators when heat is not needed. When heat is required, the thermostat operates the Thrush Circulator, which quickly forces hot water through the Flow Control Valve to every radiator in the house, restoring room temperature almost instantly. Then the Flow Control Valve shuts off all further circulation, preventing overheating and attendant waste of fuel. Running time of burner is reduced.

THRUSH WATER CIRCULATORS

Silent, Efficient, Electric Motor Driven Water Circulating Pumps

Thrush Water Circulators are correctly designed to lift the load of cold returns and assure instantaneous heat delivery. They are made in four sizes, as listed below. Motors are 110 V. 60 Cycle, A.C., Capacitor type, silent in operation and free from radio interference.

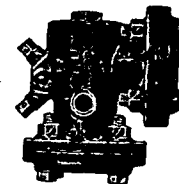
No.	Size	Motor	Gallons Delivery	Radiation Capacity	Shipping Weight
14	1 in.	1/10 Hp	25 per Min	500 sq ft	53 lb
15	1 1/2 in.	1/6 Hp	50 per Min	1500 sq ft	72 lb
22	2 in.	1/6 Hp	75 per Min	2500 sq ft	74 lb
23	3 in.	1/4 Hp	140 per Min	5000 sq ft	94 lb



The complete Thrush Flow Control System consists of a Thrush Water Circulator of the proper size, a Thrush Flow Control Valve of proper size, a No. 12 Thrush Pressure Reducing Valve, a No. 4 Thrush Differential Pressure Relief Valve and a special Thrush Air Tight Pressure Tank of proper size. This system converts an open gravity job to a closed pressure system with automatic filling and maintenance of proper water supply, automatic relief of excess pressures, adequate provision for expansion when the water is heating and complete mechanical control of circulation. When used in conjunction with the regular thermostatic controls with which most automatic heating plants are already equipped, the heating plant becomes completely automatic, flexible and quick to respond to the slightest need for heat.

THRUSH FLOW CONTROL VALVES

The Thrush Water Circulator and the Thrush Flow Control Valve used together provide complete circulation control as the Flow Control Valve works by pressure head generated by the Circulator. There are no complicated fittings. Installation is inexpensive and easy on old or new jobs. Flow Control Valve has expansion opening to provide for escape of excess pressure and elimination of air and gases. (Manually operated set screws permit closing or opening permanently but are not needed for normal operation).



PATENT No. 1,931,419

No.	Size	Connection	Radiator Capacity	Shipping Weight
114	1 in.	Screwed	300 sq ft	3 lb
115	1 1/2 in.	Screwed	1000 sq ft	9 lb
120	2 in.	Flanged	1500 sq ft	19 lb
125	2 1/2 in.	Flanged	2500 sq ft	26 lb
130	3 in.	Flanged	5000 sq ft	39 lb

Davis Engineering Corporation

HEAT TRANSFER SPECIALISTS SINCE 1915

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90 West Street
New York City

Paracoil

General Office and Factory:
Elizabeth, N. J.

PRODUCTS—Davis Automatically Fired Boilers, Paracoil Storage Water Heaters, Instantaneous Water Heaters, Fuel Oil Heaters, Oil and Water Coolers, Distillers, Feed Water Heaters, Feed Water Filters and Grease Extractors, Steam Traps, Evaporators, Converters, Preheaters, Hot Water Generators, Condensers, Heat Exchangers

Years of trouble-free service without excessive maintenance charges have been the aim in the design of Paracoil apparatus. Paracoil Heat Exchange Products are preferred by many leading industrial, naval and building engineers. Many large ships and famous buildings have Paracoil heat transfer equipment. Special heat transfer problems invited. Write for bulletin on products in which you are interested.



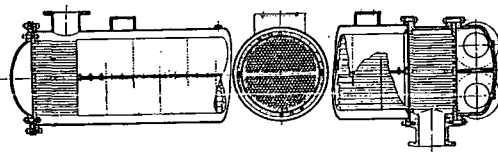
PARACOIL STORAGE WATER HEATER

For heating and storing hot water for apartment houses, office buildings, hospitals, laundries, etc. Operates with boiler water, exhaust or live steam as heating medium. Steel plate storage water shell with removable seamless drawn copper heating coils. Maximum efficiency of heat transfer. Any pressures specified. Capacity 40 to 25,000 gal. per hour. Capacity tables, calculating data, etc. on request. Write for catalog.



TUBULAR HEAT EXCHANGERS

For cooling lubricating oil or water and similar heat exchange uses. Removable or non-removable tube bundle as required. Transverse or Longitudinal Baffling as required. Floating tube sheet and head construction or shell expansion joint compensates for expansion and contraction strains. Built in single and multi pass types. Small space requirements. Write for information.



Section of Paracoil Tubular Heat Exchanger



PARACOIL INSTANTANEOUS U-TUBE TYPE WATER HEATER

Ideal for apartment houses, hotels, etc. Connected below water line of boiler, heats water supply when fires are banked. Cast-iron or steel shell, heating elements of seamless drawn copper.

Capacities 300 gal. and up (below water line installation) or 20 gal. and up (live steam installation).

Catalog on request.

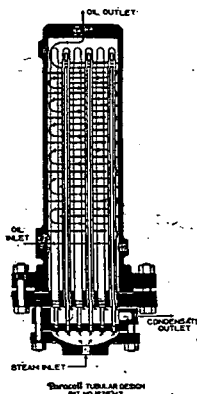
PARACOIL FUEL OIL HEATER, TUBULAR AND COIL TYPES

Rand System of Fuel Oil Preheating in Storage Tanks

Efficient and compact, removable heating element, freedom from expansion and contraction strains. Designs permitting easy cleaning.

Turbulent flow of oil assures high heat transfer. Unusually low pressure drop or friction loss. Variable baffle spacing gives variable velocity in proportion to change in viscosity. Horizontal or vertical installation.

Write for catalog.



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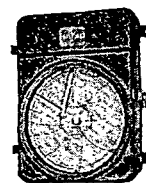
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BRISTOL'S INSTRUMENT CO., LTD., Pomeroy Street, New Cross, LONDON, S. E. 14

BRISTOL'S

REG. U. S. PAT. OFFICE

Recording Pressure Gages



Recording Pressure Gage Model 40M

For securing continuous day and night records of pressure or vacuum for steam, air, gas and liquids. Furnished with charts reading in pounds, ounces, feet, inches, metric or any other desired units for ranges from full vacuum to 12,000 lb per sq. in.

Recording Indicating Thermometers

For all commercial ranges from 60 F below to 1000 F above zero.

Electric Indicating Thermometer indicates temperature at any number of distant points.

Metameter for Telemetering

Consisting of (1) a recording receiver in your office or other central place, (2) a transmitter at the far point where the measurement is made, and (3) a simple circuit of two small gage wires (or any existing telephone line) for carrying the electric impulses automatically sent out by the transmitter, the Metameter tells accurately and instantly how pressure, temperature, or flow is fluctuating at any point a hundred feet or several thousand miles away.

No interference with or from power, light or telephone lines. Easy to read 12 in. diameter chart, with wide open scale. D.C. or A.C.; 25 or 60 cycles; self-checking.



Metameter Transmitter, Model M35T

Automatic Temperature Control

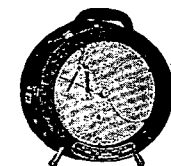
A complete line of Automatic Control Instruments is available for temperatures up to 3000 F. These are suitable for application to steam, oil, gas and electric heated equipment. Both electric and air operated models are furnished.



Free Vane Temperature Recorder Controller, Model 5240M

Direct Reading Relative Humidity Recorder

Chart record shows at a glance the trend of humidity condition. Novel vapor-sensitive hygroscopic element. Special aging process assures sustained accuracy. Eliminates calculation, and use of humidity tables. No water required. No fan used. Accurate below freezing temperature. Light portable corrosion-resisting case.



Thermo-Humidigraph, Model 4069

Flow Meter for Steam, Liquids, Gas

Employs orifice and mercury manometer system of measurement. Meter body is of forged steel, with stainless steel parts inside the mercury chambers. All connections are welded or mechanical. The stuffing box is of hardened stainless steel. Grease-packed and leak-tight.

For working pressures up to 1000 lb. Special bodies available up to 3000 lb.

Industrial Stem Thermometers

With plain or red background. Fixed thread, union connection or socket design. Straight form, or rear oblique, right side and left side angle forms.

Wet- and Dry-Bulb Psychrometer

Accurate and dependable instrument by which relative humidity, or atmospheric moisture may be determined. Available in self-contained and distance types.

Brown Instrument Division

of Minneapolis-Honeywell Regulator Co.

Main Office and Plant

Philadelphia, Pa.

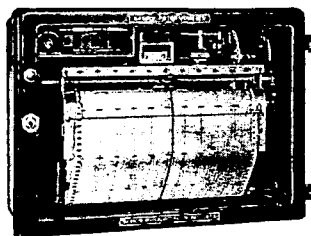
Equipment available through all Minneapolis-Honeywell Branch Offices as follows:

ATLANTA	DETROIT	LOUISVILLE	PITTSBURGH
BALTIMORE	DULUTH	MILWAUKEE	PROVIDENCE
BOSTON	EAST ORANGE	MINNEAPOLIS-ST. PAUL	ST. LOUIS
BUFFALO	HARTFORD	NEW YORK	SAN FRANCISCO
CHICAGO	HAVERHILL, MASS.	OMAHA	SPRINGFIELD, MASS.
CINCINNATI	INDIANAPOLIS	PEORIA	SYRACUSE
CLEVELAND	KANSAS CITY	PHILADELPHIA	

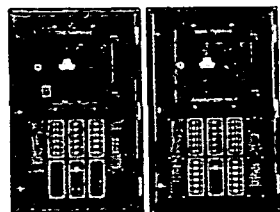
In Canada: Main Office and Factory: TORONTO, CANADA; Branches: CALGARY and MONTREAL
Distributing Offices in all other principal cities

Brown Instruments

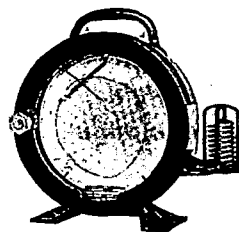
The extent to which air conditioning equipment is being used in office buildings, theatres, stores, industrial buildings, etc., has opened up a wide demand for indicating and recording resistance thermometers because the temperatures throughout these air conditioning systems should be checked periodically to get the best results at minimum operating cost.



Brown Recording Resistance Thermometer



Brown Balanced Bridge Type Resistance Thermometer and Multiple Key Switch Panel



Brown Portable Thermometer

In addition to Resistance Thermometers, the Brown Instrument Company manufactures:

Thermometers
Hygrometers
Pressure and Vacuum Gauges
Potentiometer Pyrometers

Flow Meters
CO₂ Meters
Tachometers
Liquid Level Gauges

See also Pages 1044-1049

Consolidated Ashcroft Hancock Co., Inc.

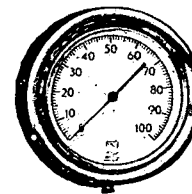
Bridgeport, Conn.

BRANCHES IN PRINCIPAL CITIES

Makers of AMERICAN INDUSTRIAL INSTRUMENTS—Since 1851

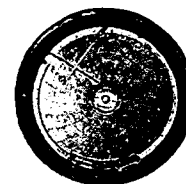
Manufacturers of Indicating and Recording Gauges; Gauge Testers; "U" Gauges; Draft Gauges; Indicating and Recording Thermometers; Tachometers; Dial Thermometers; Pressure and Temperature Controllers; Electric Temperature Controllers; Pop Safety and Water Relief Valves; Steam Traps; Engine Indicators; Counters; Absolute Pressure Gauges. Also manufacturers of Bronze, Cast Steel and Forged Steel Valves, Locomotive and Engine Room Clocks; Barometers; Mercury Column Gauges; Steam Whistles; Hydraulographs; Gauge Boards.

Ashcroft American Gauges—Ashcroft American Gauges are made in all sizes from 2½ to 12 in., for pressures from 8 oz to 25,000 lb and also for vacuum. Cases are cast-iron or cast brass. The movements are Heavy Duty and all bearings are Monel Metal. Write for Catalog No. A-59.



For Mercury Pressure and Vacuum Gauges, "U" Gauges, Draft Gauges and Mercurial Barometers, write for Catalog B-59.

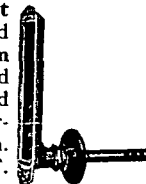
American Recording Gauges—American Recording Gauges are made for all pressures from 15 in. of water to 10,000 lb. and for vacuum. They are made in one size only to accommodate a 10 in. chart, having an effective scale width of 3½ in. The case is Die Cast with a dull black hard-rubber finish and with either bottom or back connection. The pen-arm is made of non-corrosive Monel Metal and is of the inverted type. Operating instructions are lithographed on the chart plate so that they cannot be lost.



Especially designed Seth Thomas clocks are used, and all customary time periods can be furnished.

American Recording Gauges are equipped with the Time Punch which virtually makes each instrument a time clock, since a hole is punched in the chart whenever a reading is taken. Write for Catalog E-59.

American Air Duct Thermometer—Designed especially for both warm and cold air ducts. Fitted with chromium plated frame, glass front. Furnished with 9-in. or 12-in. scale graduated 0-160 F. Write for Catalog F-59.



American Recording Thermometers—Made for recording temperatures from minus 40 to plus 1000 F. or equivalent C. Very flexible connecting tubing up to 200 ft. One size only to accommodate 10 in. chart, with an effective scale width of 3½ in.



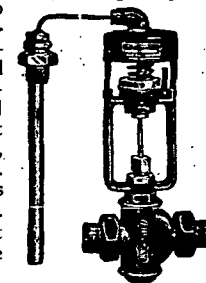
Same case as for the American Recording Gauge, so that all instruments are uniform in appearance when mounted on Gauge Boards. American Indicating Gauges and Dial Thermometers are also furnished in same case. Write for Catalog H-59.

American Dial Thermometers—American Dial (mercury-filled) Indicating Thermometer has the accuracy of the standard glass tube thermometer and the reading convenience of a dial face. Entire working mechanism is made of steel, meaning long life.



Standard size of dial 6 and 12 in. Furnished with rigid connection or flexible capillary steel tubing up to 200 ft long. For temperature ranges from minus 40 to plus 1000 F. Write for Catalog G-59.

American Precision Temperature Controllers—Self-operated and simple in construction. For regulating temperatures from 25 to 385 F. Under favorable conditions temperature will be held within 1 deg. Sensitive, rugged and accurate. For hot water service tanks, water heaters, etc. Standard ranges carried in stock. Size of valve must be specified. Write for Catalog R-59.



Hays Corporation

COMBUSTION INSTRUMENT SPECIALISTS SINCE 1901
HAYS DRAFT INSTRUMENTS

Michigan City

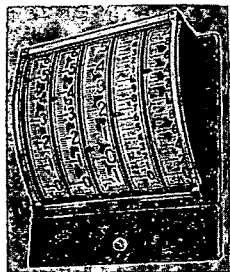
Indiana

PRODUCTS—Dry Type Pointer Draft Gages; U-Tube, Direct Reading and Vernier Scale Draft Gages; Air Filter Gages; CO₂ and Draft Recorders, Flue Gas Analyzers (Orsata); Portable Combustion Test Sets; Draft Recorders; Boiler Panels; Ignition Velocity Meters; Portable Hand Operated Gas Calorimeters, Recording Gas Calorimeters.

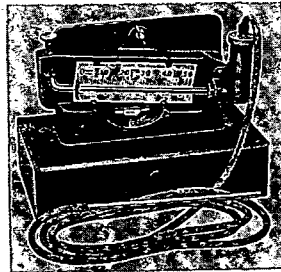
INDICATING

Hays Dry Type Pointer Gages for Draft, Pressure or Differential have Lug, wall or panel mounting. Any number of gages may be arranged compactly side by side.

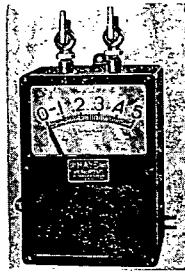
Critical accuracy is obtained from this fundamental type of portable inclined tube gage. Convenient reel for tubing. The B Dry Type Portable is accurate and rugged. Set up for service in 30 seconds. No liquids used.



V Type



Portable Inclined Tube

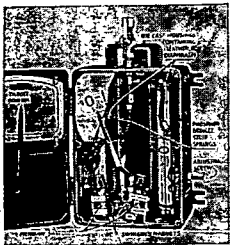


B Gage for Air Filters

Especially designed for indicating resistance through air filters. Air tight case will indicate differential pressure when connected to opposite sides of filter bank.

RECORDING

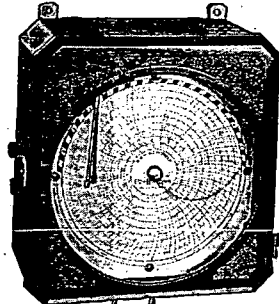
The Hays "OT" Recorder is an outstanding instrument that will put the operation of any system of ventilating or air conditioning on a recorded engineering basis. It is available also in a two-pen, two-unit style in ranges from 0 to 0.5 up to 0 to 100 inches water and secondary pressures from +0.1 to 0 to -0.4 up to 5.0 to 0 to 5.0. The Hays Super-sensitive "OH" Recorder (not illustrated) will record draft and pressure from 0 to 0.1 to 0 to 0.4 and pressure-draft from +0.1 to 0 to -0.1 to 0 to +0.05 to 0 to -0.15. Write us for special information on these two outstanding air movement instruments.



Draftrol

CONTROLLING

Any range, any sensitivity of slack leather diaphragm may be built into the Hays Draftrol. Changes in draft, open or close electrical contacts in the non-tilting sealed mercury switches. We invite correspondence on applications of the Hays Draftrol Gages.



Hays "OT" Recorder

STATIC PRESSURE AND AIR FLOW

Illinois Testing Laboratories, Inc.

422 N. LaSalle Street, Chicago, Illinois

TESTING ENGINEERS AND MANUFACTURERS

"Alnor" Indicating Pyrometers—Portable and Stationary, "Alnor" Surface Temperature Pyrometers, "Alnor" Distant Reading Resistance Thermometers, "Alnor" Velometers (Boyle Type)—Air Velocity Meters.

"ALNOR" (BOYLE TYPE) VELOMETER

The Instantaneous Direct Reading Air Velocity Meter

Heating, ventilating, and air conditioning engineers as well as equipment manufacturers, industrial engineers, or anyone who has need for an instrument to give quick, accurate air velocity information will find the Velometer an indispensable instrument.

It is a direct reading instrument which gives accurate and instantaneous readings of the speed and direction of air motion measured in feet per minute.

The instrument is housed in an attractive black bakelite case 5 1/4 in. x 5 3/4 in. x 2 3/8 in. in size and weighs approximately 2 lb.

The air which enters through the meter, actuates a movement comprising a vane, control hair springs, pointer, and magnetic damping system.

No mathematical calculations or stop watches are necessary to determine the velocity reading as the pointer indications are a direct measurement of air speed.

The Velometer is furnished in two types: (1) the Shutter Type; (2) Tube Type. Standard double ranges 0-300 low range 0-3000 high range. Other special ranges higher and lower can be furnished.

Shutter Type—The shutter type has a shutter which opens or closes the ports in the instrument case for changing the scale range.

To use the shutter type Velometer it is merely necessary to hold the Velometer with the ports in line with the air stream.

Tube Type Velometer—The tube type Velometer is the same as the shutter type except that attachments are used with a suitable jet for the high scale readings. The air then enters or is drawn through this jet. With the tube type Velometer accurate velocity readings of the discharge or suction grilles, ducts or other restricted or small openings can be easily and quickly obtained.

Various types of jets are obtainable each especially designed for special function.

No. 2 Averaging Jet—In commercial work for ordinary duct openings or grille

use, it is simpler to take average readings over a 4 in. circle and for this service the No. 2 averaging jet is offered.

No. 5 and No. 6 Jets—Either bent or straight jets of this type having small orifices give exact velocities over a 1/2 in. diameter surrounding the end of the jet. These jets are ideal for exploring the velocities through irregular shaped grilles, leaks in duct systems, and for measuring velocities of slotted grilles where other type of velocity meters cannot be used.

Duct Jets are offered for readings within ducts. The duct jet is in effect a double jet using two tube attachments and two meter fittings. The air enters the meter through one side of the jet and is returned and discharged back into the duct through the other side of the jet.

Write for Literature

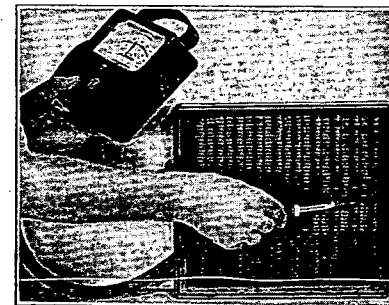


Fig. 1—Showing Tube Type Velometer with No. 5 Angle Jet taking air velocity of grille

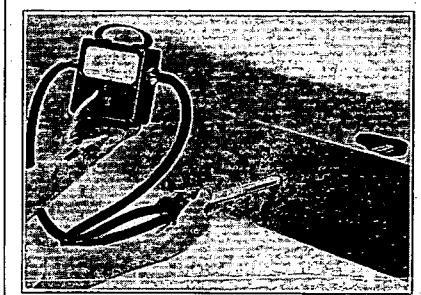


Fig. 2—Showing Tube Type Velometer with Duct Jet taking readings directly in the duct. Duct Jet furnished 18 in. or 36 in. long

Leeds & Northrup Company

General Office and Works: 4941 Stenton Avenue, Philadelphia, Pa.

Branch Offices:

CHICAGO

ST. LOUIS



CLEVELAND

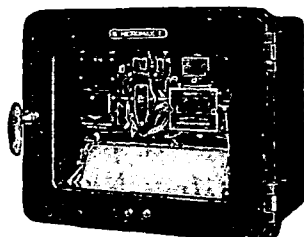
DETROIT

LOS ANGELES
SAN FRANCISCO

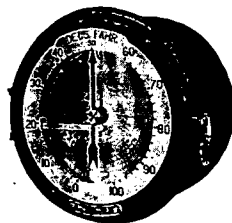
PITTSBURGH
HOUSTON

TULSA

RUGGED, NULL-TYPE INSTRUMENTS THAT ARE RELIABLE



Model S Micromax Recorder
Records from 1 to 16 points on a single strip-chart. Extremely open record. Can operate signals. (About 1/16 size)



Model R Micromax Recorder
Records 1 or 2 points on a round-chart. Has extremely readable dial. Can operate signals. (About 1/18th size)



Switchboard Indicator
Hand-operated. Can be connected through selector switches to any number of points. (About 1/18th 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 detector locations. The detectors (resistance thermometer bulbs called Thermohms), 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; Micromax Model R, for related pairs such as wet and dry bulb; indicators with switches for any number of Thermohms; or indicating and recording combinations.

This equipment is fundamentally reliable. Instruments and Thermohms are highly responsive, yet rugged in construction. A complete system is easy and economical to install, regardless of distances. It is easy to operate and demands minimum maintenance. Thermohms and instruments are interchangeable, and 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 . . . to get maximum return on the entire conditioning investment.

T-362

Electrical Instruments for the Heating Plant

The facts needed to operate a modern heating plant so as to save fuel, to protect equipment, and to operate efficiently at varying loads are provided reliably by rugged L & N instruments. Readings can be indicated or recorded or both. Recorders can be equipped to operate signals or alarms that warn the operator of extreme conditions. In some cases the instruments control automatically.

Micromax Model S provides a permanent record of conditions at from 1 to 16 points on one wide-scale chart. Micromax Model R concentrates on conditions at one point, provides a permanent record, and has a giant indicating dial that can be read at a glance. The switchboard indicator provides intermittent checks on conditions at one or several points.

In the heating plant, L & N measuring, signalling or controlling equipment is used for:

Metermax Combustion Control.
Furnace Pressure Control.
Smoke Density Analysis.
Flue Gas Analysis (Per cent CO₂).
Flue Gas Temperatures.
Steam and Water Temperatures.
Boiler Furnace Temperatures.
pH—Alkalinity (or Acidity) of Water.
Electrolytic Conductivity of Water.
Percent Leakage of Cooling Water.

The Palmer Company

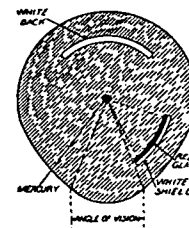
426 Clay Street

Cincinnati (St. Bernard), Ohio

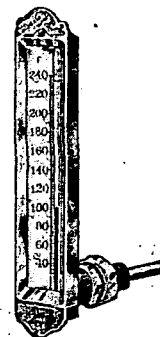
Manufacturers and Originators—"Red-Reading-Mercury" Thermometers

PRODUCTS—Indicating Thermometers with the RED column.

"RED-READING-MERCURY" was originated by Palmer. We hold the basic patents on the reflected RED color. See illustration. A strip of red glass is drawn into the tube and when the mercury column rises, in the tube, the RED color is reflected onto it. Nothing is done to harm the purity of the mercury.



INDUSTRIAL STYLE THERMOMETERS



No. 387

Furnished in all sizes and with STRAIGHT or various ANGLES styles. With Fixed taper thread, Union connection, Separable Socket or flange fittings.

With the RED column, these thermometers are very easy to read.

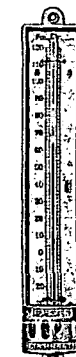
ACCURACY GUARANTEED; MADE STRONG AND DURABLE.

OUTDOOR THERMOMETERS

Here is an ideal thermometer for use outdoors or inside. The metal case is of bronze which will not corrode. Graduated scale, giving very accurate temperature. White Duco finish. Nickel-plate or Chromium finish also furnished. With the RED column, it can be seen a great distance.

Other styles of Wall Thermometers as well as Laboratory styles furnished with the RED column.

Write for FREE catalog No. 200-C.



No. 2180
Thermometer

Instruments for Testing AIR CONDITIONING EQUIPMENT

When air-conditioning equipment is installed, the contractor must have reliable and accurate instruments to make tests. Only with such instruments can he give satisfactory results.

SLING PSYCHROMETER

This pocket style instrument can be carried anywhere and quick tests made. Wet and dry-bulb tubes guaranteed accurate. Furnished with metal case, leather covered.



No. 14200

ASPIRATING PSYCHROMETER

(This is not illustrated). The Aspirating Psychrometer for testing humidity is recommended where readings of greatest precision are required. Contains small motor and blower so the air is blown over the wet and dry bulb thermometers. Two styles:

Battery operated for use where there is no electric connection.

Electrically operated, to plug in on light circuit.

With carrying case; neat, compact, light weight. Very handy instrument.

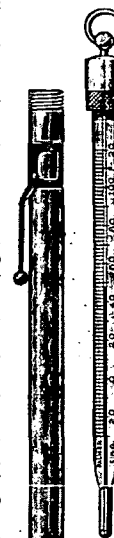
POCKET THERMOMETERS

It is helpful to carry a pocket test thermometer, so that it is available at any time.

This is a reliable, guaranteed accurate thermometer. Easy to read with the RED column.

Repairs—We can repair all makes of mercury thermometers and furnish "Red-Reading-Mercury" at no extra cost. The type of repair work we do will add years to the life of the thermometer.

A trial Order will convince you.



No. 13240

Taylor Instrument Companies

Rochester, N. Y., U. S. A.

NEW YORK
CHICAGO
BOSTON

IN CANADA—TAYLOR INSTRUMENT COMPANIES OF CANADA, LTD., TORONTO

PHILADELPHIA
PITTSBURGH
CLEVELAND

LOS ANGELES
INDIANAPOLIS
SAN FRANCISCO

ST. LOUIS
CINCINNATI
TULSA

DETROIT
ATLANTA
MINNEAPOLIS

Manufacturing Distributors in Great Britain, Short & Mason, Ltd., London

Manufacturers of Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure and Humidity

Taylor Recording Thermometers—Temperature ranges and time requirements vary greatly in heating and ventilating work. Taylor Recorders are made in scale ranges and time periods to meet these needs.

In their handsome new cases, these instruments

are beautiful and efficient, particularly adapted for heating and air conditioning applications. They may be had for surface or flush mounting. When set in panel boards, the polished flanges make an effective installation.

Write for special information suitable to your needs.

Taylor Electric Contact Temperature Control—These instruments combine in the same case an electrically-operated temperature controller with an indicating thermometer. One tube system operates both units.

The New Taylor "Fulscope" Recording Controller—An air-operated controller so versatile that practically any character of process control can be obtained, regardless of time lag in apparatus, by a simple screw driver adjustment on a graduated dial—without

Taylor

requiring a skilled operator or interruption of service.

Vast improvements in entire mechanism.

Easily changed from direct to reverse-acting, or vice-versa—no extra parts.

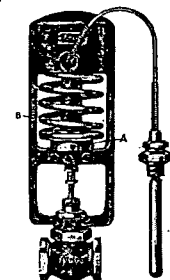
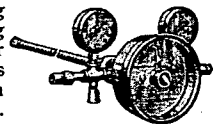
Compensates for fluctuations in air-pressure supply.

Die-cast case; dust-, moisture-, and fume-proof.

Available forms: for controlling temperature, pressure, temperature and pressure, rate of flow, liquid level.

Where extreme load changes or badly balanced operating conditions exist, the Taylor "Dual-Response Control Unit" is the only positive means of maintaining control-point. Write for literature.

Taylor Type-P Controller—A compact and very sensitive controller, ideal for air-ducts, air-washing machines, cooling rooms and similar applications. Uses compressed air as an actuating medium.

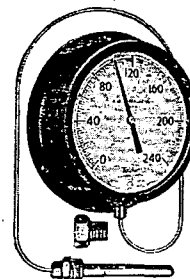


Taylor Self-Acting Temperature Controller—Adapted for use on hot-water storage tanks, etc. It is "self-acting" in that it requires no auxiliary motive power, such as compressed air, to open and close the steam valve. As heat is applied to the bulb, the volatile liquid inside sets up a vapor pressure proportional to the temperature. This pressure is transmitted to a "stack" of metal diaphragms attached to the upper end of the valve stem, thus moving the valve disc toward the valve seat.

The valve can be closed at any desired

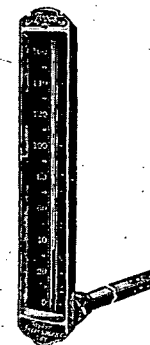
temperature, or a throttling action can be obtained. Not practicable on pipe lines having steam pressure over 125 lbs. Should be installed in a vertical position on top of a horizontal line.

Operating ranges 110 deg. to 170 F., 130 deg. to 190 F., 170 deg. to 240 F. or 220 to 280 F. as specified.



as a clock or steam gage.

Taylor Thermometers for Air Ducts, Etc.—The Taylor line of industrial thermometers presents many styles and scale ranges with bulbs for every application. Suitable for air ducts, kiln temperatures and oven temperatures. For detailed information, write direct, mentioning your requirements. All these Taylor Industrial Thermometers have lens-front glass tubes, giving great magnification to the liquid, making it easy to see and read.



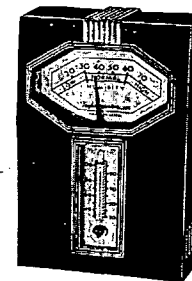
Taylor Sling Psychrometer—The advantage of this form of Wet-and-Dry-Bulb Hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as in whirling the bulbs they are subjected to perfect circulation. Consists of two accurate etched stem thermometers mounted on a die-cast frame, with the bulb of one covered with a wick to be moistened.

These thermometers have scales of 0 to 100° F, graduated in 1/2 degree divisions. A copper case protects the tubes

when not in use.

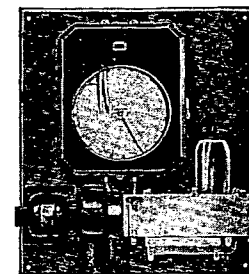
Taylor Hampton Model Humidiguide (Direct-Reading)—A hygrometer giving direct humidity percentages, in a smart modern case suitable for home, office or public buildings. Finish is satin black with chrome trim.

The Permacolor Thermometer is filled with non-fading, easy-reading red liquid.



Taylor Recording Hygrometer—This instrument records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.

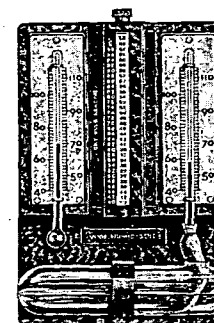
Type shown above with motor-driven fan for conditioned rooms or passages in which circulation is poor. Can be supplied without fan for installations where circulation across bulb is good.



Taylor Anemometer—This instrument is ideal for measuring air velocities with the fan revolutions indicated on the dial. Available in various models for a wide range of air speeds and registration limits.



Taylor Humidiguide—A handsome small hygrometer for the wall of the home, office, school or other building where a neat, easy-reading and inexpensive instrument is desired. It is self-contained, requiring no charts or separate tables. Frame is Mahogany Bakelite.



United States Gauge Co.

44 Beaver Street,



New York, N. Y.

Factory at Sellersville, Pa.

Makers of Quality

INDICATING AND RECORDING PRESSURE GAUGES

All Sizes and Types for Every Purpose

U. S. GAUGES—U. S. Gauges are made in all sizes from 2 to 12 in. inclusive for pressures from 1 lb. up to 50,000 lb., and for vacuum. Cases may be cast-iron, cast brass, drawn steel and drawn brass for wall mounting or flush mounting. For severe service long wearing hardened steel or bushed movements may be supplied.

For service on Steam Heating Systems—

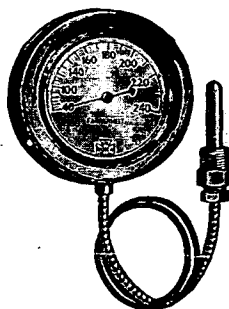
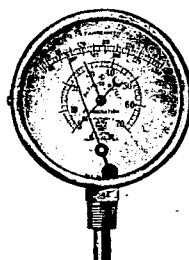
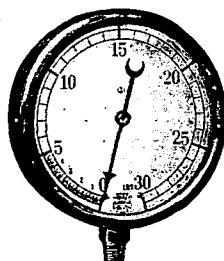
- Steam Gauges.
- Compound Pressure and Vacuum Gauges.
- Retard Gauges.
- Compound Retard Gauges.
- Steam Gauges with Internal Siphons.

For Hot Water Heating Systems—

- Altitude Gauges.
- Tank-in-Basement Gauges.
- Altitude and Pressure Gauges.
- Combination Altitude Gauges, and
 - (a) Bimetal Thermometers.
 - (b) Glass Tube Thermometers.
 - (c) Vapor Tension Distance Type Thermometers.
- Glass Tube Hot Water Thermometers.

U. S. RECORDING GAUGES—U. S. Recording Gauges are made in 8½, 10 and 12 in. sizes for pressures from 1 lb. up to 50,000 lb., and for vacuum. Cases may be cast-iron or cast brass for wall mounting or flush mounting. Pen arms are made of non-corrosive metal. Especially designed clock movements are used. Charts can be furnished for customary time periods.

U. S. DIAL THERMOMETERS—U. S. Dial Thermometers are of the vapor tension type with open scale reading in the central and upper portion of the scale. Cases may be cast-iron, cast brass, drawn steel or drawn brass for wall mounting or flush mounting. Supplied in all sizes from 2 to 12 in. inclusive, for temperature ranges from 40 F. to 800 F. Furnished with rigid connection bulb or with flexible capillary tubing up to 100 ft. long.



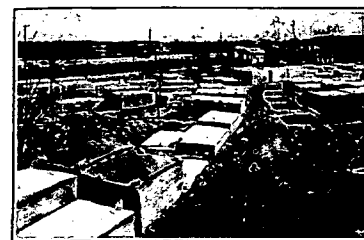
The Agasote Millboard Company

Trenton, New Jersey

Manufacturers of Quality Products Since 1909

HOMASOTE

THE WEATHERPROOF INSULATING AND BUILDING BOARD



Homasote at Home—Protection from the Elements Unnecessary

HOMASOTE WEATHERPROOF INSULATING AND BUILDING BOARD

Homasote can be used as a combination insulating and structural material. Important facts in the selection of an insulating material are its efficiency as an insulator, its structural strength, its moisture resistance, its plaster adhesion, its resistance to air infiltration, whether it is fire-retardant and the cost of installation and upkeep.

Homasote is manufactured in BIG sheets, 8 ft x 14 ft, 8 ft x 12 ft, 6 ft x 12 ft, and even multiples thereof, making it possible to cover the average size wall in one piece.

It may be used on the exterior of a building even without painting.

Comparative tests on air infiltration are shown below. These tests were made by the Lewis Institute, Chicago, Ill. The

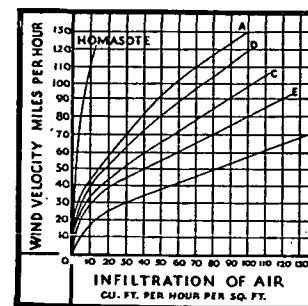
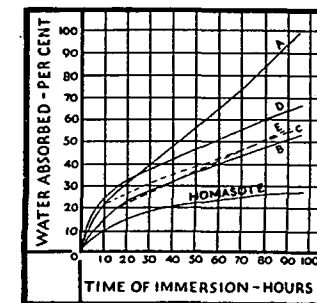


chart shows that at a wind velocity up to 40 miles per hour, 18 to 112 times more air passed through the other boards tested than through Homasote and at higher velocities, the infiltration through these other boards increased in a much greater proportion.

Homasote has very high moisture resistance, according to tests made by the Robert W. Hunt Co., Chicago. Reference to the chart shows that the other boards took up 101 per cent to 274 per cent more moisture.



HOMASOTE TYPE MN FIRE-RESISTING INSULATING BOARD

Homasote Type MN provides not only insulating value but also protection from fire. It is made in ½ in. and ¾ in. thickness, and in sizes of 4 ft x 7 ft, 4 ft x 8 ft, 4 ft x 10 ft, 4 ft x 12 ft, and 4 ft x 14 ft. In the ¾ in. thickness, Homasote Type MN takes a one hour fire rating.

The thermal conductivity of Homasote Type MN is 0.43 Btu per hour, per square foot per 1 deg Fahrenheit, per inch thickness.

It is ideal for air conditioning ducts or for wall, floor or roof areas in fire zones.

The insulating value of Homasote Type MN protects the studding behind it in a hot fire, and Homasote Type MN will not shatter if water is applied to it when hot.

Write Direct to THE AGASOTE MILLBOARD COMPANY, Trenton, N. J., for samples and descriptive literature.

Armstrong Cork Products Company

Building Materials Division

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Capital City Supply Co.....CHARLESTON, W. VA.
Stearns-Roger Mfg. Co.....DENVER, COLO.
Fischer Cement & Roofing Co.....LITTLE ROCK, ARK.
Gay Engineering Corporation.....LOS ANGELES, CALIF.
Fischer Lime and Cement Co., Inc.....MEMPHIS, TENN.
H. T. Steffee.....NEW ORLEANS, LA.
John R. Livezey.....PHILADELPHIA, PA.

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Asbestos Supply Company.....SEATTLE, WASH.
Asbestos Supply Company.....SPOKANE, WASH.
Asbestos Supply Company.....TACOMA, WASH.
John R. Livezey.....WASHINGTON, D. C.

For detailed technical information, samples, and descriptive literature, ask any office or representative

PRODUCTS—Armstrong's Corkboard, Armstrong's Cork Covering, Armstrong's Vibracork, Armstrong's Corkoustic, Armstrong's Temlok, Armstrong's Temcoustic, Armstrong's Insulation Sundries.

Corkboard Sizes

Armstrong's Corkboard is furnished in rigid boards 12 in. by 36 in., 12 in. x 32 in., 18 in. x 36 in., 24 in. x 36 in., and 36 in. x 36 in., in several thicknesses: 1 in., 1½ in., 2 in., 3 in., 4 in., and 6 in.

Insulating Efficiency

The thermal conductivity of Armstrong's Corkboard, depending on the density, is 0.27 to 0.29 B.t.u. per hour per inch thickness at 90 deg F mean temperature (U. S. Bureau of Standards).

The value of adequate and efficient insulation is covered in the text section of this book (Chapter 5) and the tables on pages 112 to 124 indicate the savings which can be effected by using 1½ in. or 2 in. of corkboard in standard wall and roof construction. The reduction in heat loss amounts to from 50 per cent to 75 per cent. This means that an adequate thickness of corkboard on walls and roofs reduces the heat wasted, and, therefore, the heat requirements of the house by 25 per cent to 40 per cent.

Air Conditioning

An adequate thickness of corkboard insulation is essential for any air conditioned room or structure. Insulation reduces the amount of heating or refrigerating equipment required to produce desired temperatures and is highly important to the satisfactory and economical control of humidity conditions.

The insulation of air conditioning equipment assures economical and efficient operation by minimizing refrigeration losses. De-humidifying chambers and ducts as well as pumps and brine storage tanks may be insulated with Armstrong's Corkboard. The economies resulting from the application of Armstrong's Cork Covering to all cold lines repay the cost of the insulation and assure additional refrigeration savings.

The elimination of noise and vibration transmission is of primary importance in air conditioning work. Armstrong's Vibracork, made in three densities, is ideal for this purpose. It does not take a set, is not affected by atmospheric moisture, and will not deteriorate in service.

Products—Information

Additional Armstrong insulating and acoustical products especially suited to heating, ventilating, and air conditioning work include:

Armstrong's Corkoustic.
Armstrong's Temlok.
Armstrong's Temcoustic.
Armstrong's Insulation Sundries.

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 branch office or representative or the Armstrong Cork Products Company, Building Materials Division, Lancaster, Pa.

The Eagle-Picher Lead Company

General Offices: Temple Bar Building, Cincinnati, Ohio

Offices in All Large Cities



EAGLE INDUSTRIAL INSULATION

The Eagle-Picher Lead Company manufactures a complete line of industrial insulation materials, effective for a complete range of temperatures. Representative products are:

Eagle Super "66" Plastic Insulation

For application on practically all forms of heat producing and heat transferring equipment. "Springy ball" construction provides remarkable heat-saving efficiency up to 1800 F. Easily applied with a trowel. May be applied on any clean surfaces. Great coverage—60 sq. ft. 1 in. thick per 100 lb. 100 per cent reclaimable.

Eagle Blanket Insulation

Eagle Insulating Wool felted and secured between metal fabrics. For large surfaces where temperatures reach as high as 1200 F. Available in flexible or rigid form. Easy to cut and fit. Sizes—2 ft x 4 ft and 2 ft x 8 ft. Thicknesses range between 1 in. and 8 in.



Eagle "66" quickly applied with a trowel on hot and cold surfaces

Eagle Asbestos Cements

Inexpensive asbestos fibre cements of good mixing, trowelling and adhesive properties. For temperatures up to 1000 F.

Eagle Insulseal (Waterproofing Cement)

Durable coating designed to protect all kinds of insulation from moisture, water, air infiltration, fumes; from vibration, abrasion. Ready mixed, easily and quickly applied.

Other Products

Eagle "99" Finishing Cement, Hair Felt, Pipe Covering and Blocks (all types), Insulating Wool, Boiler Setting Cement.

Data and Specifications

For complete specifications and technical data on Eagle Industrial Insulation, see Sweet's Engineering or Power Plant Catalog.

EAGLE HOME INSULATION

The Eagle-Picher Lead Company manufactures two types of "mineral wool" insulation for homes: (1) in granulated form for pneumatic application in new and existing construction. (2) in bat form for new construction.

Both types are wall-thick, non-structural, extremely light-weight, non-corrosive, fire-proof—with thermal conductance (in applied thickness of 3½ in.) of only 0.074 Btu. (The over-all conductance would be considerably lower).

Eagle Home Insulation keeps homes cooler in summer (12 to 15 deg cooler than outdoor temperatures), and warmer in winter (fuel saving ranges between 10 and 40 per cent).

Eagle Insulating Wool easily blown into spaces between wall studs

For Pneumatic Application

In granulated form, Eagle Insulation is blown into hollow spaces between wall studs and between joists in the attic floor by a special pneumatic process. No building alterations are necessary, whether the house is of frame, brick or stucco construction. No muzzing up inside. Work is done by skilled contractors, licensed by Eagle-Picher.

Bat Form



Eagle wall-thick bats quickly installed in new construction

Eagle Insulating Bats are rectangular pads 15 in. x 18 or 23 in. x 3½ in., designed to fit snugly between studding and joists. Irregular spaces around doors and windows filled by cutting bats to exact size.

Data and Specifications

For complete specifications and technical data on both types of Eagle Home Insulation, see Sweet's Architectural Catalog.

Alfol Insulation Company

Incorporated

Chrysler Building :-: New York, N. Y.

Agents in Principal Cities

INSULATION for

Fans
Blowers
Pumps
Ducts
Houses, Buildings, etc.

Turbines
Dehumidifiers
Air Conditioners
Boilers, Pipes, etc.



ALFOL
Patented in 34 Countries

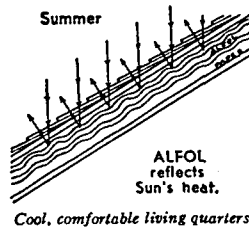
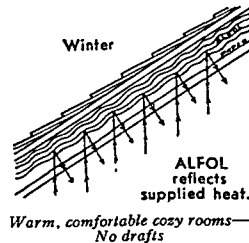
INSULATION for

Refrigerators
Refrigerator Cars
Refrigerator Trucks
Refrigerator Boxes
Refrigerated Rooms
etc.

Ships
Ovens
Ranges
Tanks
Stills

ALFOL consists of single or multiple sheets of polished pure aluminum foil installed between wall studs, furring strips, ceiling joists or roof rafters. When heat tends to flow through such insulated structures, 95 per cent of radiant heat is reflected back toward its source. In summer, stifling outdoor heat is driven off. In winter, practically all supplied heat is retained in the house where it is needed.

The reflection of heat is a revolutionary principle in insulation engineering which has won widespread recognition and acceptance by engineers and scientists throughout the world. ALFOL is now

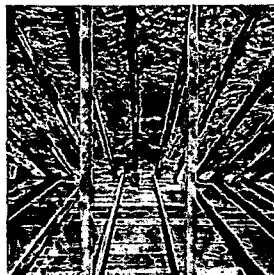


extensively used by many of the largest manufacturing and industrial plants, U. S. Navy, and others, as well as in houses and buildings of every kind and size.

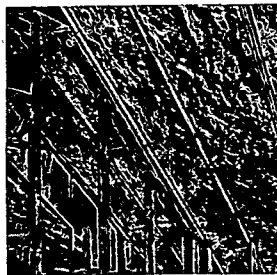
ALFOL will save up to 75 per cent or 80 per cent of the heat loss that would occur through an uninsulated house.

It consequently costs less to heat an ALFOL insulated home. Many owners report over one-third saving in fuel costs.

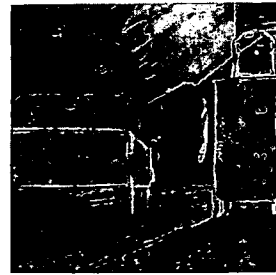
ALFOL is installed only by Authorized Applicators. All work is done in strict accordance with ALFOL Standard Specifications.



Alfol Roof Insulation Keeps Houses Cooler in Summer and Warmer in Winter



Alfol Applied Between Roof Rafters Reduces Heat Loss Over 80 per cent



Extra Living Quarters, Just as Tenable as Any Other Part of the House, May be Obtained by Insulating Attic Roofs with Alfol

Aluminum Foil Does Not Tarnish

Alfol Insulation Company

Incorporated

Data on Heat Savings Effected by Alfol

Construction	Heat Transmission, B.T.U./Hr./Sq. Ft./°F.			Heat Transfer Stopped—%	
	Not Insulated	1 layer ALFOL	2 layer ALFOL	1 layer ALFOL	2 layer ALFOL
Wood Stud Wall—4-in. studs	0.26	0.12	0.09	53%	65%
12 in. Brick Wall—Furred	.25	.11	.09	56%	64%
Open Attic Floor	.69	.17	.12	75%	83%
Wood Shingle Roof	.47	.15	.11	68%	77%
Concrete Roof—8-in. slab—suspended ceiling	.33	.14	.10	58%	70%
Slate, Tile or Composition Shingle Roof	.56	.16	.12	71%	79%

These Coefficients are computed from the results of tests conducted on actual wall sections by Prof. Gordon B. Wilkes, M.I.T. and by Prof. F. B. Rowley, U. of Minn. They include the heat loss thru studs, joists, rafters, etc. Consequently, they give an accurate measure of the heat loss in an actual insulated structure.

ALFOL ADVANTAGES

UNAFFECTED BY DAMPNES

ALFOL is non-porous and has no capillarity. It cannot absorb moisture. Moisture infiltration in non-metallic insulations increases conductivity frequently as much as 20 per cent. Such variations in conductivity cannot occur in ALFOL.

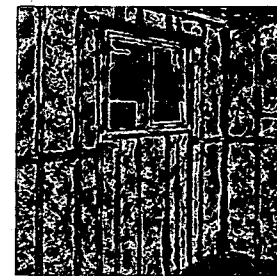
INSULATING EFFECT

Transmittance coefficients (see table above) are determined from tests made on actual wall sections insulated with ALFOL. They accurately measure ALFOL's insulating effect including the heat loss through and around studs. Published coefficients for other materials are based on 12 in. x 12 in. panel tests—materials preheated in "bone-dry" condition and at specified densities. They do not include the heat loss at studs, nor the effect of moisture infiltration, variation in density, thickness, etc. Many noted authorities on heat transfer recognize that the performance of ALFOL in actual use is unexcelled.

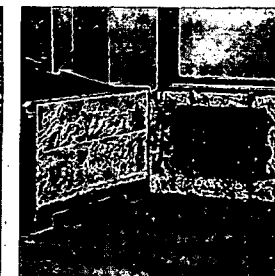
DUST AND DIRT cannot appreciably affect ALFOL's uniformly high insulating value.

FIREPROOF, CLEAN AND SANITARY

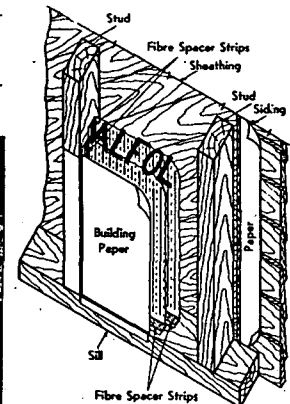
No Heat Storage Capacity—Preheats or cools in approximately one-fourth the time required for other insulations.



This Photo Clearly Shows How Alfol Stops Cold Air Leaks Around Windows and at Studs



Recessed Radiators Give off About 16° More Heat When Insulated With Alfol



Alfol is Flanged on Edges and Nailed to Face of Studs, Joists or Rafters, Through Fibre Spacer Strips

Alfol Reflects Heat as a Mirror Reflects Light

The Celotex Corporation

919 N. Michigan Ave., Chicago, Ill.

Mills: NEW ORLEANS, LA.

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SPOKANE, WASH.
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PARIS, FRANCE
LONDON, ENGLAND

CELOTEX
BRAND

INSULATING CANE BOARD

(Reg. U. S. Pat. Off.)

Builds - Insulates - Decorates - Subdues Noise

CELOTEX PRODUCTS:

Building Board
Lath
Sheathing Board
Finish Plank
Tile Board
Adhesives
Batten Strips
Hard Board
Tempered Hard Board

Black Tempered
Hardboard
Tempered Concrete
Form Board
Panel Board
Studio Board
Hardboard Tile (Tempered)
Roof Insulation
Vaporproofed Roof
Insulation

Vaporproofed Low Tem-
perature Insulation
Ferox Insulating
Protection Course
Insulation Blocks
Ornaments and
Mouldings
C-X Wallboards
Rock-Wool Products

Celotex Cane Fibre Insulation

In the manufacture of Celotex, long tough fibres of bagasse (cane) are properly refined, thoroughly sterilized, effectively water-proofed, firmly felted and securely interwoven to produce large boards of maximum strength consistent with the light weight necessary to assure high heat-stopping value. The "largest board in the world" is produced at Marrero, La., 12 ft wide and 1000 ft long! Commercial sizes are cut from this immense board by means of highly efficient automatic machines. Careful technical control assures uniform high quality.

The thermal conductivity of Celotex is 0.33 Btu per hour, per square foot, per 1 degree Fahrenheit, per inch thickness (based on a density of 13.5 lbs per cubic foot and a mean temperature of 70 °F.) Tests conducted at Armour Institute of Technology and at recognized laboratories confirm this figure.

The Celotex Corporation maintains an engineering and research staff which is available for investigations of all types of insulation installations. Engineers are invited to address their problems to The Celotex Corporation, Chicago, Illinois.

The Ferox Process

The Ferox Process (patented) is exclusive with Celotex. All Celotex cane fibre products are manufactured under the Ferox process (patented) and therefore effectively resist damage by Fungus Growth, Dry Rot and Termites (white ants).

The Ferox Process is not a surface treatment—it is integral. The chemical complex used is insoluble in water. It is non-volatile—odorless—permanent.

Celotex Building Board

The original cane fibre insulation. Neutral tan color. Two-surface utility. One side sanded; the other with a Tapestry texture. Size—4 ft wide, and 4, 5, 6, 7, 8, 8½, 9, 9½, 10, and 12 ft long. ½ in. and 1 in. thick.

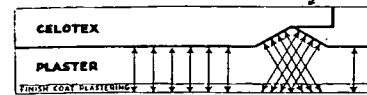
Celotex Tile Board

An attractive wall and ceiling treatment. Type Double A has an interlocking reversible bevel joint; smooth and tapestry textured surfaces alternated if desired. Suitable for all types of design. May be applied over existing walls and ceilings or as a new finish. Type A has beveled square edges; otherwise it is the same product as Type Double A. Neutral tan color. Also furnished in several well chosen tints to simplify the task of suitable decoration. With Type Double A Joint—½ in. thick only—with ¾ in. and 1 in. furnished on special order. Sizes range from 6 in. x 6 in. to 24 in. x 48 in.

Celotex Finish Plank—½ in. thick; 6 in., 8 in., 10 in., 12 in., and 16 in. wide; 6, 7, 8, 9, 10, and 12 ft long. Long edges Beveled and Beaded. Furnished with Type Double A, reversible bevel joint whereby either side can be exposed, thereby making possible a wide range of unusual designs.

Celotex Lath

A natural bond for plaster—a continuous plastering surface providing special resistance to lath cracks—eliminating lath marks—beveled edges to reinforce plaster—patented. Shiplapped joints (see diagram). Size: 18 in. x 48 in. ½ in., ¾ in., and 1 in. thick.



Celotex Sheathing Board

Full Thickness, sanded or unsanded. Insulation and structural strength. May be used with standard frames. ½ in., ¾ in., 1 in. thick; 4 ft wide; 4, 5, 6, 7, 8, 8½, 9, 9½, 10 and 12 ft long.



Celotex Ferox Insulating Protection Course

Waterproofing in some form is essential to the stability and efficient performance of many structures. Bituminous or membrane waterproofing is generally applied to areas exposed to hydrostatic pressure or conditions of dampness and moisture. Foundations and floors of buildings, tunnels, subways, and other construction below ground surface are frequently waterproofed. Swimming pools, reservoirs, retaining walls, bridges, and sumps or pits are similarly treated.

It is customary to protect bituminous floor waterproofing from damage by heavy traffic, and waterproofing on foundation walls from abrasion by back fill. Practically all construction specifications definitely require protection courses where waterproofing or dampproofing is used. Celotex Ferox Insulating Protection Course size: 22 in. x 47 in.; thicknesses, ½ in. and 1 in.

Celotex Vaporproofed Low Temperature Insulation

A moisture - proofed, water - proofed, vapor - proofed low density insulation. Each block encased in a sealed membrane. For all low temperature requirements, including Coolers (Beer, Meat, Creamery, etc.), Fruit and Vegetable Storage Rooms, Packing Plants, Fur Storages, Air-Conditioned Spaces, General Cold Storage Rooms, and Freezers. Conductivity 0.30 Btu per inch—odorless. Ferox-treated. Sizes: 18 in. x 18 in., 18 in. x 36 in., 9 in. x 36 in. Thicknesses: 1 in., 1½ in., 2 in., or any multiple of ½ in.

Celotex Rock-Wool Products

Highly effective wall-thick insulating material made from Molten Rock. Absolutely incombustible, vermin-proof and permanent.

Batts, wall thick, fit between studs and rafters. Light in weight (about 1½ lb per sq ft). Size: 15 in. x 23 in. **Loose** and **Granulated** available to place into open spaces in walls and floors.

Celotex Roof Insulation



Preferred as insulation over wood, concrete, steel, unit tile and poured gypsum roof decks. Size 22 in. x 47 in. Full thickness ½ in. Also furnished laminated from 2 to 8 plies. **Celotex Vaporproofed Roof Insulation**, Ferox treated, of special low density Celotex encased in vapor-proofing membrane.

Dependable Service

The Celotex Corporation distribution and service facilities cover the whole United States and 82 countries overseas. This world-wide organization includes trained insulation engineers who are at your service whenever some unusual insulating problem or project is before you.

Authoritative data, technical notes and specific material on thermal or sound insulation are available on request.

Write to the Celotex Service Bureau, 919 North Michigan Avenue, Chicago, Ill., for special problems.

Ehret Magnesia Manufacturing Co.

Valley Forge, Pa.

HEAT AND COLD INSULATIONS

HEATING—Ehret's 85% Magnesia, Air Cell Coverings, No. 18 Insulating Cement.

AIR CONDITIONING—Nicolfelt thermal and sound insulation, Ehret's Hair Felts, Cork Products.

85% MAGNESIA

Ehret's 85% Magnesia, in pipe covering, block and plastic forms, has become, during the past 40 years, renowned in every industry where insulating problems arise. It is pre-eminently suited for all temperature conditions up to 600°F.

Made from basic, hydrated, magnesium carbonate, the material which engineers agree is unsurpassed for heat insulating purposes due to its myriads of dead entrapped air cells.

Magnesia is manufactured from dolomite rock, mixed with asbestos fibre binder and moulded with a controlled pressure to insure proper strength without destroying the minute air cells. 85% Magnesia covering can be salvaged, if necessary, and ground up into cement. It is light weight and although it can absorb many times its own weight in water, it will dry out in service without impairing its efficiency. For outdoor use 85% Magnesia is furnished with an asphalt saturated asbestos felt jacket.

Substitutions for 85% Magnesia insulations are very expensive.

Its high thermal efficiency and long life free from shrinkage and deterioration make 85% Magnesia the most economical.

ENDURO HIGH TEMPERATURE INSULATION

For those having problems with insulation at temperatures from 600°F to 2000°F, Enduro will prove of special interest. Information on request.

Recommended Insulation Thickness for Flat Surfaces

	Temperature of Hot Surface				
	Under 300	300 to 450	450 to 600	600 to 700	700 to 800
Enduro.....	1 1/2	2 1/2	3	1 1/2	2
85% Magnesia.....	1 1/2	2 1/2	3	2	2
Total.....	1 1/2	2 1/2	3	3 1/2	4

(Plus a 1/4 in. Cement Finish.)

HEAT SEAL HOME INSULATION

Loose Wool—Granulated Wool—Wool Bats

PLUMBING—Ehret's Wool Felt Pipe Covering, Anti-Sweat Pipe Covering, Frost-Proof Pipe Covering.

ACOUSTICAL—Ehret's sound absorption and isolation felts.

No. 18 INSULATING CEMENT

A plastic having great adhesive property, it sticks firmly to any surface. No wire or mesh supports being necessary, the application is easy and economical.

Recommended for monolithic insulation. Highly efficient.

Low in cost due to its large covering capacity.

Is effective up to 1800°F.

HAIR FELT

Ehret's Standard Hair Felt, manufactured by the Platen Process, is composed of 100% pure cattle hair. Most efficient on temperatures from 100°F above to 60°F below zero. Its efficiency, particularly at sub zero temperatures, is due to the close felting of the hair, excluding moisture-laden air which might greatly harm other types from resultant condensation.

Ideally suited for insulating tanks, refrigerator cars, steel passenger and tank cars. Usually applied in layers, each of which is thoroughly air sealed.

For brine and ammonia work, we recommend a special "built-up" type of Hair Felt, fully described in our catalog "A-26." Punched Felts are also furnished for special conditions.

Thermal Conductivities in Btu's

Mean Temp.	85% Magnesia	Enduro	No. 18 Cement
100	0.420	0.465	0.520
200	0.460	0.500	0.560
300	0.510	0.550	0.610
500	0.590	0.620	0.700
800		0.730	0.840
1000		0.840	0.930

International Fibre Board Limited

Sales Offices

OTTAWA—MONTREAL—TORONTO—WINNIPEG
Administrative Offices and Mills: GATINEAU, QUE.

London Office

THE TENTEST FIBRE BOARD CO. LTD.
ASTOR HOUSE, ALDWYCH, LONDON, W. C. 2., ENGLAND



TEN/TEST is a manufactured lumber made from spruce fibres, solidly pressed under hydraulic pressure into a strong, homogeneous board. The fibres are chemically treated and water-proofed during process of manufacture, until the insulation is non-hygroscopic, free from capillary attraction and moisture-resisting in service commensurate with the maximum degree of insulation obtainable.

Official Tests

Conductivity. TEN/TEST has a conductivity of 0.33 B.T.U. per hour per square foot per degree fahr. per 1 in. thick. Authority: Professor E. A. Allcut, M. Sc. M. I. Mech. E. Mem. A.S.M.E. Professor of Applied Mechanics, University of Toronto. Tests performed by Hot-Plate method. Mean temperature 47.8 deg.

Tensile Strength 228 lb. per sq. in. Tests made on 1/16 in. board cut to strips 1 in. wide and tested in a Riehle Tensile Testing Machine, the grips being 2 in. apart. 228 lb. is the mean average of seven series of tests.

Transverse Strength (equal deflection) is 28.4 lb. Test made on 1/16 in. board, 6 in. wide, 18 in. long, on 12 in. centers, and load being applied to breaking point.

Plaster Bonding Strength 2163 lb. per sq. ft. Brown and scratch plaster coats were applied to standard 1/16 in. board, and the pull registered in an Olsen Testing Machine. Authority: Columbia University Testing Laboratories, New York.

Moisture Resisting. TEN/TEST, after complete immersion in water for 24 hours, registered 37.5% increase in weight.

NOTE.—Authority for tensile strength, transverse and moisture tests; J. T. Donald & Co., Ltd., Chemical Analysts and Engineers, Montreal, Que.

TEN/TEST Products

TEN/TEST Insulating Building Board. Standard insulation for use as exterior sheathing, interior finish; between walls and under floors for sound deadening. Standard Industrial Insulation for refrigeration and the prevention of condensation. Manufactured in convenient sizes: 4 ft. wide and up to 17 ft. long, 1/2 in. to 2 in. thick.

TEN/TEST Notch Board Plaster Base. Insulating plaster base having tongue and groove interlocking joints. Provides an effective bond with plaster without use of metal lath at joints. Sizes: 16 in. wide; 32 in. and 47 3/4 in. long. Thicknesses from 1/2 in. to 2 in.

TEN/TEST Roof Board. An effective roof insulation. Manufactured in two sizes: 1 x 4 ft. and 2 x 4 ft. Thicknesses from 1/2 in. to 2 in.

TEN/TEST Ashlar Block (Acousti "A"). For interior decoration and acoustical correction. Absorbs 35 per cent of incident sound at a frequency of 512. Can be supplied in a variety of designs and sizes to harmonize with any decorative treatment, allowing the architect much freedom in design and finish of churches, auditoriums, theatres, etc. Ashlar Blocks have bevelled edges, standard or to suit, can be left in the natural color or tinted as desired.

TEN/TEST Mouldings. An effective trim and finish for joints, corners, etc. Available in widths of 3/4 in. to 10 in. and lengths up to 12 ft.

TEN/TEST Moulding Edge Wall Panels. Conceals joints and provides excellent decorative treatment. Featured in widths 11 in. to 47 3/8 in., lengths up to 12 ft.

HYDRO/TEST. Water proof, insulating building board, designed particularly for low temperature requirements.

The Insulite Company

Executive Offices: Minneapolis, Minnesota

Factories:
International Falls
Minnesota
Kymi, Finland

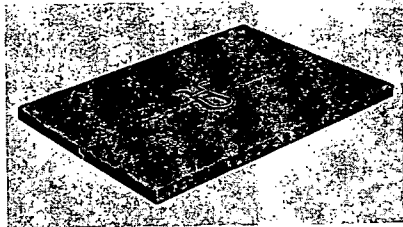
INSULITE
The Original Wood-Fiber Insulating Board

Stocked by Dealers
in All Principal
Cities Throughout
The World

INSULITE BUILDING BOARD

Insulite, the original wood fiber insulating board, has been specified by engineers and architects for 22 years—used for exterior sheathing or interior finish, duct lining, and other sound control work. Now available with two surface textures; fine screen and burlap, and in two colors; standard light color and Graylite.

Standard Building Board—This board is of a natural light color with high light-reflecting value. Thermal conductivity: 0.33 Btu/square foot/inch/hour/°F, based on a density of 16 lb cubic foot. Thicknesses: ½ in., ¾ in., 1 in.; sizes up to 4 ft x 12 ft.

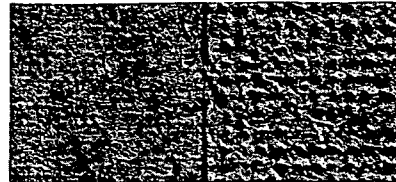


The Patented "Lok-Joint"

INSULITE TILE AND PLANK

Insulite Tile and Plank used on walls and ceilings, create attractive interiors for all types of buildings. They provide thermal insulation, and reduce the rever-

Tile—Insulite Tile, supplied in various sizes, is beveled on all edges; available with three types of joints; B-B, V-W, V-Lap; the latter two being interlocking, and the first two reversible, making it possible to combine surface textures. Standard material furnished in three surface textures; fine screen, burlap and smooth, and Graylite in burlap and fine screen only.



Fine Screen

Burlap

Graylite Building Board—Graylite is made from the same wood fibers as the standard board, but during the manufacturing process each fiber is covered with an asphalt emulsion, slightly increasing the density of the board, and greatly adding to its tensile strength and moisture-resisting qualities. Thermal conductivity: 0.35 Btu/square foot/inch/hour/°F.

INSULITE LOK-JOINT LATH

An insulating plaster base, with a patented "Lok" that firmly locks the sheets of Insulite together between supporting members, thus providing a rigid, level base for the plaster and insuring an adequate thickness of plaster. Fabricated from standard Insulite into units 18 in. x 48 in., shiplapped long edges. Thicknesses: ½ in., ¾ in., 1 in.

beration of sound. Fabricated from Standard light colored Building Board and Graylite Building Board with their respective thermal conductivities.

Plank—Random width Insulite Plank, in lengths up to 12 ft, can be applied either horizontally or vertically. The beveled long edges with B-B and V-W reversible joints, the latter interlocking, may be had either with or without bead. Supplied with two surface textures; burlap and fine screen in both the standard light color and the Graylite.

INSULITE ROOF INSULATION

Insulite Roof Insulation is made from the same wood fiber as standard Insulite. It can be used over any roof construction. The standard size is 24 in. x 48 in., for convenient handling, and with either offset or square edges. Furnished in thicknesses of ½ in., 1 in., 1½ in., and 2 in. Conductivity 0.32 Btu/square foot/inch/hour/°F.

Asphalted Roof Insulation

Insulite Asphalted Roof Insulation is a new and improved product, exclusive with Insulite. Treated with an asphalt emulsion during the process of manufacture, it provides those qualities of greater resistance to moisture and greater durability so desirable in roof insulation, combined with an ideal base for bonding to the roof deck and to the roofing. Furnished in full ½ in. thickness and in multiples of ½ in. up to 2 in. Size: 24 in. x 48 in.

INSULITE HARDBOARD PRODUCTS

Tough, durable grainless boards with hard, smooth, surfaces in various densities, and sizes up to 4 ft x 12 ft.

DualBoard—¼ in. thick; golden oak brown; has lowest density of all HardBoard products.

DeLuxe DualBoard—¼ in. thick with smoother surface and greater density than DualBoard.

HardBoard—⅜ in., ½ in., ¾ in., 1 in., 1½ in., golden oak color; much greater density than DualBoard.

Tempered HardBoard—Same thicknesses as HardBoard; burl walnut color; extremely high density.

PanelTile—Same as Tempered HardBoard except that it is scored in 4 in. squares to simulate tile.

INSULITE COLD STORAGE INSULATION

Cold Storage Insulation is a low density Insulite product which has found a wide acceptance for use in refrigerator cabinets, refrigerator cars, ice houses, storage plants, breweries and other places where low temperatures are maintained. The thermal conductivity is 0.29 Btu/square foot/inch/hour/°F and is available in eleven standard sizes and in thicknesses of 1 in., 1½ in., 2 in., 3 in., and 4 in.



Applying Insulite Roof Insulation

INSULITE SEALDSLAB

Fabricated from special low density board, each block of Seal slab is "sealed dry" with a ⅜ in. impregnation of a specially developed asphalt which insures an effective seal against moisture absorption after Seal slab is asphalt coated or dipped during application on the job. The factory primed surfaces also insure an excellent bond for subsequent asphalt coatings. Available in same sizes and thicknesses as the standard Cold Storage Insulation. Seal slab is especially adapted for use in freezers; meat, beer and creamery coolers; milk cooling tanks; fruit and vegetable storage rooms and general cold storage rooms.

INSULITE FIBEROCK

Insulite Fiberock insulation is a rock wool product, treated for moisture resistance, with low percentage of shot and with sufficient resiliency to prevent it from settling. Conductivity: 0.26 Btu/square foot/inch/hour/°F. Available in three forms:

Loose Fiberock—A fluffed form of rock wool insulation for hand packing in walls. Furnished in 35 lb bags, enough material to cover 15 sq ft surface area, wall-thick.

Granulated Fiberock—A granular form of rock wool which can be conveniently poured into place. Usually used over ceilings. Also supplied in 35 lb bags, enough to cover 13 sq ft surface area, 3½ in. thick.

Fiberock Bats—Rock wool in bats 15 in. x 23 in. to fit snugly between studs and joists. The bats are wall thick and require no special fastening. Packed in cartons of 9 bats each.

Johns-Manville

Executive Offices
22 East 40th Street, New York, N. Y.
Offices in All Large Cities



Johns-Manville Home Insulation

J-M Home Insulation involves the application of J-M Rock Wool (actual rock fibre) within the walls and the roof or attic spaces of residences, stores, apartment buildings and other structures.

J-M Home Insulation is *thick* insulation, remarkably effective in providing year round comfort and in reducing fuel bills. (On hot summer days homes insulated with this material are up to 15° cooler; in winter fuel bills are reduced up to 40%). J-M Home Insulation is non-combustible, sanitary and odorless, and will not support vermin.

Furnished in two forms: Type A for blowing into existing construction; and Type B bats for new homes.

Blown Method—Type A

In the Type A form, Rock Wool is blown by air into the spaces between studs in outer walls and between rafters or joists in attic floors. The insulation thickness in the walls corresponds to stud depth, approximately 3½ in., and the density does not exceed 10 lb. per cubic foot. J-M Home Insulation has been installed by this method in thousands of existing homes.

This type of material is installed by J-M Approved Home Insulation Contractors, who are equipped with the necessary apparatus.

J-M Insulating Board and Plaster Lath

J-M Insulating Board is a light weight, efficient material, having high insulating value and moisture resistance and unusual structural strength and rigidity. Furnished 4 ft. wide, in lengths up to 12 ft. Thicknesses ½, ¾ and 1 in.



Applying J-M Home Insulation bats in new home

Bat Method—Type B

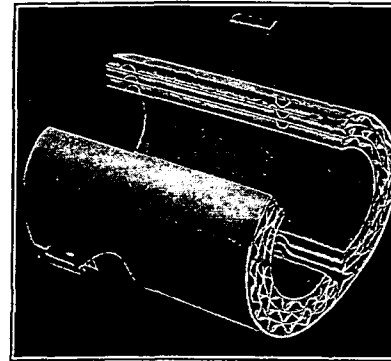
Home Insulation Type B is furnished in resilient bats, 15 in. by 18 in., or 15 in. by 23 in., full stud thickness. This form is widely used for homes or buildings under construction. The bats can be readily pressed between studs or beams and are easily cut or torn to fit odd-shaped spaces, conforming perfectly to the space occupied and providing a continuous insulation of even density.

Data and Specifications

For technical data and specifications on both types of J-M Home Insulation, write for brochure HI-17A.

J-M Insulating Lath is the same material as Insulating Board, except that it is furnished 18 in. by 48 in. with long edges ship-lapped and all edges beveled. The material is furnished in thicknesses of ½, ¾ and 1 in.

Johns-Manville Pipe and Boiler Insulation



J-M Pre-Shrunk Asbestocel Pipe Insulation

J-M Pre-Shrunk Asbestocel Pipe Insulation

J-M Pre-Shrunk Asbestocel is a radically improved material in which, by the use of waterproofed asbestos paper, shrinkage troubles have been minimized. It is used for hot water or low pressure steam piping, including supply and return mains, branches and risers.

Supplied in three finishes: The regular canvas finish; and the new high-speed asbestos paper or aluminum finished material which clinches on tight with quick-fastening staples.

All types are furnished in 3-ft. sections in standard thicknesses of 2, 3, and 4 plies, each ply approximately ¼ in. thick.

J-M 85% Magnesia

Recommended as the most efficient 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 ½ in. to 4 in. thick.

J-M Pre-Shrunk Wool Felt

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 waterproofed felts shrinkage troubles have been minimized.

Supplied in two finishes, the regular canvas and a smooth, dull-coated aluminum. In either 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 pipe sizes from ½ in. to 5 in.

J-M Asbesto-Sponge Felted

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.

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 block forms.

J-M Pre-Shrunk Asbestocel Sheets

Asbestocel Sheets are used for insulating warm-air ducts, flues, heater casings and fan housings in the ventilating system. Temperature limit 300 F. Furnished 6, 9, 12, 18 and 36 in. wide by 36 and 72 in. long, from ½ in. to 4 in. thick.

J-M Rock Cork

J-M Rock Cork is made of rock wool and a moisture-proof binding ingredient molded into sheets for insulating refrigerated rooms and air conditioning ducts. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance, its high insulating efficiency is maintained indefinitely.

Furnished 18 in. by 36 in., in 1½, 2, 3 and 4 in. thicknesses. Also 18 in. by 18 in. by 1 in. thick.

Detailed Specifications

Specifications for the use of any J-M Insulating Material may be had on request.

Mundet Cork Corp.

450 Seventh Avenue

New York, N. Y.

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork, Natural Cork Isolation Mats, Cork Tile, Cork Bulletin Board, and all kinds and varieties of Cork Specialties.

Branches

ATLANTA, GA.
BOSTON, MASS.
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Agents

BALTIMORE, MD. The McCormick Asbestos Co.
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CLEVELAND, OHIO G. S. Ross
FREDERICKSBURG, VA. Fredericksburg Insulation Co.
HARTFORD, CONN. The Hartford Cement Co.
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OKLAHOMA CITY, OKLA. Standard Roofing & Material Co.

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PORTLAND, OREGON Pacific Asbestos & Supply Co.
SALT LAKE CITY, UTAH Louis A. Roser
SEATTLE, WASH. Pioneer Sand & Gravel Co.
TULSA, OKLA. Standard Roofing & Material Co.
UTICA, N. Y. George Weisenberger
WASHINGTON, D. C. Asbestos Covering & Roofing Co.

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers at all times to assist and advise in the preparation of specifications pertaining to cork. This service is also available to any one who has a cold insulation or a vibration isolation problem, and is rendered without obligation. Our complete catalogue is filed in Sweet's Architectural Catalogue, and will be sent on request. It is replete with valuable information and data that should always be within reach of every specification writer whose field touches our products.

Contract Service

We contract for the erection of our products. In this way we may be certain that our material is installed in accordance with best established practice. This gives a definite advantage to an owner, in that divided responsibility for a given installation is eliminated. No contract involving cork is too large, too small or too far away. All materials and workmanship are unqualifiedly guaranteed.

Mundet "Jointite" Corkboard

Mundet "Jointite" Corkboard is 100 per cent pure cork, fabricated in accordance with the U. S. Government Master Specification, and is unsurpassed in its field. It is used for all cold insulation services and for acoustical correction. We manufacture only one grade of corkboard. Mundet "Jointite" Corkboard is sold in the standard 12 in. x 36 in. sheet. Standard thicknesses are $\frac{1}{2}$ in., 1 in., $1\frac{1}{2}$ in., 2 in., 3 in., 4 in. and 6 in.

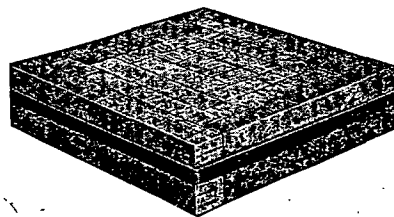
Mundet "Jointite" Cork Pipe Covering

Mundet "Jointite" Cork Pipe Covering is the complement of Mundet "Jointite" Corkboard and is used for all types of cold lines. The three thicknesses in which it is

manufactured make it suitable for pipes carrying sub-zero to 50 F. temperature. The pipe covering comes in sections 36 in., long. A complete line of standard fitting covers is available in the three thicknesses.

Mundet Cork Vibration Isolation

The transmission of machine vibration can be easily and permanently prevented by the use of Mundet cork isolation. The machines commonly associated with the heating and ventilating industry are best isolated with Mundet Natural Cork Isolation Mat. This form of isolation is fabricated from blocks of pure cork. These blocks are held together within a rigid steel frame or bound with asphalt paper applied with hot asphalt top and bottom. Steel bound isolation mat is



Above is shown a Steel Bound Mundet Natural Cork Isolation Mat. Note the natural cork strips within the steel frame.

usually used under exposed mounts, and asphalt paper bound isolation mat, under concrete foundations, of the envelope type. Mats are constructed to fit under any type of machine foundation.

For loads exceeding 2,000 lb per square foot, we manufacture Mundet Machinery Isolation Cork, which is a board form of compressed granulated cork and comes in three densities.

Both types of isolation are furnished in 1 in., $1\frac{1}{2}$ in., 2 in., 3 in., 4 in. and 6 in. thicknesses, depending on the class of service.

Silvercote Products, Inc.

161 E. Erie Street, Chicago, Ill.



SILVERCOTE INSULATION

Fabric—Manufactured from two external sheets coated with a mineral pigment and polished to a highly reflective surface possessing the essential characteristics hereinafter described for all Silvercote surfaces. The sheets are laminated with specially prepared asphaltum and reinforced with an interlining of fabric, imbedded in the lamination, consisting of $\frac{1}{2}$ in. mesh of strong jute cords. Two silver-like surfaces are exposed. The material is strong, flexible; and can be creased, folded, and tucked into the most inaccessible places without injury. Weight: 80 lb. per 1000 sq. ft.

Coreboard—Laminated of 7-ply of pulp board. 16 point liners coated on the exposed sides and polished to highly reflective surfaces. Finished board is $\frac{3}{8}$ in. thick with 2 silver-like surfaces exposed. Weight: 1000 lb. per 1000 sq. ft.

Insulation Board—Manufactured in a manner similar to wood fibre wallboard, to which is applied a top liner coated with pigment and polished to a highly reflective surface. Reverse side has a manila liner, dead level, beater sized and tacky; a perfect surface for painting. Finished board is $\frac{3}{16}$ in. thick. Weight: 550 lb. per 1000 sq. ft.

SILVERCOTE SURFACES

The surfaces of all Silvercote products consists of a non-metallic, homogeneous pigment, polished into a silver-like sheen which reflects radiant heat to a marked degree. They possess the following essential properties:

1. The surfaces are not metallic and are consequently free from oxidation, to which all metal surfaces (except gold and platinum) are subject.
2. The surfaces are waterproof.

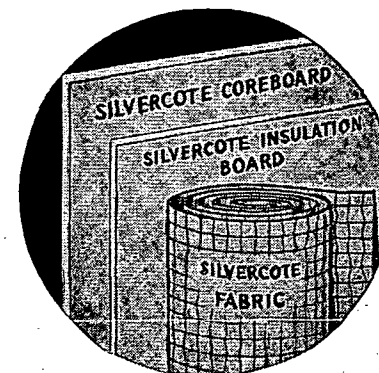
3. The surfaces are highly moisture resistant, equal in this respect to sheet lead.
4. The surfaces are not affected by acids or gases encountered in the customary uses of thermal insulation, whether in building construction, refrigeration or the cold storage fields.
5. The surfaces are entirely homogeneous and impervious to air infiltration wherever used.
6. The surfaces are non-conductors of electric and thermal energy and are designed to establish a difference in surface temperatures on transverse surfaces in the direction of heat flow.

Silvercote Insulation Fabric-Conductance.
0.33 = Resistance 3.03

Silvercote Insulation Board-Conductance.
0.49 = Resistance 2.04

Silvercote Insulation Coreboard-Conductance
0.265 = Resistance 3.77

For complete specifications and technical data see Sweet's Catalog, or address Silvercote Products, Inc., 161 East Erie Street, Chicago, Ill.



Reynolds Corporation

Executive Offices: 19 Rector Street, New York City

Western Sales Representative

400 Wrigley Building

Chicago, Ill.



Pacific Coast Sales Representative

345 Ninth Street

San Francisco, Calif.

REYNOLDS METALLATION*

Metallation* is bright, polished, genuine sheet aluminum, cemented to one or both sides of heavy, tough kraft paper. Properly installed, it serves as effective heat insulation for dwellings and buildings, reducing heat losses through walls and roof during winter, and maintaining more comfortable conditions in hot weather.

Metallation because of its light weight, has negligible heat storage capacity. In summer very little heat is stored in the insulation, to be released into the rooms after sundown and in winter very little heat is absorbed from within the room. Buildings which are intermittently heated or cooled can be heated or cooled quickly and economically.

Type A—Genuine sheet aluminum cemented to both sides of heavy, tough kraft paper with 1 1/4 in. plain paper margins exposed. Type A is especially designed and scored for installation between studs, joists or rafters, with paper edges nailed inside framing members to form a curtain of insulation between two dead-air spaces.

Type B—Genuine sheet aluminum cemented to both sides of heavy, tough kraft paper. Type B is especially designed for application horizontally or vertically over studs, rafters or joists, in a way that provides an air space on each side of the Metallation.

Type C—Genuine sheet aluminum cemented to one side of heavy, tough kraft paper. Type C is especially designed for use as building paper, under sheathing or roof boards and in types of construction in which only one surface can face an air space.

Type	Width of Roll	Length of Roll	Weight of Roll	Area of Roll
A	17 in	175 ft 5 in	18 lb	250 sq ft
B	32 in	93 ft 9 in	16 lb	250 sq ft
B	25 in	120 ft	16 lb	250 sq ft
C	32 in	93 ft 9 in	13 lb	250 sq ft

*Metallation, Metallated Ecod and Ecod are trade-marks of Reynolds Corporation, duly registered in the U. S. Pat. Off.

See SWEETS Architectural Catalog for complete details or write the nearest Reynolds Corporation sales office. Metallation is sold by leading lumber and building material dealers throughout the U. S. Used by U. S. Government, leading architects and engineers.

ADVANTAGES OF METALLATION

High, lasting, insulating efficiency which remains unimpaired by moisture absorption. Non-corrosive. Low weight and bulk. Wind-proof, vermin-proof and odorless. Low cost. Easy to apply. Saves fuel; reduces heating and air-conditioning requirements.

NOTES FOR HEAT TRANSFERENCE

TABLE on opposite page

aComputed by standard methods from data in AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS GUIDE, Vol. 12 pp. 71-92, 1934, and from "Thermal Insulation of Buildings," American Architect, May, 1934, pp. 89-100

bOutside film resistances (f_o) are based on a 15 mph wind velocity; inside film resistances (f_i) are based on still air.

cAll air spaces are at least 3/4 in. wide.

dThe thicknesses of 1 in. yellow pine sheathing and flooring is approximately 3/4 in.

ePlaster 3/4 in. to 1 in. thick is applied to ECOD or expanded metal plaster base.

fWall insulated as in Column IV with added 1 in. air space between brick and sheathing faced with Type C Metallation on outside of sheathing.

gFour inches of face brick and four inches of common brick.

hCurtain of either Type A or Type B Metallation in furred space, having one side facing masonry and the other side facing Type C or ECOD Metallation.

iThe thickness of oak flooring is approximately 1 1/4 in.

jType A of Type B Metallation is so applied to joists or rafters, or flanged between them as to form an air space. One side of the Metallation faces the building material and the other side faces still air.

kType A or Type B Metallation is so applied to joists or flanged between them as to form an air space. One side of Metallation faces Type C or ECOD Metallation, the other faces still air.

lSpace insulated as in Column IV except that Metallation nearest still air is Type A or Type B.

mNailing strips measure approximately 3/8 by 3 1/2 in. and are spaced 2 in. apart.

nType A or Type B Metallation is applied between rafters and nailing strips to form one air space facing Metallation and a series of small air spaces facing shingles.

oRoof insulated as in Column IV except that Type A or Type B Metallation is applied as in Note N

pType A or Type B Metallation is applied between nailing strips and rafters and also to under side of rafters.

qType A or Type B Metallation is applied as in Note P and also as a central sheet parallel with the roof.

PERCENTAGE OF HEAT TRANSFER STOPPED BY REYNOLDS METALLATION*

Coefficients (U) are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides of the construction. Columns (%) show the percentage of heat which REYNOLDS METALLATION stops from flowing through the respective constructions.

ILLUSTRATION	Construction Number	CONSTRUCTION	I II III IV							
			Uninsulated	Type C or Ecod Metallation on one side of air space. (Except where noted)	Type C or Ecod Metallation on both sides of air space. (Except where noted)	Type A or Type B Metallation forming curtain between two air spaces. (Except where noted)	Type A or Type B Metallation forming curtain between two air spaces, faced on opposite sides with Type C or Ecod Metallation (Except where noted)			
			U %	U %	U %	U %	U %	U %	U %	
	1	FRAME OR VENEER WALL Shingles or Clapboards, Sheathing, ^d Stud Space, Ecod Plaster, ^e	.26 0	.19 27	.18 31	.14 46	.13 50			
	2	Drop Siding, no Sheathing, Stud Space, Plaster	.37 0	.25 32	.24 35	.16 57	.15 59			
	3	1-In. Stucco with Ecod Base, Stud Space, Plaster	.50 0	.31 38	.28 44	.18 64	.17 66			
	4	1-In. Stucco with Ecod Base, Sheathing, Stud Space, Plaster	.33 0	.24 27	.22 33	.16 56	.14 58			
	5	4-In. Face Brick Veneer, Sheathing, Stud Space, Plaster	.30 0	.22 27	.21 30	.15 50	.11 (f) 63			
	6	MASONRY WALL 8-In. Solid Brick, ^e Furred Space, Plaster	.32 0	.23 28		.15 53	.15 (h) 53			
	7	4-In. Face Brick on 8-In. Hollow Clay Tile, Furred Space, Plaster	.25 0	.19 24		.13 48	.13 (h) 48			
	8	1-In. Stucco on 12-In. Hollow Clay Tile, Furred Space, Plaster	.22 0	.17 23		.13 41	.12 (h) 46			
	9	12-In. Hollow Concrete Blocks, Furred Space, Plaster	.32 0	.23 28		.15 53	.15 (h) 53			
	10	FRAME MIDDLE STORY FLOOR Single Yellow Pine Floor, ^d Joist Space, Plaster	.30 0	.22 27	.21 30	.15 50	.14 53			
	11	Oak Floor, Yellow Pine Sub-Floor, Joist Space, Plaster	.25 0	.19 24	.18 28	.13 48	.13 48			
	12	FRAME ATTIC FLOOR No Floor, Joist Space, Plaster	.69 0	.26 (i) 62	.24 (k) 65	.26 (i) 62	.15 (i) 78			
	13	Single Yellow Pine Floor, Joist Space, Plaster	.30 0	.22 27	.21 30	.15 50	.14 53			
	14	FRAME FIRST STORY FLOOR Single Yellow Pine Floor, Joist Space, no Ceiling	.46 0	.22 (j) 52	.21 (k) 54	.22 (j) 52	.14 (j) 70			
	15	Oak Floor, Yellow Pine Sub-Floor, Joist Space, no Ceiling	.35 0	.19 (j) 46	.18 (k) 49	.19 (j) 46	.12 (j) 66			
	16	MASONRY FLOOR—SUSPENDED CEILING 4-In. Reinforced Concrete, Furred Space, Plaster	.37 0	.25 32		.16 57	.16 (h) 57			
	17	6-In. Reinforced Concrete, Furred Space, Plaster	.35 0	.24 31		.16 54	.15 (h) 57			
	18	4-In. Concrete, 4-In. Hollow Tile, Furred Space, Plaster	.27 0	.20 26		.14 48	.14 (h) 43			
	19	4-In. Concrete, 8-In. Hollow Tile, Furred Space, Plaster	.23 0	.18 22		.13 43	.12 (h) 48			
	20	FRAME ROOF Wood Shingles, Nailing Strips, ^m Rafter Space, Plaster	.30 0	.22 27	.18 (n) 40	.15 50	.12 (o) 60			
	21	Wood Shingles, Nailing Strips, Rafter Space, no Ceiling	.46 0	.22 (i) 52	.18 (p) 61	.22 (i) 52	.12 (q) 74			
	22	Composition or Asphalt Shingles, Nailing Strips, Rafter Space, Plaster	.37 0	.25 32	.20 (n) 46	.16 57	.13 (o) 65			
	23	Composition or Asphalt Shingles, Nailing Strips, Rafter Space, no Ceiling	.64 0	.25 (i) 61	.20 (p) 69	.25 (i) 61	.14 (q) 78			
	24	FRAME ROOF Wood Shingles, Sheathing, Rafter Space, Plaster	.27 0	.20 26	.19 30	.14 48	.13 52			
	25	Wood Shingles, Sheathing, Rafter Space, no Ceiling	.40 0	.20 (i) 50	.19 (k) 53	.20 (i) 50	.13 (l) 68			
	26	Slate or Cement Shingles, Sheathing, Rafter Space, Plaster	.34 0	.24 29	.22 35	.16 53	.14 59			
	27	Slate or Cement Shingles, Sheathing, Rafter Space, no Ceiling	.56 0	.24 (j) 57	.22 (k) 61	.24 (j) 57	.14 (l) 75			

The Ruberoid Co.

Executive Offices

500 Fifth Avenue, New York, N. Y.

Divisional Offices

NEW YORK

CHICAGO

BOSTON (MILLIS)

ERIE

BALTIMORE

MOBILE

RUBEROID RESIDENTIAL INSULATING PRODUCTS

Genuine RU-BER-OID Mineral Wool

From the viewpoint of efficiency, Mineral Wool is one of the finest insulating materials for residential construction. It can be used in varying thicknesses up to 4 in. It cannot deteriorate or decompose, and is fire- and vermin-proof. This insulation can be expected to save 20 to 35 per cent in fuel bills in the winter, and make the interior of the home 10 to 15 deg cooler in summer.

Genuine RU-BER-OID Mineral Wool in the loose or bulk form is clean, silky and

long fibred, a quality which overcomes the tendency to pack and settle. It is free from harsh brittle particles. This wool is offered in three forms—*loose* or bulk for packing, *granulated* for pouring, and in *pre-formed bats* for use between joists, rafters, and studding.

Loose Mineral Wool

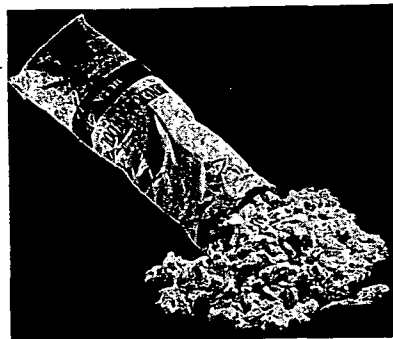
The loose mineral wool is packed in 35 lb bags containing about 3½ cu ft. A bag will cover in excess of 21 sq ft gross (including the studs or joists) 3½ in. thick when packed to a service density of 6 lb per cubic foot.

Granulated Mineral Wool

Granulated Mineral Wool is in pellet form averaging from the size of a pea to a marble. In this form it is easily poured between studs and joists and for filling in irregular spaces. Bags contain 40 lb. When applied to a service insulation density of about 7 lb per cubic foot, 4 in. thick, it gives a coverage of about 17 sq ft per bag.

Mineral Wool Bats

Mineral Wool Bats are packed in cartons containing 9 pieces, 15 sq ft. The bats are wall thickness (approx. 4 in.) measuring 15 in. x 18 in. which permits easy application between the studding of side-walls, or between the rafters under the roof.



The long silky fibres make this wool extremely stable, overcoming the tendency to pack and settle

RUBEROID INDUSTRIAL INSULATING PRODUCTS

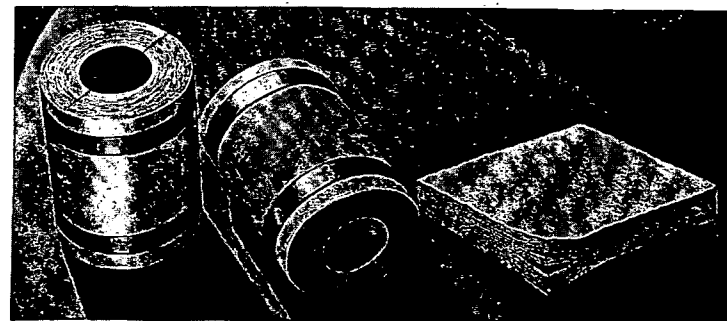
85 Per Cent Magnesia for Medium and High Pressure Steam Lines

85 per cent Magnesia pipe coverings are made of approximately 85 per cent pure carbonate of magnesia and 15 per cent carded long Asbestos fibre. Temperature limit 600 F. Light in weight, fire-proof, extremely high insulating value, yet have maximum mechanical strength consistent with efficient insulation. Supplied in

various thicknesses up to 3 in., canvas jacketed.

For Higher Temperatures MINERAL WOOL BLANKETS Limit 1800 F

Mineral Wool for flat or curved surfaces is manufactured into blankets by using annealed long fibre wool, felting it and building it up to the required thickness



Supercell Pipe Covering has 14 to 16 laminations of indented Asbestos Felt to the inch

between metal fabrics of various types. These blankets are secured with galvanized wire between the metal fabrics. Thickness from 1 in. to 6 in. Standard size blankets 24 in. x 96 in. and 24 in. x 48 in.

High Temperature Pipe Coverings Limit 1600 F

This insulation is recommended for insulating surfaces having temperatures between 600 F and 1600 F. It has low thermal conductivity, practically no shrinkage, is light in weight, and has good mechanical strength.

Supercell Pipe Covering for Low and Medium Pressure Steam Lines Limit 500 F

Supercell Pipe Insulation—A new development in the low and medium pressure field, has a temperature limit of 500 F. It has 14 to 16 laminations of indented Asbestos Felt to the inch thickness. It is considerably lighter than the old-fashioned pipe insulation, yet, due to its unique construction, provides as much as 35 per cent higher efficiency.

Other Insulating Materials for Low and Medium Temperature Steam Lines—Air Cell—Limit 350 F

Made 2, 3 and 4 ply—each ply ¼ in. thick. Furnished in either the new Pyroxylin finish or with canvas jacket. Temperature limit 350 F. Ideal for industrial hot water lines.

Watco Cell Limit 500 F

Close corrugation, 6 and 8 ply to the inch, raises efficiency standards. Either new Pyroxylin finish or canvas jacketed, as desired. Temperature limit 500 F.

Woolfelt For Hot and Cold Water Lines

Special surfacing process increases efficiency 20 per cent to 30 per cent. Weight also is reduced 20 per cent to 30 per cent. Made in three styles of liners: Asbestos Felt Paper for hot water lines; Water-proofed Tar for cold water lines; "Twin Purpose" for both hot and cold water lines. Furnished in either the new Pyroxylin finish or with canvas jacket.

Sheet and Block Insulations

Any of the pipe coverings can be made into sheet and block form. Standard sizes 6, 12, 18 or 36 in. x 36 in. Ideal for insulation of flat or irregular surfaces, such as tanks, boilers, breechings, etc.

Insulating Cements

For the finishing of asbestos sheet and block insulation, or the insulation of valves, fittings, or flanges, or any sort of irregular surfaces, the line of Ruberoid-Watson Insulating Cements is quite complete. Will take care of temperature conditions from 100 to 2000 deg F.

Detailed Specifications

Complete catalog giving specification data of any Ruberoid Insulating Product may be had upon request.

The Standard Lime & Stone Company

First National Bank Building, Baltimore, Md.

Manufacturers of
Capitol Rock Wool
Insulations



Representatives
in all
Principal Cities



The Bat is readily installed where studding is open or in new construction



The blowing method is used in existing houses between walls or floors

Home Insulation for year-round comfort is the demand of the home owner today, because a properly insulated home saves fuel, resists fire, deadens sound, is cooler in summer and warmer in winter, and sells more easily.

Capitol Rock Wool is the nearest approach to perfect insulation. In winter it provides even temperatures all over the house, eliminates drafts, and reduces fuel bills from 20 to 40 per cent. In summer, the temperature inside will be 8 to 15 deg.

cooler than outside. Variation between first and upper floors will be negligible. Capitol Rock Wool is also fire-proof, vermin-proof, and water-resistant.

In new construction it is easily installed in compact, felted Bat form by the ordinary workman. Weight: 6 lb. per cubic foot. Thermal conductivity: 0.257 BTU. In houses already built, whatever the type of construction, Capitol Grade "A" Blowing Fibre is pneumatically conveyed into the wall air spaces.

INDUSTRIAL INSULATIONS

Blankets—A reinforced felted Capitol Rock Wool designed primarily for insulating boiler settings, fire walls, boiler breechings, smoke stacks within buildings, tank cars, hot water heaters, boilers, oil stills, bubble towers, large diameter pipes, ducts, etc., and for absorbing sounds set up by machinery.

Insulating Blocks—A compressed pliable product for temperatures up to 750 F., for insulating domestic furnaces, boilers, and hot water tanks; and for commercial boilers, tanks, ducts, stills, and breechings.

Pipe Coverings—With canvas or waterproof cover for temperatures from 60 F. below zero to 1250 F.

Insulating Cement—Plastic; quickly and neatly applied: available in both high and low temperature cements.

CATALOGS and Specifications of our complete line of home and industrial insulations gladly sent on request.

United States Gypsum Company

300 W. Adams Street, Chicago, Ill.

Sales Offices:

ATLANTA, GA.
BOSTON, MASS.
BUFFALO, N. Y.
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SAN FRANCISCO, CALIF.
WASHINGTON, D. C.

PRODUCTS:

Building Board.
Insulating Mouldings.
Tongue and Groove Sheathing.
Insulating Lath.
Metal Reinforced Insulating Lath.
Roof Insulation.
Insulating Tile Board.
Insulating Tile.
Insulating Plank.
Bulk Wool.
Strip Wool.
Wool Bats.
Granulated Wool.

RED TOP WEATHERWOOD, LATH, PLANK, SHEATHING AND TILE

Weatherwood Insulating Board is a felted wood fiber product that has been treated to make it highly moisture resistant. It is formed on a single cylinder which produces a homogeneous board, free from laminations and tendencies to split through the center.

Low Thermal Conductivity

The average established in tests is 0.33 Btu's per hour, per square foot per inch thickness, per degree Fahrenheit difference in temperature.

Resistance to Moisture—Non-Absorption

Tests show that this board, after being submerged in water for a period of two (2) hours, has an average water absorption of less than 15 per cent.

Structural Strength

The tensile strength is over 350 lb per square inch, and the modulus of rupture, over 500 lb.

Durability

There is nothing in this board that will deteriorate prematurely and impair the

original insulation value. Since the board is homogeneous, the fibers being interwoven through the sheet, it will not split. The Moisture Resistant chemical with which the fibers are treated makes the board distinctly distasteful to rodents and insects.

Uniformity

Adequate laboratory control and mill inspection assure uniformity in density and structural strength.

RED TOP INSULATING WOOL

Description

Red Top Insulating Wool is an extremely light, fluffy wool insulation made of silica—a fireproof material.

The nature of the raw materials used, particularly as to purity, permits accurate manufacturing control—the product is absolutely uniform and consists of snowy white, long-fibre wool. It contains no "shot" or other non-insulating materials that add weight. Red Top Insulating Wool is springy and resilient—it will not mat.

Low Thermal Conductivity

The heat conductivity of Red Top Insulating Wool (1½ lb density) is 0.266 Btu's per inch thickness, per square foot, per hour, per degree Fahrenheit difference in inside and outside temperatures. (Tests by Professor Peebles, Armour Institute of Technology).

Unusually high in insulating efficiency, it is outstanding when, as customarily used, it is wall thick (4 in.). So used its rating is 0.066.

Light Weight (Density)

In its standard density, Red Top Insulating Wool weighs but 1½ lb per cubic foot.

COVERAGE OF RED TOP INSULATING WOOL

Type of Wool	Volume per package at standard density	Coverage per package at standard density	
		4 in. thick, including 2 x 4's	4 in. thick, excluding 2 x 4's
Strip Wool in cartons	11¼ cu ft	36 sq ft	33¾ sq ft
Bat Wool in cartons	12½ cu ft (20 bats)	40 sq ft	37½ sq ft
Bulk Wool in paper bags	6½ cu ft	20¼ sq ft	19 sq ft

Zonolite Corporation

General Offices

Fisher Building, Detroit, Michigan

Plants At

LONG ISLAND CITY JOLIET, ILLINOIS OMAHA DETROIT ST. LOUIS LIBBY, MONTANA

Trade Mark Reg. U. S. Patent Office



THE PERFECT INSULATION

ZONOLITE INSULATION FOR HOMES

ZONOLITE LOOSE FILL

Zonolite is a mineral which has been expanded many times by the application of intense heat. Each particle contains thousands of tiny dead air cells which make Zonolite a *non-conductor* of heat. And, in addition, every particle is filled with glittering, golden surfaces that make Zonolite a *reflector* of heat. Zonolite provides a natural DUAL insulation.

Zonolite has a uniform, tamper-proof density. Its *actual* insulating value in the walls of a house is always the same as the *theoretical* insulating value as shown by laboratory tests. Zonolite cannot be "fluffed" or stretched when being installed. It flows naturally, filling every nook and corner.

ZONOLITE POURS EASILY

Zonolite is easily installed in attics by pouring directly from the bag between



joists or rafters. When put between walls, Zonolite pours easily, requiring no machinery and taking a minimum of time and labor. Your local Zonolite representative will furnish accurate estimates of installation costs.

Zonolite is packed in 3½ cu ft bags with the following coverage:

20 square feet.....	2 in. thick
14 square feet.....	3 in. thick
11 square feet.....	3½ in. thick

AN EFFICIENT HEAT BARRIER

One of the many tests of Zonolite's resistance to heat passage was made by the Consolidated Gas, Electric Light and Power Co. of Baltimore. A hot plate test showed a thermal conductivity rating of 0.27. In addition, "U" factor tests show its effectiveness as an insulating material.

Zonolite is FIREPROOF—a 100 per cent mineral; VERMIN PROOF—rats or mice will not burrow or nest in it; termite-proof; SOUND ABSORBING—tests by V. O. Knudsen, acoustical authority, show Zonolite to be high in sound absorption and sound insulation: 3¾ in. of loose Zonolite, in panels of 3 ft x 3 ft 6 in., showed a coefficient of sound absorption of 0.58 at a frequency of 512 d.v.; DIELECTRIC—a test by J. C. Peebles, Armour

Institute of Technology, showed that 20,800 volts were required to puncture a one-inch layer of Zonolite between one-inch brass balls; CHEMICALLY INERT—tests of every description prove that Zonolite will not rot, rust, or deteriorate in any way. Zonolite forms a THICK insulation; is approved by leading air conditioning firms for use with their units.

ZONOLITE INSULATING PLASTER

A prepared mixture of carefully selected calcined gypsum and Zonolite. The growing interest in insulation is bringing this plaster into widespread use, as it provides an inexpensive insulation in cases where economic or construction difficulties render a complete insulating job imprac-

ticable. Tests made by J. C. Peebles of Armour Institute of Technology show heat conductivity of Zonolite Insulating Plaster, 0.90.

Since it contains no sand, Zonolite Insulating Plaster is very light in weight—Zonolite plaster weighs only 10 lb per yard on wall—thus it is extremely easy to handle and spreads rapidly. It is uniform, since it is not dependent upon varying grades of sand. It also possesses a high degree of sound-insulating value, and is free of map cracks.

Zonolite Insulating Plaster, in a texture finish, is furnished in a wide variety of soft, attractive tints.

This adds another advantage to the list which includes comfort in winter and summer, fuel-saving, sound insulating, fire safety and ease of handling.

ZONOLITE HIGH TEMPERATURE CEMENT

Ideal for insulating tanks, boilers, furnaces, etc., where the insulation must withstand temperatures up to 2300 deg. It has a thermal conductivity factor of 0.46 at 100 deg. and up to 1.9 at 2000 deg. ZONOLITE HIGH TEMPERATURE CEMENT BLOCKS are made for industrial oven insulation and have proved highly efficient. These blocks are made 1 ft x 3 ft with thicknesses varying from 1 in. up to 4 in.

ZONOLITE COLD STORAGE INSULATION

Made of Zonolite and a moisture-proof binding element. It is in sheet form, for insulating refrigerated rooms. Sturdy, serviceable; does not support vermin. It is so resistant to moisture that its efficiency will continue indefinitely.

Available in sheets of all sizes; 1 to 4 in. in thickness.

ZONOLITE INSULATED AIR DUCTS

(Patents applied for—all infringements will be prosecuted)

Recognizing the insulation defects of the ordinary air duct, Zonolite engineers decided to perfect the product as a whole rather than simply to engineer a new style of insulation for this type of work. In using the conventional type of sheet metal construction, the velocity of the air is greatly reduced because of the necessity of covering the inside of the duct with sound absorbing material, and as most of these coverings are not fire-proof, a fire hazard is introduced. In the Zonolite patented style of construction, the duct is constructed of a specially designed wire



Showing equipment used for extensive tests by the Detroit Edison Co.

1. Conventional-type Air Duct.
2. Zonolite Air Duct.
3. Zonolite Air Duct with Insulation Applied.

mesh rather than a solid strip of metal, and is then covered with Zonolite fire-proof Air Duct Insulation, which has unusual insulating and sound absorbing values and eliminates the necessity for an inner lining. Tests by leading air conditioning firms show a reduction in noise of 11½ decibels. With the Zonolite patented type of duct, it is now possible to use a much smaller duct and higher air velocities, which make for more pleasing, practical, and economical home and building installations.

Zonolite Patented Air Ducts are made exclusively by local sheet metal shops licensed by the Zonolite Corporation.

ZONOLITE INSULATED ROOFING

An unique combination gives a proper insulation and roofing material in one product. A combination of high grade asphalts and Zonolite is bonded together in such a way as to withstand great changes in temperatures without cracking or softening. The latest practice is to use Zonolite roofing with two plies of felts over it. This construction is practically permanent as the product is 100 per cent mineral composition and can be installed in any weather. In addition, if the roofing felts should ever crack, any entrance of moisture could not cause the product to deteriorate in any way, which makes the maintenance problem a very simple one; nothing more than an occasional mop coat being needed to make the roof last indefinitely.



Zonolite all-mineral roof block as installed on Chrysler Engineering Building.

Wilson & Co.

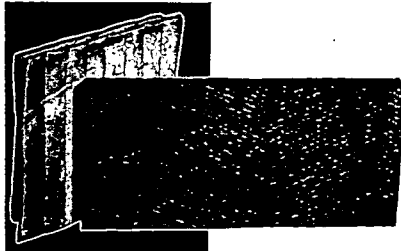
INSULATION
MATERIAL



FILTERAIRE
AIR FILTER

4110 S. ASHLAND AVE. CHICAGO, ILLINOIS

HAIRCRAFT, HAIRBESTOS AND NATURZONE



Modern specifications are demanding insulating materials that not only give adequate protection, but also lend themselves to the contours of modern design.

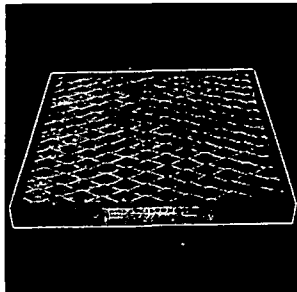
For this purpose, Wilson's Haircraft and Hairbestos (flexible blanket type materials) are ideal. Haircraft is made with 100 per cent hair filler between Kraft Duplex Asphalt paper, stitched in rows

INSULATION

every 2 in., the width of the material. It has a low conductance of 0.26 in a one-inch thickness. Hairbestos is of the same character as Haircraft, with the exception that it is fire-resisting and covered with crinkled asbestos paper.

Naturzone insulation is the standard board form of insulating material for use where a rigid form of material is necessary. It is made of sterilized, deodorized hair compressed into board form of standard size, 36 in. x 16 in. x 1 in., 1 1/2 in., 2 in. and 3 in. thick. It has a conductivity of 0.27 per square foot per hour per inch of thickness. The ability to knit itself into a single unit in a very short time makes the efficiency of Naturzone a superior form of insulation.

FILTERAIRE AIR FILTERS



maintain an expanded condition, pressing constantly outward against the sides of the container, preventing packing and sagging. A special fixative maintains the form of the entire pad. The outlet side of the pad is treated with a special adhesive oil for catching and retaining the dust.

Filter Cell Rating

Capacity per cell.....800 CFM
Face velocity.....300 FPM
Resistance per unit (two filters 0.18 in w.g. tandem arrangements).

These ratings are for standard 20 x 20 x 2 or 16 x 25 x 2 Filteraire filters. Special size filters are supplied on order.

The fibers on the face of the filter are formed mechanically into corrugations and are not coated with adhesive. This treatment allows the dust to enter the cell instead of packing up on the surface, insuring exceptional dust-holding capacity.

In the Filteraire filter, Wilson & Co. has introduced Keratin fibers (animal hair). They are thoroughly sterilized and deodorized and subjected to successive mechanical and chemical treatments.

In the individual cell, the central portion is built of fibers which are highly resilient. This characteristic helps to

The Ric-wil Company

Agents in
Principal Cities



ESTABLISHED IN
1910

Union Trust Bldg.

NEW YORK-CHICAGO-SAN FRANCISCO

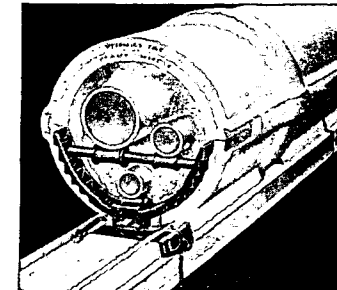
Cleveland, Ohio

Conduit—Standard conduit is vitrified salt glazed tile or cast-iron with Loc-liP Side Joints, bell and spigot type, unlined or lined. Tile and cast-iron in 24 in. sections, sizes 4 to 27 in. inside diameter. A Super-Strength tile conduit on which laboratory tests by A.S.T.M. 3 point method show an average crushing strength of approximately 3000 x per foot diameter per lineal foot. A light weight cast-iron conduit specially designed for bad water conditions and a heavy cast-iron conduit for extra heavy duty.

Base Drain—Standard Base Drain is vitrified salt glazed tile for tile conduit and extra heavy tile or cast-iron for the cast-iron conduit, in 24 in. lengths. Special U-Type Base Drain to hold return line. Support Blocks for use with round tile construction also available.

Pipe Supports—Standard Pipe Supports for tile conduit are of the externally supported type with rollers for single or multiple pipes. Load is carried entirely on side shoulders of Base Drain. Pipe Supports for cast-iron conduit are of the internal type and are cast integral with the conduit. Pedestal Pipe Supports also available. Details of Ric-wil Alignment Guides and Anchors furnished upon request.

Insulation—Dry-paC Waterproofed Asbestos Insulation, non-settling and non-corrosive—packed into the conduit becomes a solid matted mass without cracks, joints or openings—positively non-capillary and water-repellant. Has low conductivity and insures highest thermal efficiency—non-waterproof kind also furnished. Lined conduit insulation is in-



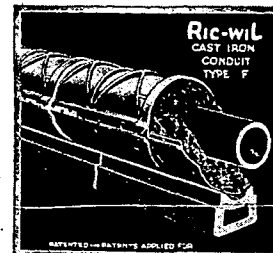
Cutaway view at pipe support showing how Ric-wil parts interlock. Note Loc-liP Side Joints and how pipe assembly is independent of the conduit. Cut shows Type DF System, lined conduit, with Dry-paC Waterproofed Insulation.

tegrally moulded with lining of diatomaceous earth mixture. Sectional pipe covering, sponge felt, 85% magnesia, etc., can also be furnished.

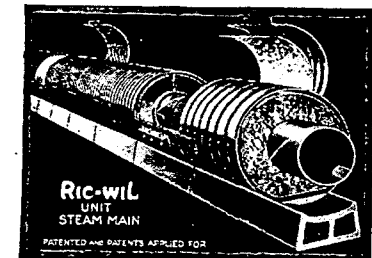
Accessories—Shutter sleeves, filter cloth, asphalt joint cement, waterproofing compound, manhole covers, and other accessories will be furnished as desired.

Engineering Service—Full cooperation with architects and engineers. Installation supervision if desired.

Technical Data—Tabulated Steam Heating Rates, Test Reports, Service Detail Bulletins, Catalog Bulletins, Central and District Heating Bulletins and Architects and Engineers Detail Sheets available upon request.



Ric-wil Cast Iron Conduit has ample strength with minimum weight for use under roadways, railroad tracks, or other places subject to extreme loads and vibration. It is installed without delays and without added construction or engineering.



Ric-wil Unit Steam Main, a prefabricated, ready-to-install unit, 13 1/2 ft long, including conduit (Arco Iron), pipe, insulation, and accessories. Ideal for speed and economy on district heating projects.

Underground Steam Construction Co.

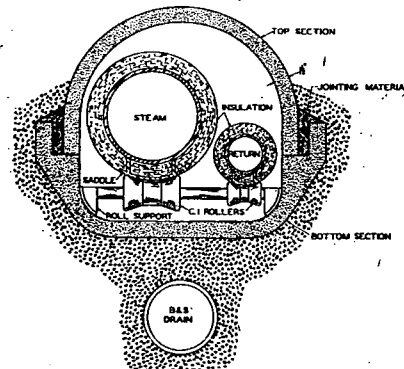
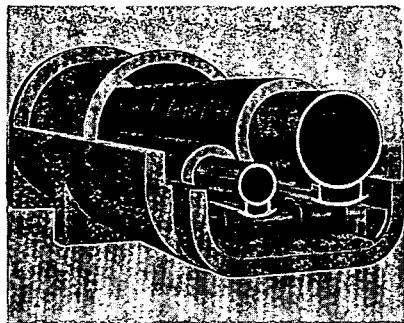
75 Pitts Street, Boston, Mass.

PRODUCTS—Engineering and Contracting of Steam Line Installations. Underground Steam Conduits.

USCCO PRE-CAST CONCRETE CONDUIT

This unit marks a long step forward in the economics and mechanics of laying underground steam lines.

Its Outstanding Features Are:



Strength resulting from flat reinforced bell; from the longitudinal joints, and from the fact its sections are 4 feet long, reducing the usual number of joints necessary.

Shape which allows 12 per cent greater inside capacity than circular conduit of like diameter. This permits larger pipes in a given size conduit and more room for drainage, pipe supports and rolls.

Ease of Installation resulting from the fact that all top and bottom halves mate. Bottoms may be laid for any distance, the piping installed and then the tops brought up and laid. This speeds up the work and protects idle halves from damage. They can be stored away from the job.

Materials are high strength cement with suitable aggregate, amply reinforced with wire mesh.

Pipe Supports are of cast-iron and quickly installed. They may be placed anywhere in the conduit. The weight of the pipe holds them in place.

Joints may be of any standard accepted brand of jointing compound or they may be of standard Portland cement mortar.

Builders Iron Foundry

23 Coddling Street

Providence, R. I.

Representatives in Principal Cities

The **CHRONOFLO ELECTRIC FLUID METER** for nearby or long distance transmission of flow rates, quantities, pressures, temperatures, etc.

The **SHUNT METER** for steam, air and gas.

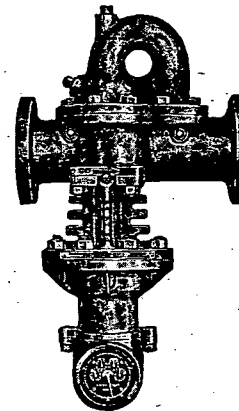
The **VENTURI METER** for boiler feed water supply, and other main pipe lines.

CHRONOFLO METERS AND CONTROLLERS

"From hundreds of feet to hundreds of miles"



Chronoflo—Mechanically measures flow, pressure, temperature, or position in terms of time; electrically transmits this time duration equivalent of the quantity to any point—or points—nearby or miles away. There a Receiving Instrument mechanically indicates, records, or registers it. If more desirable the same "impulse" may be used to operate flow, pressure, temperature, or position controllers. Regular A.C. current is used and accuracy is not affected by voltage variation. Suitable for nearby or long distance transmission over private wires or public telephone channels. Write for Bulletin 273.



Shunt Steam Meter, Type K.S.

THE SHUNT METER

A low priced, practical, mechanical, meter easily installed and accurate over a wide range. Regularly used by many District Heating Companies for measuring steam sold; also ordered and re-ordered by leading industrial plants to meter steam or air furnished tenants, departments, and processes.

The illustration shows the complete meter. Installation consists simply of bolting to flanges in the flow line; no connecting pipe or electric wiring. A portion of the entering steam is deflected by an orifice through nozzles against the blades of a turbine located in the upper or shunt passageway. The speed of the turbine is kept low by a dampening fan on the vertical turbine shaft, at the bottom end of which is a magnetic drive to the totalizer dials.

The meter is installed as a unit in 2, 3 and 4-in. lines; for larger capacities the installation is made in a by-pass around an orifice in the main line.

	Steam Pressure Lbs per Sq In. Gauge	Rated Capacity of Saturated Steam—Lbs. per Hour (Meters with largest orifice)					
		2" Meter	3" Meter	4" Meter	6" Meter	8" Meter	10" Meter
Low Pressure Meters	0	600	1,400	2,530	5,660	9,850	15,200
	5	800	1,900	3,350	6,500	11,300	17,500
	15	1200	2,750	4,750	7,860	13,700	21,100
	30	1650	3,400	5,850	9,550	16,600	25,600
	50	2000	4,050	7,000	11,400	19,800	30,500
Standard Meters	50	2650	6,000	10,600	22,200	39,000	60,500
	100	4150	9,500	16,600	29,000	51,000	79,200
	150	5000	11,400	20,000	34,400	60,500	94,000
	200	5700	13,000	22,700	39,000	68,600	107,000
	250	6300	14,400	25,200	43,200	76,000	118,000
Extra Heavy Meters	275	6600	15,200	26,500	45,000	79,400	123,000
	300	6900	15,700	27,500	46,800	82,500	128,000

Minimum Capacities equal one-tenth tabulated quantities. Many other smaller and intermediate capacities available. Bulletin No. 255 with complete capacity tables, data sheet and price list mailed upon request.

Baldor Electric Co.

General Offices: 4370 Duncan Avenue, St. Louis, Mo.

Single Phase
PolyPhase
Direct Current

BALDOR
A BETTER MOTOR

MOTORS
1/30 to 15 Hp

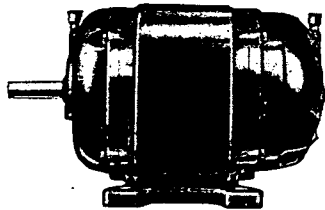
NEW YORK.....14 East 17th St.
CHICAGO.....325 W. Huron St.
DETROIT.....1034 St. Aubin
BUFFALO.....22 Kenton Rd.

BOSTON.....31 Irvington
MILWAUKEE.....3331 North 50th St.
PHILADELPHIA.....2019 Rittenhouse

PITTSBURGH.....1438 Park Blvd.
MINNEAPOLIS.....Vendome Hotel
GRAND RAPIDS, 257 Eastern Ave. S.E.
KANSAS CITY.....2004 Grand Ave.

The Baldor Electric Company has pioneered and developed many of the important improvements in motor design for heating, ventilating and air conditioning service.

For 15 years Baldor has specialized in designing and manufacturing quality built motors exclusively and the phrase "A Better Motor" is not merely an advertising slogan but a definite manufacturing policy, rigidly adhered to. Every Baldor motor is specially engineered for the particular service intended and each is fully guaranteed.



VARIABLE SPEED MOTOR For Fans and Blowers

5-speed polyphase motors for blower ventilating and cooling systems. Speeds are regulated by simple, durable, well-engineered controller. At high speed, the motor delivers full load. The lower speeds will be approximately 50, 65, 80 and 90 per cent of full load speed.

FEATURES OF BALDOR MOTORS

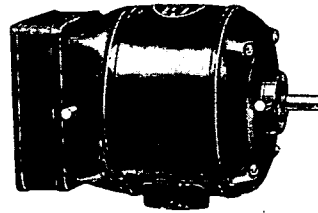
Each motor dynamically balanced.
Sine wave slot construction.
Interchangeable frames.
Solid, cushion base or ring mounting.
Wool packed sleeve bearings.
High grade steel ball bearings.

The BALDOR Line Includes:

Single Phase—Repulsion Induction.....	1/8 to 7 1/2 Hp
Capacitor Type.....	1/30 to 1 1/2 Hp
Split Phase.....	1/30 to 1/2 Hp
Direct Current.....	1/8 to 3 Hp
Polyphase—Squirrel Cage.....	1/8 to 15 Hp
Double Squirrel Cage.....	3 to 15 Hp
Unit Heater Motors—All Types.....	1/30 to 1/2 Hp

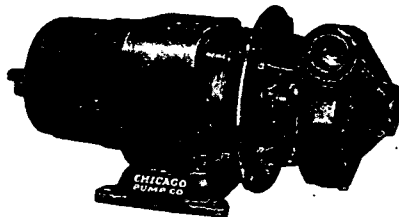
Standard Horizontal and Vertical Motors for all applications. Ball Bearings in all sizes. Sleeve bearing up to 2 Hp.

Bulletins giving complete information and full technical data sent upon request.



UNIT HEATER MOTOR

A complete line of types and sizes for all unit heater applications. Single phase, split phase and D.C. Constant, 2-speed and 4-speed quiet, vibrationless operation. 1 shot lubrication. All with inter-changeable frames. Solid, cushion base or ring mounting.



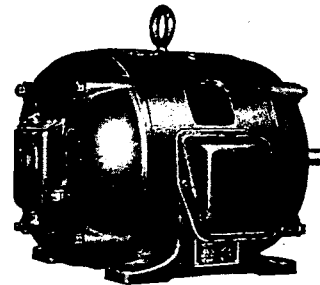
Many of the famous air conditioned trains are equipped with Baldor Motors. Oversize commutator and dual brushes totally enclosed in dust-proof housing.

Baldor Motors meet N.E.M.A. standards for starting currents, efficiency and power factor.
Approved by central stations.
No radio interference.

Century Electric Company

1806 Pine Street, St. Louis, Mo.

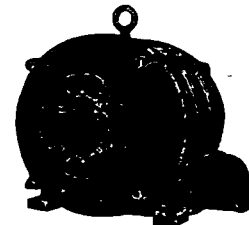
Offices and Stock Points in Principal Cities



Century Multispeed Squirrel Cage Motors

1/8 TO 200 HORSE POWER

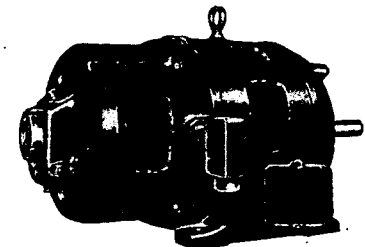
Especially adapted to meet the variable or constant speed requirements of Fans, Ventilators, Refrigerators and similar apparatus where adjustable speed change is a requirement Built for 2, 3, 4 or more speeds—automatic, push button or manual control Wide or narrow speed ranges, such as 1800/1200 or 1800/600 down to 900/450 r.p.m. (60 cycle) Conventional open or splash proof type Special speed combinations are also available Ball Bearings or phosphor bronze sleeve bearings.



Century Squirrel Cage Induction 3-Phase Motors

1/8 TO 600 HORSE POWER

Specially suited for all heavy duty, general purpose application—Stokers, Pumps, Compressors and Similar Equipment. Extremely rigid in construction with liberal mechanical strength and the ability to withstand the shock, pounding and stress of gear, chain or belt drive Refined cast-iron frames and end brackets Armature commercially indestructible Available in standard or totally enclosed fan cooled types Ball bearings or phosphor bronze sleeve bearing.



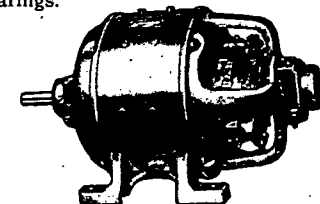
Century Slip Ring Induction Polyphase Motors

1/4 TO 250 HORSE POWER

Desirable for installations where high-starting torque is required and low-starting current is desirable or where heavy inertia loads increase the time required to bring the load up to full-speed.

This type of motor is adapted to two classes of service—constant speed or adjustable varying speed. The only difference between the two installations is the secondary control equipment.

Ball bearings or phosphor bronze sleeve bearings.



Century Direct Current Motors

1/8 TO 150 HORSE POWER

Built with shunt or compound windings, for all commercial voltages Open, semi-enclosed, fully enclosed ratings Phosphor bronze sleeve bearings with ring oilers, or grease lubricated ball bearings; also vertical ball bearing types Constant, variable and adjustable-speed types The commutator, the most important part of a Direct Current Motor, is made up of hard drawn copper bars, insulated with amber mica between the bars and mica V rings between the bars and the steel shell used to clamp them together:

ALSO

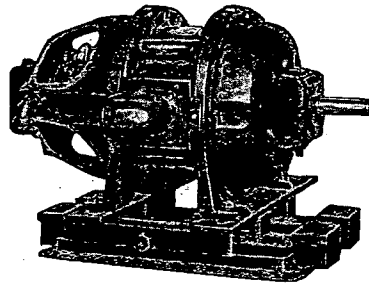
Blower Motors.
Unit Heater Motors.
Cushion Mounted Motors.
Refrigeration Motors.
Vertical Motors.

GENERAL ELECTRIC COMPANY SCHENECTADY, N. Y.

SALES OFFICES, WAREHOUSES, SERVICE SHOPS AND DISTRIBUTORS IN PRINCIPAL CITIES
For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc.,
address Merchandise Department, Bridgeport, Conn.

HEATING, VENTILATING AND AIR CONDITIONING MOTORS

The complete line of motors manufactured by the General Electric Company offers you a motor with electrical and mechanical characteristics best adapted to each of your compressor, fan or pump application.



MB Wound rotor quiet operating induction motor on sound isolating base

SOME G-E MOTORS AND THEIR USES

Application	Speed	Type Winding	Type	Horsepower Range	Classification
Fans and Centrifugal Pumps	Constant or Adjustable	Shunt	B & CD*	1/8-200	Direct Current
Reciprocating Pumps and Compressors		Compound	B & CD*	1/8-200	
Small Direct Connected Fans	Constant	Resistance Split Phase	KH*	1/40-1/3	Single Phase Alternating Current
		Reactance Split Phase	KX	1/6-1/3	
Belted Fans, Centrifugal Pumps	Constant or 3-Speed	Low Torque Capacitor	KC*	1/50-10	
	Constant or 2-Speed	High Torque	KC*	1/4-10	
		Capacitor	KC*	1/8-10	Polyphase Alternating Current
		Repulsion Induction	SCR	1/8-10	
Pumps, Compressors, Fans	Constant or Multi-speed	Squirrel Cage	K or KB*	1/4-1000	
		(Low Starting Current)	KF	7 1/2-75	
Reciprocating Pumps and Compressors		(High Starting Torque)	KG	3-100	
Pumps, Compressors, Fans	Constant or Adjustable	Wound Rotor	M & MB*	1/2-1000	
	Constant	Synchronous	TS	25-2000	

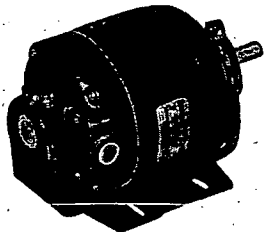
*Quiet Motor.

This list covers only the principal motors used in buildings, for other types and modifications (Vertical, Enclosed, etc.) refer to nearest sales office.

G-E "Quiet Motors" are specially designed and tested for this service and carry a name plate so stamped. The G-E sound insulating bases are applied for each definite rating and are designed to eliminate critical speed vibrations for each definite rating. These products are the result of extensive research in the problem of noise elimination.

When writing specifications, ask for motors "designed and tested for quiet operation."

This Company will gladly assist in the solution of any electrical problem in relation to heating and ventilation



KH split-phase induction fractional horsepower motor

GENERAL ELECTRIC COMPANY SCHENECTADY, N. Y.

SALES OFFICES, WAREHOUSES, SERVICE SHOPS AND DISTRIBUTORS IN PRINCIPAL CITIES
For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc.,
address Merchandise Department, Bridgeport, Conn.

CONTROL FOR HEATING, VENTILATING AND AIR CONDITIONING MOTORS

The right G-E control with the right motor for an application insures ideal operation.

The General Electric line of standard control offers manual or automatic equipment for compressors, fans, or pumps driven by any type motor which you require, providing full protection for your motor, especially those listed on the preceding page.

For special applications General Electric controllers can be designed to meet your exact requirements.

The following is a list of typical control equipment applicable to all motors listed on the preceding page.

Full voltage automatic starters with thermostatic control for fans, or pumps.

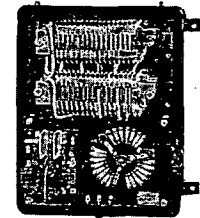
Automatic reduced or full voltage starters for synchronous motors driving compressors.

Manual or automatic speed regulating controllers for wound rotor motors driving fans.

Manual full voltage starters for pump motors.

Manual speed regulating switches for small capacitor motors driving fans.

Capacitors for improving power factor.



CR7785 Controller (cover removed) for Use with Wound Rotor Motors 1-5HP

ACCESSORIES

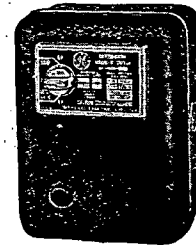
Electrically operated valves.

Thermostats.

Float switches.

Pressure switches.

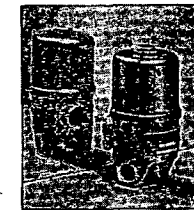
Indicating Push buttons.



CR 7006 — Full Voltage Magnetic Switch for Use with Inductor Motors



CR1061 Fractional Horsepower Motor Starting Switch for Wall Mounting



CR-9507-C Electrically Operated Valve

Motors and control of one manufacture insure perfect operation, simplifies installation and insures good service for the entire installation.

This Company will gladly assist in the solution of any electrical problem in relation to heating and ventilation

See also Pages 838 and 839

Westinghouse Electric & Manufacturing Co.

East Pittsburgh



Pennsylvania

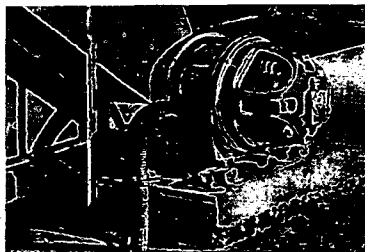
Research, Application and Design Engineers will cooperate and solve any electrical problem for heating and ventilating systems.

Sales Offices and Service Shops in all Principal Cities

Quiet Operating Motors

The Westinghouse Company has long recognized the need for quiet operating machinery and has devoted a vast amount of research work to the development of special instruments to measure noise accurately. The human element, which gives variable results, is eliminated in all testing procedure.

Westinghouse Quiet Operating Motors are specially designed and *individually*

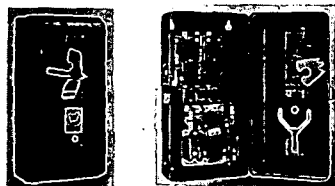


A Type CW Variable Speed, Quiet Operating Motor Driving Fan

tested under loaded and unloaded conditions in a sound-proof room. A special nameplate is your assurance of their high standard of quiet operation.

Lasting quietness is assured by rigid, unit cast motor frames which contribute to the permanence of bearing alignment and air gap. Rotors are given a special dynamic balance, practically eliminating noises due to vibration.

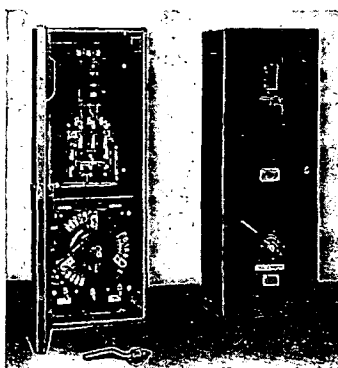
Motor Control



5 hp "De-ion" Combination Linestarter with Magnetic Contactor and Disconnect Switch in same enclosure.

Starters, speed regulators, thermostats, switches and Nofuze circuit breakers are available for every requirement.

For fan drives, the new Westinghouse Type O-1 Controller offers unusual advantages. This unit assembly includes the equipment ordinarily housed in three separate cabinets. Primary Linestarter,



Open and Closed Views of Type O-1 Fan Drive Control, Wall-mounted Type

speed regulator and Nofuze breaker for circuit disconnect and protection are all mounted and wired in a neat cabinet for wall or floor mounting. Installation is greatly simplified and its appearance is improved.

Arc Welding



FlexArc A-C Welder for Simplified Fabrication

For the fabrication of thin-gauge materials such as ducts, Westinghouse offers the FlexArc A-C welder. Its operation is extremely simple and it uses less than 15 cents worth of power an hour. One man can wheel it easily from place to place. It operates from the ordinary single-phase power circuit.

See also Page 819

Jones & Laughlin Steel Corporation

AMERICAN IRON AND STEEL WORKS

Jones & Laughlin Building, Pittsburgh, Pa.

WELDED AND SEAMLESS STEEL TUBULAR PRODUCTS

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: $\frac{1}{8}$ in. to 16 in. O.D. inclusive.

J & L Copper-bearing Steel Pipe, when specified, can be supplied in standard weight, or extra strong, black or galvanized. Use of this product is recommended for long life, where piping is to be exposed to the atmosphere or other alternate wet and dry conditions.

Jones & Laughlin Steel Pipe is made of soft, weldable steel rolled from solid ingots made to a special analysis which, checked over a period of years, has proved to be very uniform in quality. The steel pipe produced from this special grade of J & L Steel 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 and is well within the allowable tolerances as to dimensions and weights, and true to round in the outside diameter.

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 surface of the pipe is carefully cleaned so that when the galvanized coating is applied it adheres strongly and does not tend to flake off.

J & L Seamless Pipe

J & L Seamless Pipe is made in three weights; standard, extra strong and double extra strong. Sizes: $\frac{3}{8}$ in. nominal to 14 in. O.D. 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—highly developed attributes of J & L Seamless—make this product especially adaptable for air, steam, gas and gasoline lines, boilers, refineries, dry kilns, refrigerating systems and other exacting applications.

J & L Hot Rolled Seamless Steel Boiler Tubes

J & L Seamless Boiler Tubes are manufactured in accordance with the A.S.M.E. Boiler Code and comply with the A.S.T.M. Specifications and the rules and regulations of the Bureau of Navigation and Steamboat Inspection of the U. S. Department of Commerce. They are supplied in a full range of standard sizes, from 1 in. O.D. to 6 in. O.D. inclusive.

The process by which Jones & Laughlin manufactures seamless boiler tubes is largely responsible for the unusually high ductility of the product. It is a process in which a forging action is predominant. This forging action gives to the steel the greater density and higher ductility that may be expected of any forging operation. It makes it stronger yet more pliable and, therefore, more easily formed in its cold state. Forging also effectively eliminates any such imperfections as air holes or blow holes that may be present in the steel.

Inspection of this product begins with the careful selection of the steel for the billets and continues without interruption through every stage of manufacture. It is your assurance of receiving only the very finest boiler tubes that can be manufactured.

Other J & L Tubular Products

J & L also manufactures Reamed and Drifted Pipe in sizes 1 in. to 6 in. inclusive, Dry Kiln Pipe, Pipe for Refrigeration Service, Water Well and Irrigation Casing, Line Pipe and a complete line of Oil Country Tubular Products in welded and seamless.



The American Brass Company

General Offices—Waterbury, Conn.

Manufacturing Plants:

ANSONIA, CONN., TORRINGTON, CONN., WATERBURY, CONN., BUFFALO, N. Y., DETROIT, MICH., KENOSHA, WIS.

Offices and Agencies:

BOSTON, MASS., One Forty Federal St.
 STRACUSE, N. Y., 207 E. Geddes St.
 PROVIDENCE, R. I., 131 Dorrance St.
 NEW YORK, N. Y., 25 Broadway
 NEWARK, N. J., 20 Bradford Place
 PHILADELPHIA, PA., 117 South 17th St.
 WASHINGTON, D. C., 1511 K St., N.W.
 ATLANTA, GA., 10 Forsyth St.
 PITTSBURGH, PA., 535 Smithfield St.
 CLEVELAND, OHIO, 925 Euclid Ave.
 DAYTON, OHIO, 32 North Main St.
 CINCINNATI, OHIO, 101 West 4th St.
 CHICAGO, ILL., 1326 W. Washington Blvd.
 ST. LOUIS, MO., 408 Pine St.
 HOUSTON, TEXAS, 609 Fannin St.
 SAN FRANCISCO, CALIF.,
 235 Montgomery St.
 LOS ANGELES, CALIF., 411 West Fifth St.

CANADIAN PLANT: ANACONDA AMERICAN BRASS LIMITED, New Toronto, Ontario

PRODUCTS—Anaconda Deoxidized Copper Tubes and Fittings; Anaconda "85" Red-Brass Pipe; Anaconda "67" Brass Pipe; Everdur Metal for storage heaters, storage tanks, ducts and air conditioning equipment



ANACONDA DEOXIDIZED COPPER TUBES

After studying performance records, The American Brass Company endorses Anaconda Copper Tubes, connected with solder fittings, in sizes up to 2 in. where conditions of pressure and heat are normal. Copper Tubes available in long lengths and coils may be bent for ordinary turns and offsets, offering obvious installation advantages. Anaconda Copper Tubes are drawn to the accurate size required for use with solder-type fittings.

Anaconda Copper Tubes are furnished in two wall thicknesses—Type "K" and "L." Type K, the heavier, conforms to U. S. Government Spec. No. WW-T-799

and A.S.T.M. Spec. No. B-88-33. Anaconda Type K Tubes are recommended for:

1. Underground water service lines.
2. Hot and cold water lines.
3. Sprinkler systems.
4. Low pressure steam lines.
5. Oil Burner suction and return lines.

Anaconda Type L Copper Tubes meet Government and A.S.T.M. Specifications for this class and are considered suitable for interior plumbing, although the heavier grade, Type K, is preferred. Data on the comparative wall thicknesses and weights are included in the accompanying table.

Lengths and Coils—Anaconda Deoxidized Copper Water Tubes in sizes up to and including 1½ in. are furnished soft in 30, 45 and 60-ft coils; also hard or soft in 20-foot straight lengths. Sizes over 1½ in. are furnished hard or soft in straight lengths only.

Weights of Anaconda 67 Brass, 85 Red Brass and Copper Water Tubes

Nominal Size In.	Diameters—Inches						Wall Thickness Inches		Approx. Weights—Pounds per Foot					
	Copper Water Tubes						Copper Water Tubes		S.P.S. Pipe					
	Type K						Type K		67 Brass					
	Outside	Inside	Outside	Inside	Outside	Inside	Type K	Type L	67 Brass	85 Red Brass	Copper	Type K	Type L	
1/8	0.405	0.281	0.250	0.186	0.250	0.200	.0620	.0320	.0250	0.246	0.253	0.259	0.085	0.068
1/4	0.540	0.375	0.375	0.311	0.375	0.315	.0825	.0320	.0300	0.437	0.450	0.460	0.134	0.126
3/8	0.675	0.494	0.500	0.402	0.500	0.430	.0925	.0490	.0350	0.612	0.630	0.643	0.269	0.198
1/2	0.840	0.625	0.625	0.527	0.625	0.545	.1075	.0490	.0400	0.911	0.938	0.957	0.344	0.284
3/4	1.050	0.822	0.875	0.745	0.875	0.785	.1140	.0650	.0450	1.240	1.270	1.300	0.641	0.454
1	1.315	1.062	1.125	0.995	1.125	1.025	.1265	.0650	.0500	1.740	1.790	1.830	0.839	0.653
1 1/4	1.660	1.368	1.375	1.245	1.375	1.265	.1460	.0650	.0550	2.560	2.630	2.690	1.040	0.882
1 1/2	1.900	1.600	1.625	1.481	1.625	1.505	.1500	.0720	.0600	3.040	3.130	3.200	1.360	1.140
2	2.375	2.062	2.125	1.959	2.125	1.985	.1565	.0830	.0700	4.020	4.140	4.230	2.060	1.750
2 1/2	2.875	2.500	2.625	2.435	2.625	2.465	.1875	.095	.080	5.830	6.000	6.140	2.920	2.480
3	3.500	3.062	3.125	2.907	3.125	2.945	.2190	.109	.090	8.310	8.560	8.750	4.000	3.330
3 1/2	4.000	3.500	3.625	3.385	3.625	3.425	.2500	.120	.100	10.850	11.170	11.410	5.120	4.290
4	4.500	4.000	4.125	3.857	4.125	3.905	.2500	.134	.110	12.290	12.660	12.940	6.510	5.380

NOTE—S.P.S. Pipe available up to and including 10 in. Copper Water Tubes up to and including 8 in. Other seamless tubes up to 26 in. O.D.

The American Brass Company

ANACONDA COPPER TUBE FITTINGS

A complete line of Anaconda Fittings, both solder and flared tube types, is available for the assembly of copper tubes. For size range of fittings, see accompanying table. All three following types of fittings are furnished in elbows, tees, couplings and unions, including a complete range of reduction and adapter combinations. Solder fittings are checked with maximum and minimum gages to insure accuracy and the close tolerance required for dependable solder connections.

Wrought Copper Solder Fittings—Provide copper to copper connections, are uniform in quality and free from porosity. Generous depth of cup and unusually high shoulder give well made joints an extra factor of safety.

Cast Bronze Solder Fittings—Are made of "steam bronze" alloy (85 per cent copper, 5 per cent zinc, 5 per cent tin and 5 per cent lead). All cast fittings are carefully gauged and checked by air pressure under water. Test joints (¾ in.) withstood a straight pull of 7000 lb at 70 F and 1500 lb at 210 F. At all temperatures between 70 and 210 F, test joints withstood internal pressure exceeding 2000 p.s.i.

Solder—That used in the foregoing tests was 95 per cent tin—5 per cent antimony—the alloy recommended for best all-around results. Melting point is 465 F.; cooling range is only 15 F. "Hard" solder is, of course, ideal, and where unusually high temperatures are encountered, it is recommended that a high strength, high melt point solder such as Sil-Fos be used (1300 F).

Cast Bronze Flared Tube Fittings—Serving a definite need for assembling soft tubes in sprinkler systems, underground lines, etc., these fittings are also made of "steam bronze" alloy.

Anaconda Deoxidized Copper Tubes—Types "K" and "L" ½ in. to 8 in. incl.

Anaconda Wrought Copper Solder Fittings ¼ in. to 2 in. incl.

Anaconda Cast Bronze Solder Fittings ¼ in. to 10 in. incl.

Anaconda Cast Bronze Solder Valves: Gate Valves ½ in. to 2 in. incl.

Globe and Stop and Drain Valves ½ in. to 1 in. incl.

Anaconda Cast Bronze Flared Tube Fittings ½ in. to 2 in. incl.

ANACONDA "85" RED-BRASS PIPE

Anaconda "85" Red-Brass Pipe is offered as the highest quality corrosion-resistant pipe commercially obtainable at

a moderate price and is recommended for use in localities where highly corrosive water conditions prevail. These waters may be relatively low in hardness, high in carbonic acid gas content and low in alkalinity.

Anaconda "85" Red-Brass Pipe contains 85 per cent copper and conforms to government specifications for Grade "A" water pipe. The words "Anaconda 85 Red-Brass" are stamped in the metal at intervals of one foot throughout each length.

ANACONDA "67" BRASS PIPE

Anaconda "67" Brass Pipe can be depended upon to give lasting service in all localities where normal water conditions prevail.

Anaconda "67" Brass Pipe contains 67 per cent copper and conforms to government specifications for Grade "B" water pipe. It can be identified by the numerals "67" following the name "Anaconda" stamped in the metal every 12 in. in each length.

Anaconda Publication B-1 describes Anaconda Pipe, Tubes and Fittings in detail, and offers suggested methods of assembly. Write for it.

***Everdur Tanks—**Nearly all copper, Everdur is a special non-rust alloy which combines the strength of medium carbon steel with ready weldability. It is an ideal material for durable, rustless water tanks of every description—from domestic range boilers to giant storage heaters for hotels, laundries, hospitals, textile plants, schools or breweries.

Everdur is made in all commercial shapes, including tank plates which have physical properties as given in A.S.T.M. tentative specifications B96-34T. For additional data, and names of fabricators, address our nearest office.

Everdur for Air Conditioning Equipment—Everdur Metal has been used with marked success for fans and blowers, ducts, humidifiers (air washers) and for various cast and wrought parts of other equipment items subject to corrosive influences.

Because of its strength and welding properties, Everdur may be substituted for steel in the same gauges, fabricated by substantially the same methods, and with the same equipment.

"Everdur" is a registered trademark identifying products of The American Brass Company made from alloys of copper, silicon and other elements.

Chase Brass & Copper Co.

Incorporated

Subsidiary of Kennecott Copper Corporation



Waterbury

For data on Chase Copper Radiators see page 999

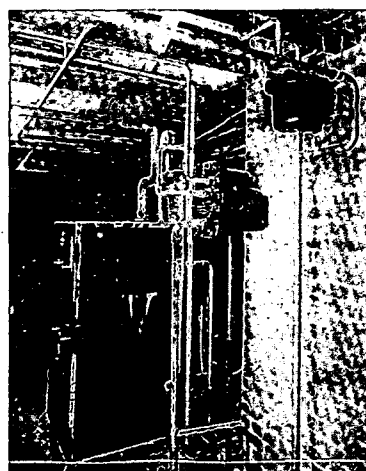
Connecticut

RUSTLESS HEATING LINES OF CHASE COPPER TUBE AND CHASE SWEAT FITTINGS

Here are a few reasons why copper tube should be used for heating lines. 1. Copper does not rust, therefore the water or steam inside the lines remains clean. 2. Absence of rust and scale formation assures permanent original steam or water carrying capacity. 3. The resistance to flow in copper tube is low. 4. Delicately adjusted thermostatic valves and traps are not rust-clogged out of commission. 5. Copper is inexpensive and as satisfactory for giving service as any material known.

TEMPER—Both hard and soft tubing are available for heating lines. In remodelling work, and in replacing old heating systems the soft copper tube will be found particularly useful. It can be worked down between walls and around corners, long 60 ft. coils eliminate many useless connections. For all soft copper tube we recommend the extra heavy gage, (known by United States Government specifications as "Type K"). For the average new installation we recommend the light gage copper tube ("Type M").

PRESSURE—Chase copper tube is adaptable for all low pressure heating systems. While we do not recommend the use of copper tube with any steam system having more than 30 lb. pressure, the factor of safety over such pressure is very great. The bursting pressure, for example, of $\frac{3}{4}$ in. Chase copper tube and sweat fittings when connected together is 3,050 lb. per square inch.



RETURN LINES—While we recommend copper tube for all heating lines, it is in the return lines that the greatest amount of rusting takes place. Architects and Engineers have learned by experience that these are the lines that are apt to rust and in some installations need replacement, after a very short period of service.

CATALOG—The information on these two pages is necessarily brief. If you would like the detailed story write for a copy of our book, "Chase Copper Tube for Heating Lines."



This is a typical vapor installation. Notice the neatness of the copper tube and sweat fitting heating lines. Also the absence of union connections.

CHASE COPPER TUBE FOR HEATING LINES



Copper tube and sweat fittings are used for the run-outs from main to risers. This minimizes resistance to flow of steam.



Steam Carrying Capacities

The steam carrying capacity of copper tube is greater than that of iron pipe of the same nominal size. For example, in a steam main, 2 in. copper tube is capable of taking care of a load of 444 sq. ft. of radiation as compared with 386 sq. ft. for 2 in. iron pipe. This is an increase of 15.1 per cent.

For equal steam pressure drop the capacity of copper tube is on the average 10 per cent greater than for the same nominal size of iron pipe. Because of this it is frequently possible to use tube of smaller size.

Pipe Covering

Copper Water Tube uses the same standard pipe coverings as other heating pipes, but in most cases requires one size smaller covering. This is a saving in the cost of covering.

Costs

The slight extra cost of Chase copper tube and sweat fittings is a very small amount in dollars to pay for the advantages of a copper tube installation.

SIZES AND WEIGHTS

Nominal Size	Outside Diameter	Inside Diameter	Wall Thickness	Pound per Linal Foot
$\frac{1}{2}$ Inch	0.625 Inch	0.569 Inch	0.028 Inch	0.203 Lbs.
$\frac{3}{4}$ "	0.875 "	0.811 "	0.032 "	0.328 "
1 "	1.125 "	1.055 "	0.035 "	0.464 "
$1\frac{1}{4}$ "	1.375 "	1.291 "	0.042 "	0.681 "
$1\frac{1}{2}$ "	1.625 "	1.527 "	0.049 "	0.940 "
2 "	2.125 "	2.009 "	0.058 "	1.46 "
$2\frac{1}{2}$ "	2.625 "	2.495 "	0.065 "	2.03 "
3 "	3.125 "	2.981 "	0.072 "	2.68 "
$3\frac{1}{2}$ "	3.625 "	3.459 "	0.083 "	3.58 "
4 "	4.125 "	3.935 "	0.095 "	4.66 "

See also Page 999

Mueller Brass Co.

Port Huron, Mich.

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ALBANY, N. Y.	DALLAS, TEXAS	HARRISBURG, PA.	NEWARK, N. J.
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Canadian Sales and Manufacturer

CANADA WIRE AND CABLE CO., LTD., Toronto, Canada

PRODUCTS—STREAMLINE Copper Pipe and Seamless Tubes; STREAMLINE Hard Copper Pipe and Fittings; Valves and Flared Fittings for Mechanical Refrigeration; STREAMLINE Refrigeration Fittings; Forgings of Brass, Bronze and Copper; Castings of Brass and Bronze; Rod; Screw Machine Products; Fabricated Parts and Special Nickel and Chromium Plated Parts.



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 Co., Division of Mueller Brass Co., Port Huron, Mich.

The Streamline 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 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 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 Co.

Port Huron, Mich.

DIVISION OF MUELLER BRASS CO.



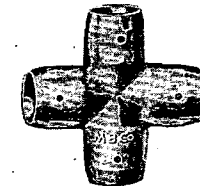
90 Deg Elbow



Tee



Tee
Copper to Inside I. P. S.



Crosses

The tapered ends, since they are the thinner sections of the fitting, hasten the cooling of the solder at these points and facilitate the completion of the joint.

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. B 88-33. For most piping purposes, hard drawn pipe is used though annealed material is supplied where bends are to be made. Three weights of Streamline Copper Pipe, Govt. Types K, L and M, are made in all sizes, and an additional lighter weight is made in sizes 3 in. and larger. The latter is used mainly by the paper industry for pressures not over 125 lb.

This range of weights permits its use for water or air pressures up to 400 lb.

Streamline Fittings are furnished in sizes from 1/4 in. to 12 in. inclusive with a full range of reducing sizes.

Fittings above 6 in. are flanged and may be had with either A.S.A or rivetted pipe standard flanges. Mating flanges are soldered to the pipe.

The Streamline Fitting permits the use of thin-walled copper pipe and places a non-rusting, non-clogging piping system within the reach of the ordinary investor. Vibration is not localized at the joints, but is harmlessly dissipated throughout the system. Copper Pipe has the property of transferring the heated element (steam or hot water) from the point of generation (boiler) to the radiators quickly and with slight temperature drop.

During the last five 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, clog 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.

Republic Steel Corporation

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TONCAN COPPER MOLYBDENUM IRON

What Is Toncan Iron?

Toncan Iron is a highly refined open hearth iron with which is alloyed the correct proportion of copper and molybdenum. As such, it possesses the maximum rust-resistance of any ferrous material in its price class.

Thousands of rigid tests and the performance of untold tons of Toncan Iron in actual service point to the greater economy in the heating and ventilating systems of which the sheet metal and pipe are Toncan Iron.

Advantages of Toncan Iron

(1) It resists to a higher degree more of the many and varied types of corrosion than any other ferrous material in its price class. This resistance is not confined to the surface or "skin" of the metal, but is uniform throughout.

(2) It combines the high rust-resistance of an alloy iron with many of the desirable physical qualities of less resistant ferrous materials.

(3) It is one of the most workable of materials. Sheets form easily. Pipe may be handled like any iron.

(4) Unlike other ferrous materials, cold working—cutting, bending, punching, threading, etc.—in no way affects the rust-resistance of Toncan Iron.

(5) It welds easily by any of the usually accepted modern methods. The use of Toncan Iron welding rod insures an installation of equal rust- and corrosion-resistance throughout.

(6) A uniform and tightly adherent galvanized coating can be applied, thus adding the protection of a heavy coating of zinc to the already high rust-resistance of the base metal itself.

(7) Through its longer, trouble-free life, it has been found to cost far less per year of service. Its use is more than an economy. It is insurance against early sheet and pipe failures and costly replacements.



Toncan Iron Sheets

Toncan Iron is available in various sheet forms. Plain sheets may be had black or galvanized, in gauges No. 8 to 28, widths from 24 to 50 in., and lengths from 10 to 13 ft., depending upon gauge and width. The Toncan Iron trade mark is stenciled in two or three places on every sheet.

Toncan Iron Pipe

Toncan Iron Pipe is available in standard and extra heavy weights; black or galvanized; in sizes from 1/4-inch to 16-inch O.D. All Toncan Iron Pipe, 2-inch and larger, is electric resistance welded, and combines the foregoing advantages of Toncan Iron with the advantages of Republic's electric welding process—100 per cent weld, perfect roundness, uniform diameter and wall thickness, and freedom from scale. Toncan Iron Pipe, "black" finish, is painted blue; galvanized finish is marked with two blue stripes. Couplings are stamped TM.

Source of Supply

Toncan Iron Sheets and Pipe are stocked by jobbers in all large cities; leading contractors everywhere use Toncan Iron and are glad to supply it where specified. If, for any reason, you cannot obtain Toncan Iron, write to Republic's nearest sales office.

Literature

Two 64-page books, "The Path to Permanence" on Toncan Iron Sheets and "Pipe for Permanence" on Toncan Iron Pipe, will bring you the complete story. Write for your copies.

Other Republic Products

Republic Steel Corporation manufactures hundreds of iron, steel and alloy products, among which of interest to heating and ventilating engineers are Enduro Stainless Steel in sheets and other usual forms, steel pipe, and steel or Toncan Iron boiler tubes.

Revere Copper and Brass Incorporated

Executive Office: 230 Park Avenue, New York City

MILLS—BALTIMORE, MD., TAUNTON, MASS., NEW BEDFORD, MASS.
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REVERE COPPER WATER TUBE

Revere Copper Water Tube is recommended for heating lines, refrigerant lines, heat control lines and for other heating and ventilating piping. This tube is seamless, 99.9 per cent pure copper, completely deoxidized, with a gun-barrel finish inside.

It is furnished in three types known as "K," "L," and "M," which meet Government and A.S.T.M. specifications. In general the Type "K" is used where corrosive conditions are severe, and types "L" and "M" where these conditions are normal. For most heating work, Types "L" and "M" are satisfactory.

Types "K" and "L" are furnished in both hard and soft tempers; Type "M" in hard temper only. The hard temper is used for new and exposed work; the soft temper for hidden replacement work and where flexibility is essential.

Fittings—This tube is joined with Streamline soldered fittings or with any standard make of compression fittings. Thus, threading is eliminated and metal in S.P.S. pipe used only for cutting threads is saved. The wall thickness of Revere Copper Water Tube is uniform throughout and the joints are leak-proof, vibration proof and stronger than the tube itself.

The Streamline fitting is so made that friction is reduced to a minimum and partial stoppage due to the lodgment of sediment, etc., is prevented.

Advantages of Revere Copper Water Tube—In brief these are as follows:

1. Revere Copper Water Tube has many installation advantages which make for economical first cost, as well as maintained economy. With Streamline Fittings it can be installed in a minimum of space. The soft temper tube can be bent where needed, as for example, around spandrel beams. This saves installation time and eliminates fittings. Furring is greatly reduced. Joints may be easily disassembled.

2. Revere Copper Water Tube cannot rust, so that it is not necessary to install oversize pipe. Life-time service is assured for all uses. Flow is smooth and uninterrupted and turbulence and noises are greatly reduced.

3. In forced hot water systems, it has

been found practicable to install Copper Water Tube in smaller sizes than in the case where rustable pipe is used. The same holds good for some steam installations.

4. In general, pipe covering can be one size smaller with Copper Water Tube than for the same size S.P.S. pipe because of the smaller outside diameter of the tube.

Concealed Radiation—Revere manufactures two distinct types of convectors: the Robras, box fin type, and the Rocop, tubular convector, which meet almost all conditions of service, price, etc.

To Heating and Air Conditioning Manufacturers—Revere Engineers are anxious to cooperate with manufacturers of heating and air conditioning equipment and to make recommendations with reference to the use 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 Thick-ness In.	Wt. Lbs. per Ft.	O.D. in In.	Wall Thick-ness In.	Wt. Lbs. per Ft.	O.D. in In.	Wall Thick-ness In.	Wt. Lbs. per Ft.
3/8	.500	.049	.269	.035	.198	.025	.144		
1/2	.625	.049	.344	.040	.284	.028	.203		
5/8	.750	.049	.418	.042	.362	.033	.258		
3/4	.875	.065	.641	.045	.454	.032	.328		
1	1.125	.065	.839	.050	.653	.035	.464		
1 1/4	1.375	.065	1.040	.055	.882	.042	.681		
1 1/2	1.625	.072	1.360	.060	1.140	.049	.940		
2	2.125	.083	2.060	.070	1.750	.058	1.460		
2 1/2	2.625	.095	2.920	.080	2.480	.065	2.030		
3	3.125	.109	4.000	.090	3.330	.072	2.680		
3 1/2	3.625	.120	5.120	.100	4.290	.083	3.580		
4	4.125	.134	6.510	.110	5.380	.095	4.660		
5	5.125	.160	9.670	.125	7.610	.109	6.660		
6	6.125	.192	13.870	.140	10.200	.122	8.910		
8	8.125	.271	25.900	.200	19.290	.170	16.460		

Recommended Operating Pressures

Type K—Hard Temper..... up to 400 pounds
Type K—Soft Temper..... up to 250 pounds
Type L—Hard Temper..... up to 250 pounds
Type M—Hard Temper..... up to 250 pounds

Temper and Lengths

Type K | Hard Temper in straight 20 ft. lengths.
Type L | Soft Temper in 30 ft., 45 ft., and 60 ft. coils.
Type M—Hard Temper in straight 12 ft. and 20 ft. lengths.

Wolverine Tube Company

SEAMLESS COPPER, BRASS AND ALUMINUM

Main Office and Mill: 1411 Central Avenue, Detroit, Mich.

NEW YORK OFFICE: 420 Lexington Avenue

Sales Offices

ATLANTA, GA.....	411 Georgia Savings Bank Bldg.	MINNEAPOLIS, MINN.....	529 S. Seventh St.
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DEOXIDIZED COPPER WATER TUBING

All Wolverine Tubing is extruded from billets of 99.9 per cent pure copper, which accounts for the very high corrosion resistance of this tubing.

Because of the one step extrusion process, metal flow is uniform and unrestricted. The tube, therefore, has uniform density, unusual strength and a mirror-like inside finish. Wolverine Tubing also has an even temper (hard or soft) for easy workability, is free of all scale or sediment, may be frozen repeatedly without bursting and is accurate in dimensions both I.D. and O.D.

Any solder or compression type fitting may be used with Wolverine Tubing to make a perfect joint, due to the close tolerances held in tube diameters.

Wolverine Deoxidized Copper Water Tubing is made to U. S. Government WW-T-799 and A.S.T.M. B-68-33 Specifications under the regular types K, L, and M.

Government Type K, A. S. T. M. Class K

Recommended for air conditioning, refrigeration, oil burner and plumbing installations. Particularly suited to underground lines and where water conditions are severe. Also for Gas, Steam and Oil Lines and Industrial Uses.

Sizes up to 2 in. approved by Underwriters' Laboratory.

Furnished in hard or soft temper in straight 20 ft lengths, soft temper in 30, 45 and 60 ft coils.

Government Type L, A. S. T. M. Class L

For oil burner, air conditioning, refrigeration and general plumbing uses.

Suitable for normal water conditions. Furnished in hard or soft temper in straight 20 ft lengths, soft temper in 30, 45 and 60 ft coils.

Government Type M, A. S. T. M. Class M

Suitable for air conditioning and refrigeration installations and for interior plumbing and heating purposes.

Furnished in hard temper in straight 20 ft lengths only.

For use with soldered fittings only.

FABRICATED TUBING

Wolverine offers a complete and competent fabrication service for tube parts. This includes bending, flaring, swaging, brazing, etc. Every operation is held rigidly to size and specification.

DEHYDRATED REFRIGERATION TUBING

Wolverine Dehydrated Soft Copper Refrigeration Tubing is guaranteed to meet A.S.T.M. Specification B-68-33. The ends of each coil of dehydrated refrigeration tubing are solder sealed and crimped in the form of the Wolverine "W."

Wolverine Refrigeration Tubing, in securely wrapped coils, is now available. This spiral crepe paper wrapping protects the coil against atmospheric contact, dirt and other foreign matter. It keeps the tubing absolutely clean until ready for use and it gives the service man a compact package to save room in his kit.

Your local refrigeration jobber can supply you with any quantity or size of Wolverine Tubing. Large stocks carried in the mill at all times for prompt shipment.

American Society of Refrigerating Engineers

37 West 39th Street, New York, N. Y.

REFRIGERATING DATA BOOK

THE current (second) edition of the biennial *Refrigerating Data Book* constitutes a one-volume encyclopedia on refrigeration in all its applications by

thirty of the leading experts in the field. The editorial policy is to include all the fundamental information on the art of refrigeration and to include as much of the practical application data as possible. The book is thus useful for the

novice as well as the experienced practitioner. In 470 pages of text data in five parts, its 30 chapters go into the theory, the means and machinery of refrigeration, and its applications. Several chapters deal with air conditioning in theory and practice. In hard leather binding, \$3.50 in the United States. A third edition (1936-38) appears in the fall of 1936.

HOW TO FIGURE AIR CONDITIONING

A NEW popular book, called "How to Figure Air Conditioning," by Harold M. Henrickson, an able engineer and writer, assembles not only the elementary facts, but shows computations for the design and application of various types of air cooling and conditioning as actually done in the field. In cloth \$1.00, postpaid; 40 pages, with a psychrometric chart and numerous working examples, definitions and the like. Treats systems with and without refrigeration for summer conditioning. The first thing of the kind! Just published!

REFRIGERATING ENGINEERING

NOW in its 31st volume, *Refrigerating Engineering* is published in a highly popular form with fully illustrated cover and with a wide range of refrigerating

topics, treated every month by the ablest men. This magazine at the same time carries as a scientific section the proceedings of *The American Society of Refrigerating Engineers*, which have given the magazine its unquestioned authority.

The editorial policy is to foster a spirit of independent technical inquiry into all questions which arise in the refrigerating industries, to give the reader a chance to share the perspective of the experienced people who write for its pages, and to bring him also the necessary background information on each topic discussed, so he may know its connection with current affairs. The rate is \$5.00, with lower introductory rates occasionally offered.

CODES AND STANDARDS

THE A.S.R.E. has a number of technical codes in its series of *Circulars*. Recent additions include the code for *testing and rating mechanical condensing units* (No. 13, 15c) and the code for *testing and rating air conditioning equipment* (No. 14, 20c). Other current data: Plant test code, corrosion prevention code, safety code, code for testing iced refrigerators. Write for free booklet *Pointers to Authors*, including style sheet and directions for locating information in refrigeration.



American Artisan

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, Chicago, Ill.

AMERICAN ARTISAN covers the field of warm air heating, sheet metal contracting and residential air conditioning.

For more than 50 years AMERICAN ARTISAN has served the warm air heating and sheet metal contracting industry. Then, as residential air conditioning began to develop and it became apparent that its growth was to be along the lines of the central, forced warm air heating system, with duct distribution of the conditioned air, AMERICAN ARTISAN naturally expanded to include it.

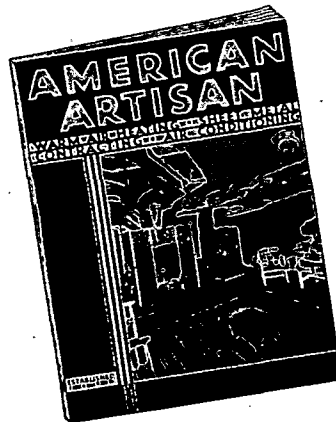
In every issue AMERICAN ARTISAN carries a special section, titled "Air Conditioning," to give specialized attention to the subject of air conditioning for homes and small structures.

Its readers are warm air heating and sheet metal contractors, jobbers and manufacturers, together with engineers who take it for its thorough air conditioning coverage of the home field.

As the demand for residential air conditioning began to develop, the advantages of the warm air type of heating system, either direct or indirect, with its duct distribution of air, became apparent.

This type of system was readily adapted to all factors of air conditioning, either through a self-contained central unit or through a central furnace to which could be added, step-by-step or as a whole, fan, washer, humidifier, filters, controls, cooling and automatic firing.

Today, as a result of this ready adaptability as well as economy, thousands of homes have winter air conditioning—forced warm air heating with air cleaning



and humidification. These homes are ready for summer cooling to be attached to the same system to give them complete year-round air conditioning.

This trend put a premium on knowledge of air handling, brought to the front in residential air conditioning the warm air heating and sheet metal contractor with his experience in "treating" air at a central place and getting it properly distributed. It presented many new engineering problems, the solutions to which were undertaken by the warm air heating industry. In short, it put warm air heating in the center of residential air conditioning.

In aiding to develop this trend, assist in the solution of the problems involved, AMERICAN ARTISAN has provided a service to its field which has made it a recognized authority on residential air conditioning practice, read by the leaders in this business.

It offers the manufacturers whose products are used in residential air conditioning work coverage of the key factors. Such manufacturers will be interested in getting a copy of "Air Conditioning—A Simplified Outline of Its Markets." This study will be sent upon request.

AMERICAN ARTISAN is published monthly. It is a member of the A.B.C. and A.B.P.

Subscription rates—\$2.00 per year; \$3.00 for two years in U. S., Canada, Mexico, Central and South America.

Foreign, \$4.00 per year.

Advertising rates furnished upon request.

Heating, Piping and Air Conditioning

Published by

KEENEY PUBLISHING COMPANY

6 North Michigan Avenue, Chicago, Ill.

HHEATING, PIPING AND AIR CONDITIONING is the publication which carries in each issue the official Journal of the American Society of Heating and Ventilating Engineers in addition to its own regular editorial section.

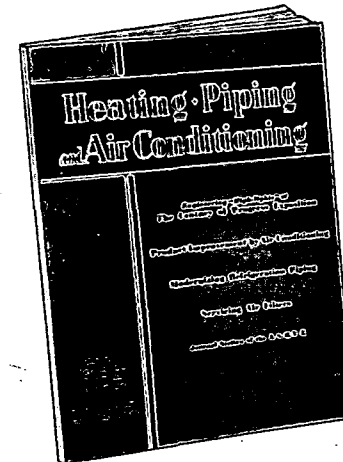
The field of this publication is that of industry and large commercial and public buildings. Its editorial policy is to give specialized attention to the design, installation, operation and maintenance of heating, piping and air conditioning systems in such plants and buildings.

It circulates among consulting engineers and architects . . . contractors . . . and engineers in charge of heating, piping and air conditioning in industrial plants, large commercial and public buildings, federal, state and city governments, school boards and public utilities.

The entire membership of the American Society of Heating and Ventilating Engineers is numbered among its subscribers, representing approximately 30 per cent of the total.

Such a coverage means, for the advertiser, consideration at all points in the selling of a heating, piping or air conditioning product . . . consideration in the selection of a product during the preparation of plans and specifications; consideration in the actual purchase of a product for installation; consideration in the year 'round buying of a product for operating and maintenance requirements.

With the rapid development of air conditioning, it has become evident that the



air conditioning field has logically divided itself into two markets: (1) Industrial and Commercial; (2) Residential.

These two markets are different in equipment used; different in engineering problems involved; different in engineering, distributing and consuming personnel . . . require, therefore, different selling jobs.

To sell the industrial and large building field for air conditioning, the manu-

facturer must win acceptance from the engineers who design, specify, install, operate and select the system to meet the particular requirements of the plant or building. The system may be central, unit or "split," but it is these engineers who are the influencing and purchasing factors.

It is to such groups that HEATING, PIPING AND AIR CONDITIONING editorially caters—exclusively in the industrial and large building field. Without waste, the manufacturer of air conditioning products and accessory equipment, such as motors, drives, controls, etc., can reach through its pages those from whom he is seeking the necessary engineering acceptance.

These facts are clearly outlined in a folder which will be sent to interested parties—"Air Conditioning—A Simplified Outline of Its Markets."

HEATING, PIPING AND AIR CONDITIONING is a member of the A.B.C. and A.B.P.

Subscription rates—\$2.00 per year; \$3.00 for two years in U. S., Canada, Mexico, Central and South America. Foreign, \$4.00 per year.

Advertising rates furnished upon request.

Automatic Heat and Air Conditioning

1900 Prairie Avenue

Chicago

DEALERS in automatic heating and air-conditioning equipment come from many industries. To reach these dealers of automatic heating and air conditioning equipment, manufacturers may choose of two ways. One is the long way around and is far more expensive. The other is the direct way and, of course, the economical way.

Choosing the first way the manufacturer must use space in the various publications, each reaching particular segments of the automatic heating and air conditioning industry—an expenditure in each publication.

Choosing the second way the manufacturer may use space in *Automatic Heat and Air Conditioning*, the publication designed to blanket the entire industry, the direct and economical way.

Your message in the pages of *Automatic Heat and Air Conditioning* reaches those dealers who are actively engaged in selling automatic heating and air conditioning equipment and specialties. This includes specialty dealers primarily interested in electric appliances, including radios, washing machines, etc., heating dealers, coal dealers, electrical contractors, oil dealers, refrigerator dealers, hardware dealers, automobile and accessory dealers, etc., who have taken on the sale of automatic heating and air conditioning equipment.

Whether or not you obtain your share of business from the automatic heating and air conditioning field depends upon



whether or not you have proper access to desirable, potential dealers. The economical and effective way of doing this is by placing your story in *Automatic Heat and Air Conditioning*.

Business analysis of the readers of *Automatic Heat and Air Conditioning* based on December Issue 1934:

Automatic Heat and Air Conditioning Dealers and Engineers.....8552

While some of these dealers were formed primarily to sell oil burners, room coolers or some other type of automatic heat and air conditioning, the big majority were primarily selling some related line such as steam and hot water heating, warm air heating, electric appliances, refrigerators, etc. Less than 24 per cent of the total comes from any single one of these groups. They are in reality specialty dealers handling more than one line.

Consulting Engineers.....	53
Manufacturers and representatives of manufacturers of automatic heat and air conditioning equipment	1,324
Utilities.....	48
Inspectors.....	54
Advertisers.....	50
Advertising Agencies.....	338
Office, file and men.....	100
Total Circulation.....	10,519

Domestic Engineering

1900 Prairie Avenue

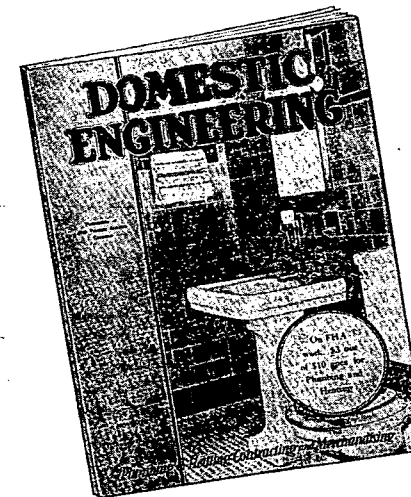
Chicago

DOMESTIC ENGINEERING covers the contracting and merchandising phases of the plumbing and steam, vapor, vacuum and hot water heating, oil burner, gas burner, stoker and air conditioning field. It concentrates primarily on the technical and mechanical developments, merchandising ideas, office and shop methods and news.

The men on the staff of *Domestic Engineering* have a total of over 200 years of experience in the plumbing and heating field. They have been in at the birth of new products and the inception of marketing plans for them; they have sat in at marketing councils where their advice was effective in securing new sales volume.

Experience gives a background to *Domestic Engineering* for which there is no substitute. It shows itself in such obvious ways as files packed with marketing data and in the adequate satisfaction of reader needs.

Since its inception in 1889, *Domestic Engineering's* first consideration—and last—has always been for its readers. Its staff has spared no effort in its endeavor to help these readers; to keep them informed on all of the latest developments in their industry; to provide a wealth of technical information direct from the pens of the leading authorities in the field; to solve for them the many difficult problems which arise in their work and to supply them with all of the important news of their industry, while it is news.



In addition, *Domestic Engineering* maintains a Readers' Service Department. As its title implies, it is the function of this department to furnish accurate, reliable information, at no cost to its readers, on various subjects pertaining to their particular business. Through this department the readers' problems are analyzed and clarified. Engineers, thoroughly familiar

with the field and its problems, offer readers dependable answers to all questions of a technical nature. Merchandising experts, with years of practical training to their credit, assist these men in formulating merchandising plans, preparing mailing pieces and direct mail letters. From this department they may also obtain reliable buying information. Names and addresses of manufacturers of equipment used in this field are available on short notice. Product information, too, may be had by the reader. Time and again this service has worked to the mutual advantage of both dealer and manufacturer; the dealer is put in touch with a trustworthy product and the manufacturer secures a reliable dealer.

The leadership of *Domestic Engineering* in its field, earned by its service to its readers and by its long established, aggressive editorial policy, has been responsible for attracting as its readers the active business builders in the industry. This same leadership has attracted to its pages the advertising messages of many of the leading manufacturers in the industry.

FUEL OIL Journal

Published Monthly at
420 Madison Avenue
New York

As early as 1931 over 35 per cent of oilburner installation contracts included orders for additional heating accessories. In 1935 almost 23 per cent of the more than 154,000 conversion burners sold were installed with *new boilers*.

An oilheating dealer thinks of an automatic heating system as a good oilburner, using good fueloil, firing a good heating system. All three are essential. The oilheating dealer needs information on all three subjects—oilburners, fueloil, heating systems. That's how FUELOIL JOURNAL is published. Every issue is soundly balanced against the dealer's working needs for information on all three subjects—heating, burners, fueloil. It is thus balanced editorially, with its circulation:

Power oilburner dealers and distributors.....	9,739
Key heating contractors, plumbing and heating contractors and engineers.....	2,108
Fueloil distributors, selling fueloil and range oil.....	3,177
Accessory and heating and plumbing supply distributors.....	691
Total Dealers and Distributors.....	15,715

Collateral Services Available to Advertisers

Key Market Studies.
Oilheating and Airconditioning Merchandising News.
Annual Survey of the Oilheating Industry.
Specific Product Surveys.
Oilburner and Fueloil Specification Chart.



Power oilburner manufacturers and their executives.....	459
Accessory manufacturers.....	332
Rangeburner manufacturers.....	173
Total Manufacturers.....	964
Total Dealers, Manufacturers	16,679
97 per cent of total circulation	
Architects, research laboratories and libraries.....	117
Advertisers, agencies and complimentary.....	248
Miscellaneous.....	139
	504
Grand Total.....	17,183

The oilheating market is hitting new high levels in both unit and dollar volume. Heating equipment advertisers long have found FUELOIL JOURNAL's advertising a profitable sales investment. It is worth your investigating.

HEATING & VENTILATING

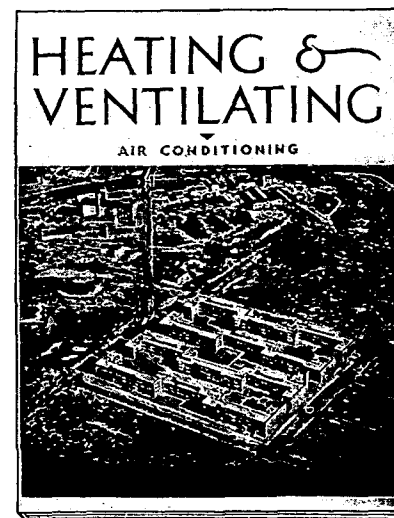
AIR CONDITIONING

THE INDUSTRIAL PRESS...Publishers
140-148 Lafayette St. New York, N. Y.

H EATING AND VENTILATING reaches the "key men" of the industry—the engineers, contractors and manufacturers who have the final word in the specification, installation, production and maintenance of the mechanical equipment utilized in the heating, ventilating and air conditioning fields.

An editorial program of outstanding alertness and authority is directed by qualified heating and ventilating engineers. A section devoted to modernization, inaugurated in 1934, typifies the forward-looking policy that has characterized the publication since its inception in 1904. News, trends, developments, personalities—every side of this important industry is faithfully and authoritatively reported in this outstanding publication.

There is a regular section devoted to new equipment, profusely illustrated and comprehensively reported. Degree-days and unit fuel consumption for various large cities in the country has been a regular monthly feature of the publication for seven years. The weather in large cities in typical localities of the country is accurately charted. Two pages of refer-



ence data appear in every issue. Reports of meetings, the activities of manufacturers, abstracts of current papers, books and pamphlets, and editorials on the planning, installation and operation of heating, ventilating and air conditioning systems in public buildings, offices, factories, schools, hospitals and homes are other contents of continuous interest.

Air conditioning, now coming into its own, has had a champion in HEATING AND VENTILATING since 1904 when, in its very first issue, an article on this then infant industry appeared. Since that time, for more than thirty years, HEATING AND VENTILATING has consistently published the news and developments of air conditioning up to its present high state of perfection and its pages have carried an impressive total of editorial lineage on this subject.

Subscriptions to HEATING AND VENTILATING are \$2.00 a year. Advertising rate cards, sample copies and market data will be gladly submitted on receipt of application.

Air Conditioning
WITH FLUID FUELS

Combined with

Oil Heat

Published by

HEATING JOURNALS, INC.

167 Madison Avenue

New York, N. Y.

TO the manufacturer wishing to reach the air-conditioning field in all its phases, OIL HEAT, with which recently has been combined AIR CONDITIONING with Fluid Fuels, offers great opportunities as an advertising medium. Its editorial content, initially devoted to the oil-burner industry alone, has broadened with the wider interests of its readers, presenting a useful and inspirational content suited to the wider field of operation.

AIR CONDITIONING's editors are widely known in the field for their fairness and knowledge and their unselfish service. They have made the editorial content valuable to the manufacturer, dealer, salesman and service man, by combining technical articles with merchandising information.

As a member of Controlled Circulation Audit, AIR CONDITIONING presents bi-annual statements covering its circulation. The last statement shows its distribution to be:

Dealers,*	12,082
Manufacturers, burner	429
Manufacturers, air-conditioning and accessory	395
Manufacturing executives	166
Branch offices	60
Miscellaneous	32

In addition, AIR CONDITIONING is sent on a rotational basis to 6,000 selected prospective dealers as follows:

Heating contractors	2,000
Electrical contractors	1,000
Fuel dealers	2,000

Specialty dealers and public utilities..... 1,000

Each name on this rotational list is selected on the basis of (a) location in known market, (b) credit rating, (c) organization and facilities for selling and installing air-conditioning equipment.

*A recent individual survey of readers shows the following pertinent facts: Of the 12,082 rated as dealers, 42% are handling winter air-conditioning equipment; 36% summer air-conditioning equipment and 30% complete air-conditioning equipment, while 69% handle boilers, 62% furnaces and 24% stokers. 100% handle oil-burners. These percentages, of course, are overlapping.

The total distribution is 16,500 copies per month.

Advertisers get excellent results due to this complete coverage, keen reader interest and responsiveness of the readers. Advertising is low cost—and produces direct business while building good will.

"We have been decidedly pleased with the results obtained since we started to use your publication this year. Your cooperative attitude also has been appreciated very much."

O. A. REITER,
Hotstream Heater Company,
Cleveland, Ohio.

AUGUST 9, 1935.

(More than 50 inquiries from a quarter-page ad in one issue.)

Plumbing and Heating Trade Journal

Published by

PLUMBERS TRADE JOURNAL PUB. CO.

515 Madison Ave., New York City

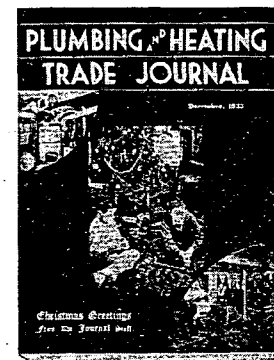
PLUMBING and Heating Trade Journal is edited to furnish a well-rounded, efficient service to the men engaged in the plumbing, heating, ventilating and air conditioning fields. To this end, it covers both the technical and business phases of their work, as well as many minor but exceedingly important ones.

It gives free technical service through a staff of practical engineers; expert merchandising assistance, and its technical and business articles are by men of recognized competence.

Thousands of readers come to THE JOURNAL each year for solutions to their technical problems and while some of the questions and answers are published in the Readers' Technical Service section in each issue of the magazine, the vast majority of them—having to do with practically every phase of heating, ventilating and air conditioning—as well as plumbing—are answered by mail, because most of the requests for help are urgent and a delay in answering would, in some cases, entail actual monetary loss to the contractor.

The Readers' Technical Service Department of THE JOURNAL is staffed by editors who have spent their lives in the business; men who were successful plumbing, heating, ventilating and air conditioning engineers before they wrote a line for publication, and who now devote their entire time to keeping abreast of the field's technical developments and using their knowledge and experience for the benefit of JOURNAL subscribers.

The technical service rendered its readers by THE JOURNAL is closely paralleled by what it strives to do for them in a business way, for it also publishes



many authoritative articles on and answers questions concerning the various ramifications of business management and prints as part of every issue, a special section devoted to selling.

One associate editor spends his entire time in the field writing articles on the business management problems of JOURNAL readers, and the practical solution of those problems.

Supplementing the business and technical articles

is a large amount of exclusive, staff-gathered news that high-lights the background of the trade's activities.

This news background is vital. It completes the industrial picture for the reader. It keeps him in intimate touch with what the various important associations and his fellow members of the craft are doing throughout the nation and it charts the trends that are likely to have a very definite influence on the future operation of his business.

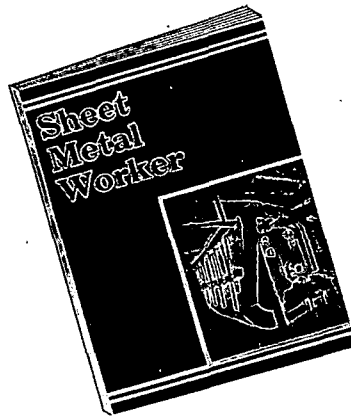
THE JOURNAL editorial department draws its news from over a hundred trained correspondents located at strategic points throughout the country—by far the largest group of exclusively editorial workers used by any paper in the industry.

It is this combination of the technical, business and news aspects of the industry that enables THE JOURNAL to achieve a finely balanced magazine that gives the reader the type of information he wants and needs, in brief, compact, time-saving form.

THE JOURNAL is a member of the A. B. C., costs \$2.00 per year by subscription and has the largest paid circulation in the heating field.

Sheet Metal Worker

45 West 45th Street
New York



SHEET METAL WORKER, established 1874—the oldest publication in the field—is published for the owners of shops in the sheet metal, warm-air heating and air-conditioning field. These key men do or control a high percentage of the buying. Their shops may generally be divided into three classifications: sheet metal shops, heating shops, and combination shops. The exclusive heating or sheet metal shops are located in the larger cities; elsewhere combination shops will be found doing a general sheet metal and furnace heating business so that they may operate profitably the year 'round.

A large, exclusive sheet metal shop does a wide variety of sheet metal work for commercial, industrial, institutional, Federal and factory buildings, such as ventilation and air-conditioning work, roofing, skylights, blow pipe including dust arresting and air handling systems, etc.

The exclusive heating contractor, handles residential, commercial, industrial, church, school and factory heating—gravity warm-air furnace installation and forced-air systems equipped with fans, filters, automatic humidifiers, air-washers and various types of controls. They also sell automatic stokers and oil burners. Essentially every furnace dealer is also a sheet metal contractor as he owns and uses machinery and tools for making ducts, fittings, etc., for heating systems. The combination shop, which predominates in the field, handles all kinds of sheet metal and heating work.

Editorially, **SHEET METAL WORKER** is divided into two sections; the first half devoted to sheet metal and general problems of the industry; the second part entitled, Modernized Heating, Air Conditioning and Merchandising, dealing exclusively with these subjects. Its editorial scope is shown by its annual index which reveals an almost endless variety of data, listed under fifty major subject classifica-

tions. Daily correspondence, however, reflects an even wider conception of the needs of the industry, the members of which, by the thousands, have called on the editors to solve their problems.

In addition to being the longest established publication in the field, its editorial and advertising staff are likewise seniors, having been identified with it from 15 to 26 years. Its editor-in-chief, Edwin

A. Scott, is regarded as an outstanding authority on trade practice and merchandising methods, and has participated in furthering many of the outstanding movements of the industry. The Edwin A. Scott Publishing Co. is the largest publisher in the world of books devoted to sheet metal, heating and allied subjects.

SHEET METAL WORKER enjoys unusual reader interest and the confidence, of thousands of substantial dealers and contractors throughout the country because of the high calibre of its editorial content. Its prestige and leadership is also recognized by prominent manufacturers in all lines, who use its advertising pages—many of them exclusively in this field—with profitable results. Two of its present advertisers have used space in it continuously for over 60 years and others over half a century.

Because of long experience and close contact with the sheet metal and warm-air heating industry, the **SHEET METAL WORKER** organization is well equipped to consult with manufacturers regarding sales, merchandising and advertising programs. A cordial invitation, without obligation, is extended to all manufacturers to take advantage of this valuable service.

Annual subscription rates—\$2.00 per year. United States and Mexico; in Canada, \$2.50; Foreign, \$3.00.

Advertising Rates Furnished on Request.

American Steam Pump Company

Plant and General Offices: Battle Creek, Michigan

DIRECT FACTORY BRANCH:
17 BATTERY PLACE, NEW YORK CITY



SALES AND SERVICE AGENCIES
THROUGHOUT THE WORLD

PRODUCTS AND SERVICE—This Company, organized 1873, offers a complete line of centrifugal, steam and power driven pumps. Your inquiry for descriptive bulletins or specific recommendation is invited.

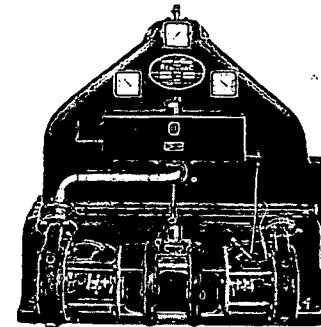
REDI-VAC HEATING PUMPS—An improved outfit. Removes air and condensation from return lines, discharging air to atmosphere and water to boiler, both functions automatically controlled. Bronze fitted throughout. No close clearances to wear rapidly, or "freeze up" during idle season—an important advantage over other types, practically eliminating periodic service expense.

RECIPROCATING VACUUM HEATING PUMPS—Time-tested design and construction. Standard equipment includes bronze fittings throughout. Motor driven, if desired. (We also build simplex and duplex steam pumps for boiler feed, etc.)

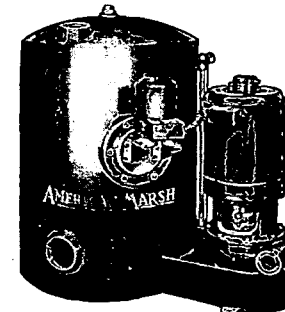
REDI-RETURN CONDENSATION UNITS—Compact, dependable and exceptionally low priced. Designed to collect and pump back to the boiler returns from high and low pressure heating systems, handling water at 210°F. Furnished complete as shown with full automatic control. Twin units can be furnished. Large range of sizes.

MOTOR-UNIT CENTRIFUGAL PUMPS—Simple and inexpensive. Ideal for air washer and cooling work, pumping clear water or brines. Pump attached directly to motor. Bearings, base and coupling eliminated. Only one stuffing box. Very compact. Fifteen sizes; 5 to over 500 g.p.m.

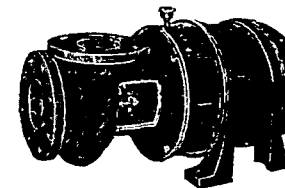
BALL BEARING CENTRIFUGAL PUMPS—Single-stage, horizontally split-case type with deep groove ball bearings, stainless steel shaft and bronze seal rings. High efficiencies insure low operating cost. (We also build multi-stage centrifugal pumps.)



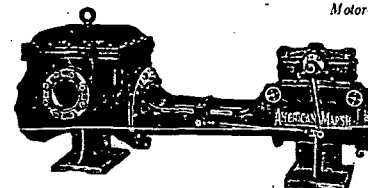
Redi-Vac Heating Pump



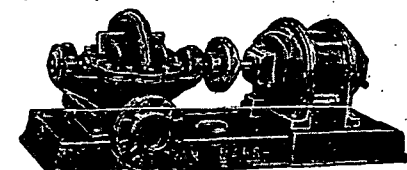
Redi-Return Condensation Unit



Motor-Unit Centrifugal Pump



Reciprocating Vacuum Heating Pump



Ball Bearing Centrifugal Pump

Chicago Pump Company

SEWAGE - CONDENSATION - CIRCULATING
BILGE - FIRE - HOUSE - VACUUM

2330 Wolfram Street

BRUNSWICK 4110

Chicago

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.

Condensation Pump and Receiver
for Low, Medium and High Pressures
Systems up to 150,000 Sq. Ft. Radiation

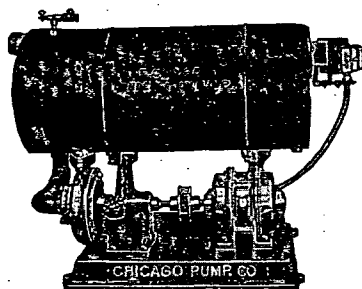


Fig. 1831—F. C. Condensation Pump

"Chicago" Condensation Pumps are built for systems ranging from 2,000 up to 150,000 sq ft of radiation, and for boiler pressures up to 200 lb. Units are built in either single or duplex—the duplex being alternated in their operation by the Automatic Alternator. For tables and complete description ask for Bulletin 129.

Vertical Condensation Pump
for Low and Medium Pressure for Systems
up to 100,000 Sq. Ft. Radiation



Fig. 1840—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 Bulletin 133.

"Sure-Return" Condensation Pump
for Low and Medium Pressure, and Systems
up to 35,000 Sq. Ft. Radiation

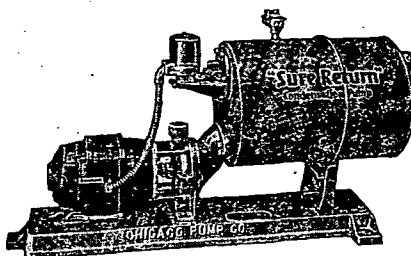


Fig. 1946

"Sure Return" Condensation Pumps and Receivers are built for systems up to 35,000 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 131.

Horizontally Split Pumps
for all Services

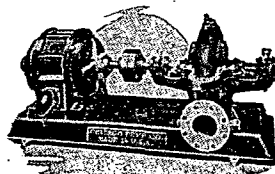


Fig. 1831—Single Stage Type "D" Pump

For any service (such as boiler feed, water supply, tank filling, circulating, fire protection, etc.). Chicago Pump Co. builds a line of horizontally split case centrifugal pumps in both single and multistages. Completely bronze fitted (except where special fittings are required) ball bearings, internal water seal, oil is filtered. "Chicago" Horizontal Pumps are built for efficient performance and long life.

"CONDO-VAC"

Return Line Vacuum Heating and Boiler Feed Pumps

Automatic Alternator is available
on Duplex Return Line Vacuum
Heating and Boiler Feed Pumps

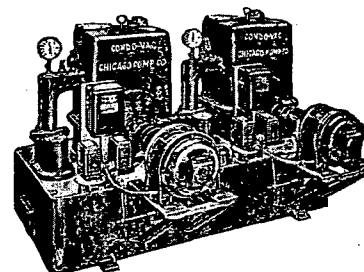


Fig. 2102—Duplex "Condo-Vacs" with Duplex Double Automatic Control

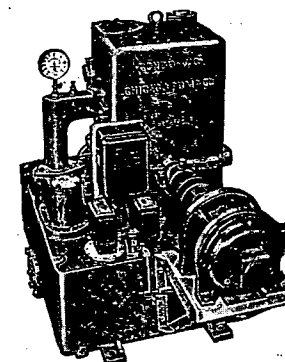


Fig. 2100—Single "Condo-Vac"

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 137, learn more about the modern vacuum pump with the long life principle of operation.

Close-Coupled Pumps

Boiler Feed, Circulating, Tank Filling,
Water Supply

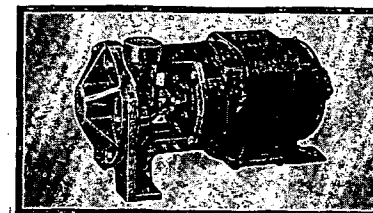


Fig. 2130—Close-Coupled, side suction pump. Capacities range from 5 to 600 G. P. M. against heads up to 189 ft. Motors from 1/8 to 20 Hp. Discharge 1 to 3 in. Both closed and open type impellers

Sewage Ejectors

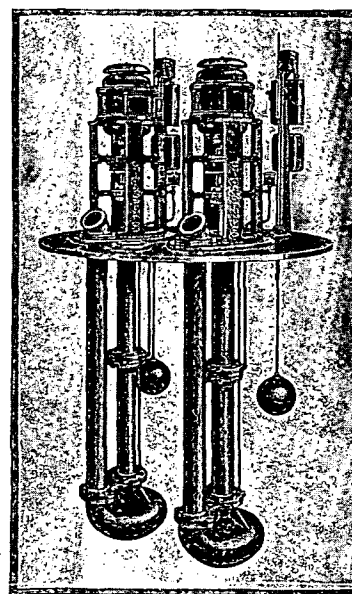


Fig. 2087—Duplex Non-Clog Sewage Ejectors with complete control equipment mounted on basin cover. "Automatic Alternator" transfers operation from one pump to the other

The Nash Engineering Company

South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities

Centrifugal Pump

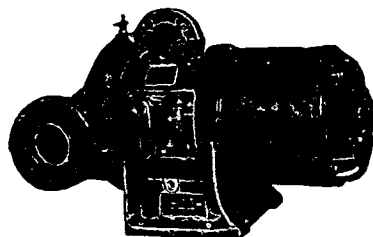
Made in standard and suction (self-priming) types. For circulating hot and cold water; boosting city water pressure; handling water in air washing and conditioning; handling ash sluicing water, etc.

Compact—motor armature and pump impeller are mounted on the same shaft. Simplified—no bearings in pump casing, one stuffing box. Accessible—impeller removable without disturbing piping or shaft alignment.

Self-priming types will handle air or gas continuously with liquid being pumped, and can be operated intermittently without foot valve.

Supplied in 1½, 1¾, 2, 3, 4, 6, and 8 in. sizes with capacity up to 2000 g.p.m. Heads up to 300 ft.

Complete data in Bulletin No. 155 on request.



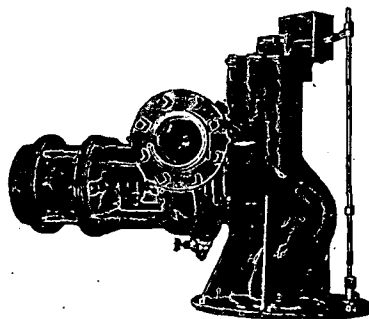
Suction Sump and Sewage Pumps

Jennings Suction Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. The Suction Sewage Pumps are equipped with a non-clog type impeller for liquids containing solids. Suction piping only is submerged. Centrifugal impeller and vacuum priming rotor are both mounted on same shaft that carries rotor of the driving motor, forming a single moving element and rotating without metallic contact.

These pumps will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit. This affords perfect accessibility for inspection or cleaning.

Capacities to meet all requirements.

Complete data in Bulletins 159, 161 and 188 on request.



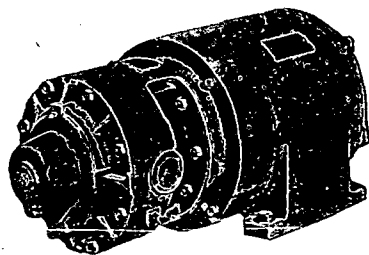
Air Compressor and Vacuum Pump

The Nash Air Compressor operates on a unique and different principle. The one moving part rotates in casing without metallic contact. There are no valves, pistons, or sliding metal vanes. There is nothing to wear, and no internal lubrication. Nash Compressors deliver absolutely clean air.

Unit illustrated is built integral with electric motor. Compact, may be installed anywhere. Ideal general service compressor. Suitable for priming pumps on water systems, handling CO₂ gas, agitation of liquids, as blood sucking pumps in hospitals, etc.

Pressure 75 lb. or vacuum 28 in. of mercury. Equipment furnished for any capacity.

Complete data in Bulletin No. 192 on request.



The Nash Engineering Company

South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities

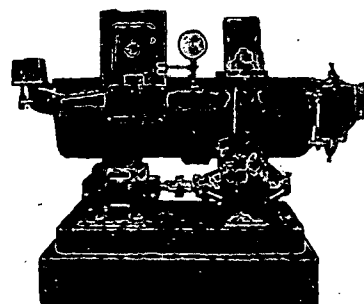
Jennings Return Line Vacuum Heating Pumps

Standard with the heating industry for over sixteen years. They remove air and condensation from the return lines of vacuum steam heating systems, discharging the air to atmosphere and returning the 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. The pump is bronze fitted throughout.

Supplied either direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation against pressures up to 40 lbs. Supplied standard in capacities up to 300,000 sq. ft. E.D.R.

Complete data in Bulletin No. 85 on request.

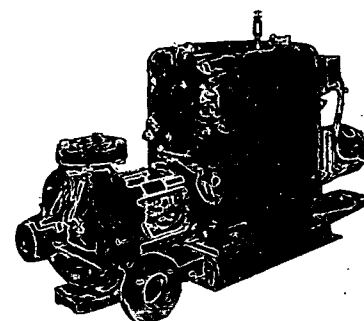


Jennings Vapor Turbine Vacuum Heating Pumps

The Jennings Vapor Turbine Heating Pump combines all of the advantages of the standard return line heating pumps 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 a 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. Over 65,000 sq. ft. and up to 150,000 sq. ft. information will be furnished upon request.

Complete data in Bulletin No. 203 on request.



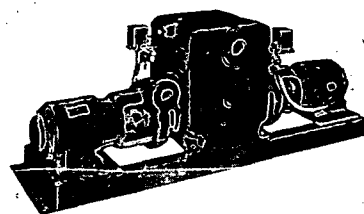
Condensation Pump and Receiver

Removes condensation from radiators in return line steam heating systems and pumps condensation back to the boiler.

They are sturdy and compact in construction, and combine receiving tank, pump and driving motor in a single assembly. Bronze fitted throughout, with Tobin bronze shaft. Impeller is of special design adapted to handling hot water with highest efficiency.

Jennings Condensation Pumps are furnished in standard sizes with capacities ranging from 1½ to 225 g.p.m. of water. For serving from 1,000 up to 150,000 sq. ft. of equivalent direct radiation.

Complete data in Bulletin No. 165 on request.





Decatur Pump Company

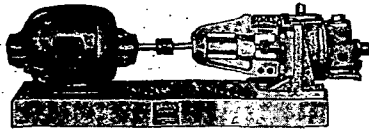
Decatur, Ill.



BURKS SUPER TURBINE PUMPS AND WATER SYSTEMS

Series 1100

Burks Super-Turbine Pumps



Fully Self-Priming.
A Heavy-Duty Pump adapted for general use with Water Systems, Condensation units, Circulating Systems, Booster Work, and will also handle effectively many other pumping requirements.

Suction lift 28 feet at normal altitudes.

BURKS CONDENSATION RETURN UNITS

Fully Self-Priming

Will Not Steambind

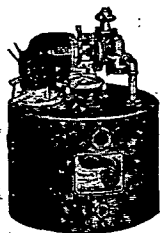
Ability to pump air alone, or air mixed with water, to the full pressure rating of the pump, means that the Burks pump will not steambind. Pump capacities range from 150 gph to 1,000 gph against boiler pressures up to 100 lbs per square inch, and will take care of up to 12,000 sq. ft., of radiation per unit.

Construction Features

Only one moving part—the impeller—and this one part does not contact with metal at any stage of operation.

Self-priming and fully automatic. Impeller and raceway of cast bronze.

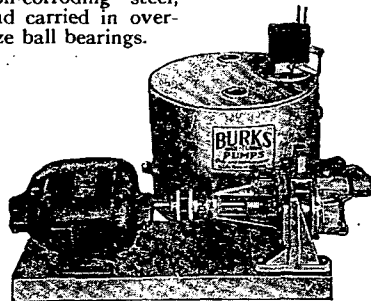
Shaft of stainless, non-corroding steel, and carried in over-size ball bearings.



Series 4700
Condensation Return Unit

Hydraulically Balanced

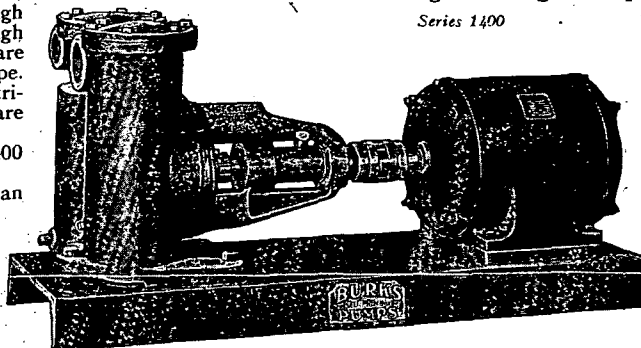
These pumps are hydraulically balanced and cannot be made to pound or hammer under any condition of operation.



Series 41100
Condensation Return Unit

Burks Self Priming Centrifugal Pumps

Series 1400



Self priming high head units have high efficiency. They are of open-impeller type. Self priming Centrifugal pumps that are different!

Capacities to 400 gpm.

Built up to an engineering ideal.



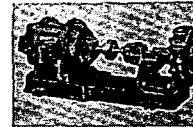
The John H. McGowan Company

General Offices: 54-58 Central Avenue

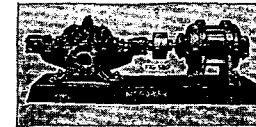
Cincinnati, Ohio

CENTRIFUGAL-STEAM AND POWER DRIVEN-TYPES

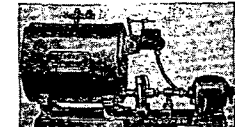
FOR WIDE RANGES OF APPLICATION



Side Suction Type



Double Suction Split Case Type



Horizontal Condensate Pump and Receiver

Centrifugal Pumps

All units embraced in the various groups of this type may be furnished for stated capacities and heads and equipped with any suitable type of drive desired.

Side Suction—Single Stage—Capacities 5 to 125 gpm for heads up to 70 ft.

Double Suction—Split Case Single Stage—30 to 3500 gpm for heads 10 to 131 ft.

Multi-Stage—Split Case Single or Double Suction according to capacity and head, 30 to 3500 gpm for heads up to 1500 ft.

Condensate Return Units—Horizontal Pump and Receiver, Motor Driven for 1000 to 100,000 sq ft radiation and return pressures from 10 to 150 lb.

Vertical Type for 3000 to 100,000 sq ft radiation and for return pressure from 10 to 50 lb.

Vertical Sump Pumps—Single and Duplex for capacities from 10 to 500 gpm and for heads up to 108 ft.

Single Steam Driven Pumps

Vacuum Pumps for radiation up to 190,000 sq ft.

Condensate Pumps and Receivers for return pressure up to 200 lb.

Boiler Feed Pumps—In Piston and End Packed Plunger Types for Boilers of 100 to 1300 hp and for pressures up to 200 lb.

Circulating Pumps—For capacities up to 350 gpm.

Duplex Steam Driven Pumps

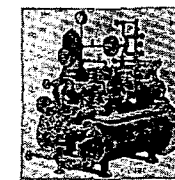
Boiler Feed Pumps—For 200 lb pressure include Packed Piston and Center Packed Plunger type for Boilers up to 3000 hp. End Packed Pot Valve Pumps 300 lb pressure patterns for boilers up to 3000 hp.



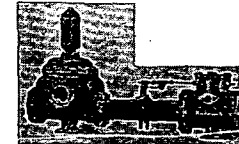
Single Steam Driven Vacuum Pump



Duplex Condensate Pump and Receiver



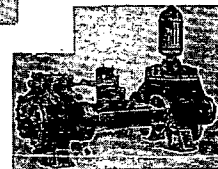
Fuel Oil Heater Sets with Duplex Pumps



Single Steam Driven Boiler Feed or Circulating Pumps

For 7000 to 95,000 sq ft radiation and for return pressure up to 200 lb.

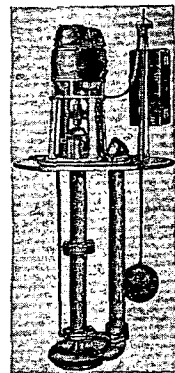
Single Sets or Duplex Sets for Boiler rating up to 9000 hp.



Duplex Boiler Feed or Circulating Pumps



Vertical Condensate Pump and Receiver



Vertical Sump Pumps

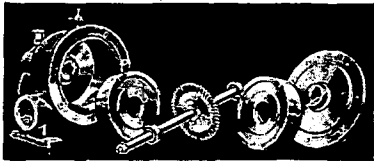
All pumps fully guaranteed for service applying to sale when properly installed

Westco Pump Corporation

6 Gaines Street, Davenport, Iowa

Branches: NEW YORK, PHILADELPHIA, CHICAGO, SAN FRANCISCO
AND LOS ANGELES. Representatives in 100 Principal Cities

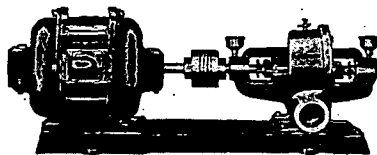
PRODUCTS—Single-Stage, Double-Suction Turbine Pumps for low or high head duty; Condensation Pumps and Receivers; Double-Suction, Self-Priming Turbine Pumps; Turbine Pumps with Gasoline Engine Drive; Automatic Water Systems; Refrigerating Pumps for Brine or Water Circulation; Underwriters' Laboratory-Approved Tank Filling Pumps for Sprinkler, Gravity and Pressure Tank Supply; Deep-Well Turbine Pumps.



Showing Pump Casings, patented Removable Liners and Westco's only moving part—the Impeller which operates without metal-to-metal contact

PATENTED REMOVABLE LINERS form the liquid channel wherein the Impeller rotates. Advantages are: 1. Life-time service from pump casings because all surfaces subject to wear or deterioration are incorporated in Liners which can, if necessary, be replaced at low cost. 2. Interchangeability of Liners permits easy, quick variation of pump capacity and head. 3. Extra Liner-Impeller sets carried on hand furnish stand-by pump protection at about $\frac{1}{4}$ usual cost.

Double-suction Intake creates perfect hydraulic balance. Capacity not affected by considerable fluctuation of head. Direct motor driven.

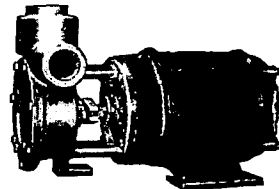


Westco Standard Type BR Pump

STANDARD WESTCOS for water supply, brine and drinking water circulation, boiler feed, etc. Furnished in cast-iron, bronze-fitted or all bronze. Sizes 5 to 400 gpm against heads up to 1000 ft.

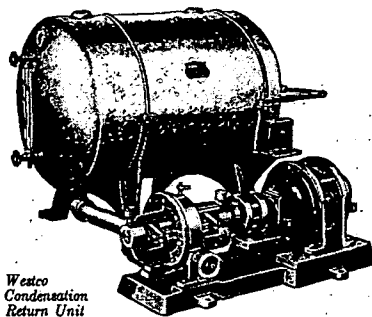
WESTCO UNI-BILT PUMPS for small capacity, high or low head duties. Especially designed for Air Conditioning and Refrigerating applications. Self-priming

models available. Sizes 5 to 90 gpm against heads up to 150 ft.



Westco Uni-Bilt Turbine Pump

Certified blueprints, specifications, catalogs containing performance and selection tables gladly sent on request. When requesting prices give: 1. Specific gravity and kind of liquid, 2. Gallons per minute, 3. Head in ft. or pressure in lbs., 4. Source of supply, 5. Power available.



Westco Condensation Return Unit

WESTCO CONDENSATION UNITS operate efficiently against 50 per cent variations (up or down) in head. Handles reasonable volumes of steam or air. Easy to clean. Extremely compact. Quiet. Sizes for service on layouts with 1000 to 50,000 sq ft radiation surface, requiring from 4 to 400 gpm against heads up to 1000 ft.

Chase Brass & Copper Co.

Incorporated

Subsidiary of Kennecott Copper Corporation

Heating Products Division



Waterbury



Connecticut

CHASE COPPER CONVECTORS

Concealed or Standard Cabinet Styles
For Hot Water or Steam Heating Systems

Advantages:

1. Made with copper tube and copper fins. Copper is the best commercially practicable material known for transferring heat.

2. The copper tubes are firmly secured to the headers by brass compression nuts, a universally accepted method of making a permanent bond not affected by strain or stress. The copper fins are corrugated to provide enlarged area for a given depth of unit and to increase strength and rigidity.

3. The copper supply tubes are $\frac{3}{4}$ in. in diameter, which insures a sufficient steam or water-way to meet all service conditions.

4. High heat output in proportion to the space required and to the weight of the radiator.

5. A copper to copper contact between fins resulting in high heat conductivity.

6. No heat resisting solder used.

7. Chase convactor ratings are *convactor manufacturers certified ratings*. They are conservative and guaranteed.

Design of Chase Convactor

The fins where they contact the tubes, have a flange or lip, which acts as a definite spacer for the fins. Part of the lips are lapped over and under each other, somewhat like shingles on a roof. These laps prevent slipping and insure a tighter bond on the tube. An intimate and permanent copper to copper bond between fins and tubes is accomplished by a patented process of forcing the tubes and the overlapped fin flanges from round to octagon shape. No solder is used.

Chase Copper Radiators are made in a number of sizes and styles to meet the varying conditions of building construction.

CHASE BRASS AND COPPER WATER HEATERS

Chase brass and copper indirect water heaters are conservatively rated. Being made entirely of rustproof brass and copper, as the name implies, they are less subject to an accumulation of water impurities and consequently last longer.

An outstanding feature of the Chase brass and copper water heater is that steam at a high pressure may be run through the coils, and water pressure up to 35 lb can be handled through the shell without difficulty.

For Complete literature on
Heating Products, write Chase
at 2 Harris St., Waterbury, Conn.

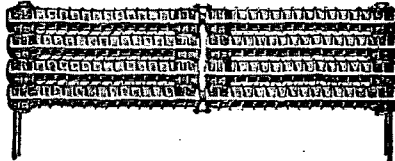
See also Pages 974 and 975

UTICA RADIATOR CORPORATION

UTICA, N.Y.

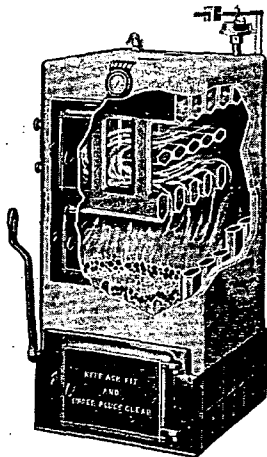
THE UTICA CAST-IRON CONVECTOR

The Utica Convector is designed for installation within enclosures and to heat effectively by convection. The cored section is ample to assure good circulation and rapid air elimination. The widely spaced fins, integrally cast, will not clog and offer minimum resistance to air flow. Utica Convectors will perform equally well on steam, vapor, vacuum and hot water systems. They are simple in construction, durable and efficient.



The Utica Convector

Made in 3½ in., 5½ in., 7½ in., and 9½ in. widths and in any length above 18 in. in multiples of 5 in.



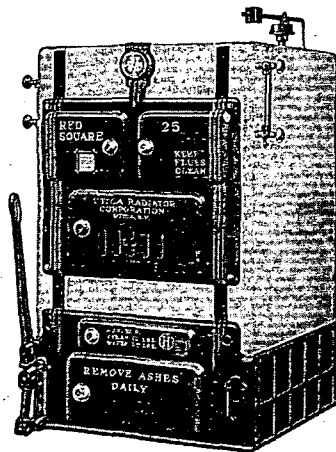
17 Series Redsquare Boiler

UTICA RED-SQUARE BOILERS are stream line in appearance and scientifically designed for efficiency and fuel economy.

The 17 Series will carry from 200 to 740 sq ft steam radiation and 330 to 1230 sq ft of water radiation.

The 25 Series will carry from 620 to 1240 sq ft of steam radiation and 990 to 1990 sq ft of water radiation.

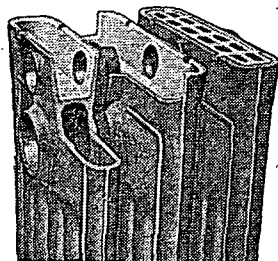
Also available in oil burning types with extended jacket.



25 Series Redsquare Boiler

Humid-Heat Radiator

The **HUMID-HEET** Radiator furnishes the required amount of radiator heat—and replenishes the air with moisture. Made in a wide range of sizes.



Cut-away view of Humid-Heat Radiator showing porcelain enameled water wells

Midget Radiators

Midget Radiators—Due to their narrow width (2-tube, 3¼ in., 3-tube, 4½ in.) and sections 1½ in. center to center, occupy one-third less space than standard tube radiators.



3-Tube Midget 4½ in. wide

Also Manufacturers of Standard Tube and Wall Radiators

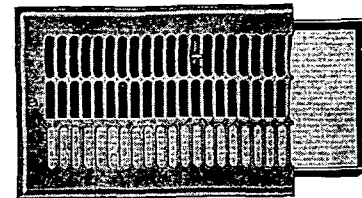
The Auer Register Co.

3608 Payne Avenue, Cleveland, Ohio

Manufacturers of Registers and Grilles for Gravity and Air Conditioning Systems; Wrought Metal Grilles for Concealing and Protecting Radiation

AIR CONDITIONING REGISTERS AND GRILLES

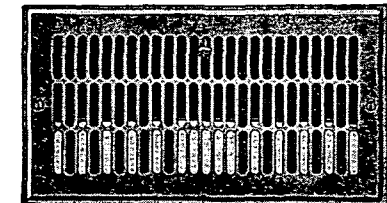
Auer Registers and Grilles for Air Conditioning have been designed to meet all modern requirements—appearance, practicability, and simplicity in installation and operation. They are made in various designs to harmonize with modern interiors. Constructed and tested to operate with high efficiency in any system of forced air, they are a departure, not merely an adaptation, from gravity registers. They are designed for easy installation in either new construction or remodelling.



No. 2030 Sidewall

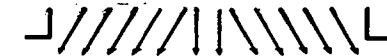
The design shown is our standard design, termed *Classic*. It is very adaptable to most interiors; other designs, however, can be furnished.

This type is made with a band frame and is installed after plastering. It is ideal for new construction or remodelling. Without a valve it is used as a return.

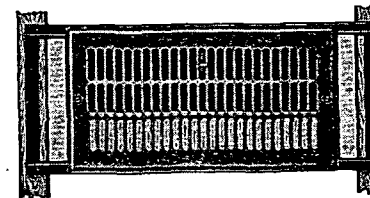


No. 2230C Sidewall

This illustration shows register directing air in three directions.



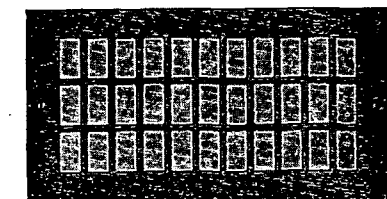
It can be made to direct air in one direction only or in two directions. The directing air blades can be set at any angle specified. Installation can be made with band frame or sliding frame as shown in previous illustrations.



No. 2010 Sidewall

This type is used in new construction exclusively. The sliding frame is anchored to studdings before plastering. After plastering, the register is easily attached to the frame and can be slid to the position desired between the studs.

The same type furnished without a valve is used as a return.



No. 2005 Oblong

Auer Registers and Grilles can be furnished in many designs.

Auer Registers and Grilles can be furnished in any standard finish. Illustrated catalogue showing complete line of registers for all purposes and chart of open areas and capacities will be forwarded on request.

Hart & Cooley Manufacturing Co.

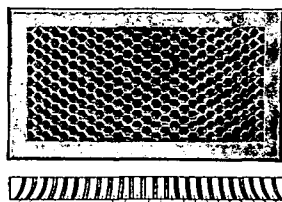
61 W. Kinzie St., Chicago

Factory: HOLLAND, MICH.

PRODUCTS

Forced Air Registers - Warm Air Registers - Directional Grilles - Stamped Grilles - Furnace and Damper Regulators - Pulleys - Chain

A COMPLETE LINE OF FORCED-AIR REGISTERS FOR AIR CONDITIONING INSTALLATIONS



No. 90 Directional Grille

No. 90 Grille is made up of a number of thin strips, which are shaped into a series of grooves. The strips, when assembled, form an exceptionally attractive grille with openings $\frac{1}{2}$ in. wide and 1 in. in depth.

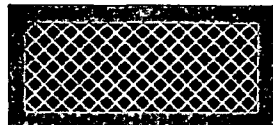
The tubes formed by the grooves in the strips may be straight, at an angle of 45 deg. to the left or right, or up and down, or in any combination thereof, thus making it possible to secure positive control of the air flow in any desired direction.

Characteristics of No. 90 Grille

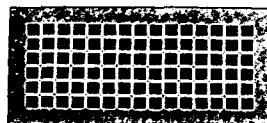
- 1. Concealment of Duct**—The depth of the grille (1 in.), together with the tubular shape of the openings, result in exceptional concealment of the duct.
- 2. Dual Control of Air Flow**—The air is controlled in two planes—horizontally as well as sideways. The air leaves the grille in a horizontal plane regardless of the approach.
- 3. Resistance**—Directional flow is obtained by changing the path of the air gradually. The turbulence which would be caused by an abrupt change in the path of the air is eliminated, thereby greatly reducing the resistance of the grille.
- 4. Noise**—The elimination of turbulence likewise eliminates the greatest cause of noise in the grille. Velocities of 2000 ft. per minute or more may be used without adding to the noise level.
- 5. Air Capacity**—The grille itself has a free area of approximately 89 per cent. When used with the 3-piece and 1-piece frames the free area is slightly reduced, owing to the clearance necessary for the frames.

Perforated Grilles

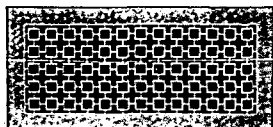
In addition to No. 90 Directional Grille, the four designs of Perforated Grilles shown below are available. These perforated grilles are easily decorated to match their surroundings and result in a satisfactory installation in those cases where directional control of the air flow is not required.



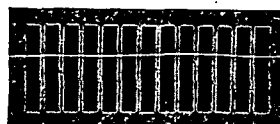
No. 30
Design
Grille



No. 50
Design
Grille



No. 40
Design
Grille

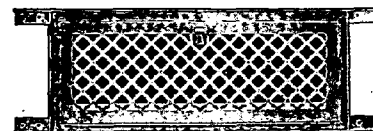


No. 60
Design
Grille

Six Types of Registers Available

All of the five grille designs shown on the preceding page are available with or without valves, and are likewise available with any of the six types of frames described below.

Sidewall Register with Streak-Proof Frame—Frame and stackhead are imbedded in plaster, resulting in streak-proof installation. Frame provides plaster lock, eliminating cracking of plaster.



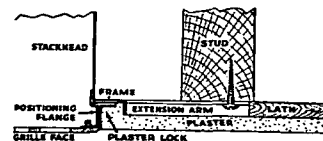
Sidewall Register with Band Iron Frame—Ideal for old house installations. Face has ample margin to cover opening in wall.

Baseboard Register with Streak-Proof Frame—Stackhead is permanently secured to frame, eliminating streaking. Face is readily removable for cleaning or redecorating.

Baseboard Register with Intergal Frame and Face—Easy to install—less expensive.

Return Air Intake— $\frac{3}{8}$ in. Projection—Designed for use where face extends above the top of the baseboard.

Return Air Intake—Flat—Designed for installations in which the baseboard height is greater than that of the intake.

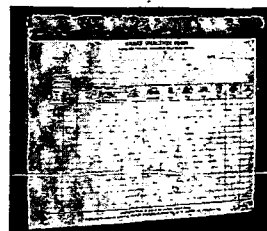
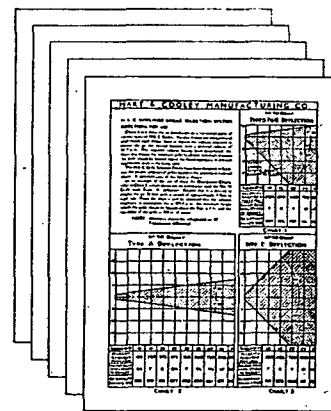


No. 3331—Sidewall Register Showing Streak-Proof Frame and Installation

H & C Simplified Grille Selection System

The H & C Simplified Grille Selection System is based on a series of charts which give complete information on air spread, velocity, required for various throws, minimum grille level required, resistance, and correct size of grille. A specimen chart is illustrated at the left.

When used with the H & C Grille Selection Forms, these charts make it a simple matter to select the proper type and size of grille, and to determine all other information pertaining to the grille which is necessary to insure proper air distribution.



The full set of charts is shown in the new H & C Air Conditioning Register and Grille Catalog now available upon request. The Grille Selection Forms are likewise available without charge, in pads of 25. For either the Catalog or Forms, apply to your jobber or direct to this company.

Tuttle & Bailey, Inc.

Main Office and Factory

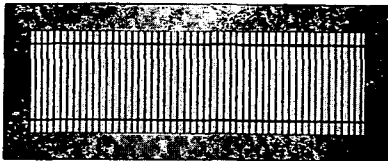
New Britain, Conn.

Branch Offices: BOSTON, NEW YORK, CHICAGO, PHILADELPHIA



PRODUCTS—Ornamental Grilles in Cast or Wrought Metals, Grilles and Registers for Heating, Ventilating and Air Conditioning, Air Control Devices, Convection Heaters. For complete information, write for Catalog No. 46 "Air Conditioning Grilles and Registers."

AIRLINE GRILLE, No. 880



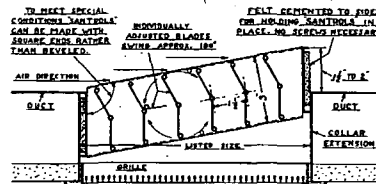
Airline Grille No. 880

Among the more popular grilles for Air Conditioning is the Airline Grille, No. 880. The core of this grille is designed to offer minimum resistance to air flow, and the construction, although rugged, presents a neat, attractive as well as sanitary appearance.

The bars on the Airline grille may be set either vertically or horizontally to provide fixed air deflections. From one to five air streams may be directed from a single grille. Standard bar settings are straight and to deflect air 22° and 42°.

SANTROLS

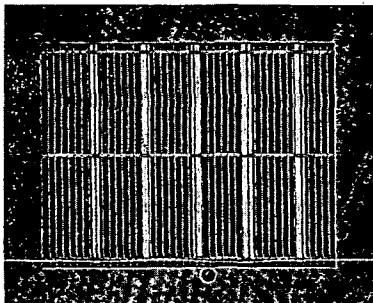
The Santrol is a new control device designed for installation back of a grille in a horizontal duct. The individually adjustable two piece blades provide both volume control and uniform distribution of air over the entire face of the grille. Blades may be adjusted by hand before installation of the grille or afterwards by means of a wire inserted between the bars of the grille core. Proper installation of the Santrol requires a four inch deep collar projecting from the side of the duct.



Santrols

VOLUME CONTROL McKNIGHT REGISTER

The McKnight register is a tested and proved device for easy control of air volume at the outlet. No dampers or diffusers are necessary on a McKnight installation as all of the control is accomplished with a key right at the register face, thus effecting a great saving of time in balancing a system. The McKnight register may be combined with an Airline design face to provide air deflection in addition to volume control.



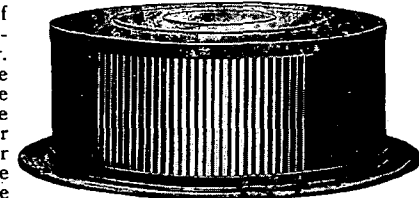
McKnight Register with Airline Face

Tuttle & Bailey, Inc.

New Britain, Conn.

CEILING OUTLETS

The use of ceiling outlets as a means of supplying air on an air conditioning installation has become increasingly popular. Tuttle & Bailey have standardized on the round outlet combined with the Airline grille, although special outlets may be furnished. Manufacturing facilities for this item now make its cost a minor factor on any installation. Ceiling outlets are furnished in the standard sizes listed in the table below. Air volumes delivered at various velocities are also given.



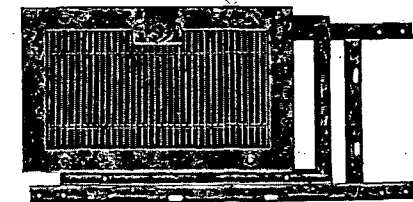
Round Ceiling Outlet

Listed Size Diameter by Height	CFM Delivered at Following Air Velocities				
	300 FPM	500 FPM	800 FPM	1000 FPM	1500 FPM
8 x 4"	181	302	482	602	905
12 x 4"	271	452	723	904	1356
12 x 6"	407	678	1085	1356	2034
18 x 4"	408	680	1088	1360	2040
18 x 6"	612	1020	1628	2040	3060
24 x 4"	540	900	1440	1800	2700
24 x 6"	810	1350	2160	2700	4050
24 x 8"	1080	1800	2880	3600	5400

AIR CONDITIONING REGISTERS
FOR SMALL HOUSE
CONSTRUCTION

Included in the Tuttle & Bailey line are many styles of registers for air conditioning in both single and multiple valve construction. There are six standard face designs all of which have been selected not only to provide beauty but also efficiency in operation.

Single valve registers are divided into two general classes—Sidewall or Baseboard, depending on the location for which they are adapted. Sidewall registers are made in three-piece, two-piece and one-piece constructions.

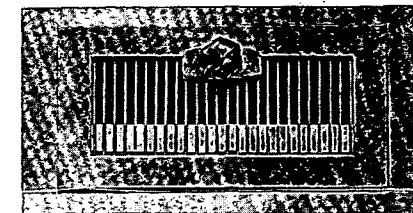


Three-piece Sidewall Register No. 888B (Streakproof)

No. 8784 BASEBOARD REGISTER
INSTALLATION

Baseboard registers are divided into two general classes—those designed to extend above the baseboard, and those designed to fit into the baseboard (see illustration).

For complete information on air conditioning registers, refer to Sec. I, Catalog No. 46.



Baseboard Register Installation

Independent Register & Mfg. Co.

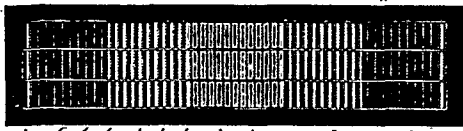
ESTABLISHED 1898

3753 East 93rd Street, Cleveland, Ohio

"FABRIKATED" FORCED AIR REGISTERS

Reg. U. S. Pat. Office

With "INDEPENDENT" Adjustable Directed Air Flow Registers and Grilles the engineer is in complete control of the direction of air flow.

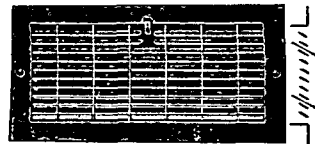


No. 381-A—Showing a combination of Adjustments



Showing the simple and easy method of adjustment. Each interior grille bar is adjusted individually.

The directional adjustment can be made at the time of installation and after the system is operating it may be easily changed at any time to make corrections to meet unforeseen or changed conditions. The method of adjustment is very simple as shown, and many directions and combinations can be worked out, to suit the need. Adjusting tools are sent with each order.



No. 311-A—Air Flow Downward Adjustable from straight to 45 deg.

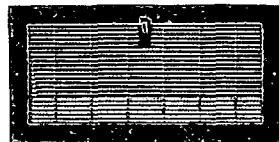
Fine Mesh "FABRIKATED" Registers and Grilles

Thinner grille bars set closer together than in the above adjustable type.



No. 322—Fine Mesh, fan like distribution

Independent "FABRIKATED" Fine Mesh Registers and Grilles can be furnished in nearly any size and with several different kinds of setting frames, and methods of mounting.



No. 312—Fine Mesh

New Independent Catalogs tell the complete story in every detail;—yours for the asking.

The Waterloo Register Company

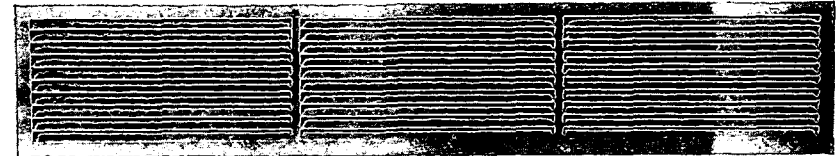
Waterloo, Iowa

Established 1902

Seattle, Wash.

Pioneers in the Manufacture of
Directional Air Conditioning Registers and Grilles

FHG-045 Venetian Type Grille

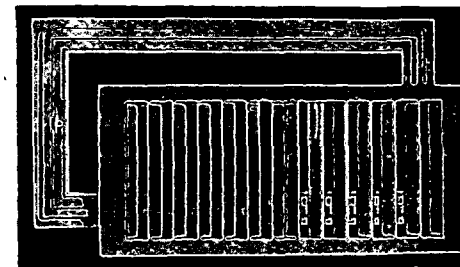


20 Gauge Louvres Anchored at 45 Degrees (55 Per Cent Open Area)

FHD-204 Venetian Type Baseboard (Register-Adjustable Louvres)



F-510½ Wall Diffuser with Vee-U Frame and Adjustable Louvres



(Waterloo) Adjustable Registers—Control direction of the air flow by operation of the louvres in sets. Five positions of directional settings are possible from full closed to full open. When operating lever is pushed back out of sight, it automatically locks the louvres.

(Waterloo) Registers—Are furnished in all finishes in standard duct opening widths of 10, 12, 14, 16, 18, 20, 22, 24, 26, 28, 30, and 36 in. and standard heights of 4, 5, 6, 8, and 10 in. Odd sizes at slight extra cost. Louvres made from 20 and 24 gauge cold rolled steel and frames are of double thickness 18 gauge. Send for Complete Catalog No. 16.

(Waterloo) Venetian Type Registers—Provide almost complete masking of duct openings, yet permit ample free area. The smooth, attractive appearance of the straight louver effect is modern and up to date.

(Waterloo) Vee-U Frame—Is a channel section of 20 gauge cold rolled steel that provides a plaster edge and flush setting for wall registers. "V" gripper is opened to admit stack head flange, then clamped tightly shut, strengthening entire stack as the frame is nailed to studding. "U" chamber allows for expansion and contraction from temperature changes and protects surrounding plaster.

WL Rod



For Adjusting Horizontal Louvres

Armstrong Machine Works

851 Maple Street

Three Rivers, Mich.

Exclusive Manufacturers of Armstrong Inverted-Bucket Steam Traps

ARMSTRONG



REPRESENTATIVES

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 Boston, Mass., Files Steam Specialty Co., 261 Franklin St.
 Buffalo, N. Y., Herr Steam Specialty Co., 360 Warwick Ave.
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 New York, N. Y., Advance Engineering Co., 69 Dey St.
 Philadelphia, Pa., Brogan & Co., 810 Race St.
 Phoenix, Ariz., John W. Ladlow, Box 1784.
 Pittsburgh, Pa., R. S. Eastman Co., 222 First Ave.
 Portland, Ore., J. C. Ross & Co., 917 S.W. Oak St.
 Richmond, Va., A. T. Shepherd, 411-12 Tenth St. Bldg.
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 South Bend, Ind., Smith-Monroe Co., 1912 S. Main St.
 Spokane, Wash., R. A. Halstead, P. O. Box 1359.
 Syracuse, N. Y., The Hopton Co., 321 Denison Bldg.
 Tampa, Fla., G. W. Neale, 504 E. LaFayette St.
 Toronto, Ont., Arthur S. Leitch Co., Ltd., 1123 Bay St.
 Vancouver, B. C., General Equipment, Ltd., 410 Homer St.
 Winnipeg, Man., Kipp-Kelly, Ltd., 68 Higgins Ave.
 Wooster, Ohio, Steam Economies Co., 1011 Beall Ave.

Armstrong traps are offered in two types for heating service, "standard" traps and "blast" traps. Standard traps are used for drip headers and unit heaters where little air is to be handled. Where large volumes of air must be removed quickly the blast trap is available.

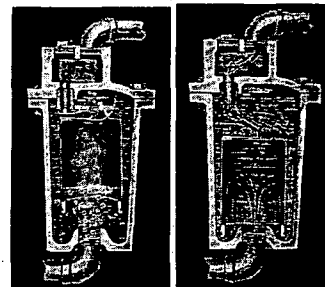
The "Blast" Type Trap—For fast handling of large quantities of air, the standard Armstrong trap can easily be furnished as a "blast" type trap by the use of a thermic bucket. The air handling capacity of this bucket is approximately 100 times as great as with the regular air vent. As long as the trap is cold a large vent in the top of this bucket remains open, and it is impossible for the bucket to float and close the valve with this vent

open. As soon as the air has been eliminated and steam comes to the trap, the heat bends the strip of thermic metal supporting a flat disc which closes the large vent. Thereafter, the trap functions in the normal manner. This thermic bucket can be supplied in any size Armstrong trap. The table on the following page gives the prices for traps so equipped.

Avoid Steam Trap Troubles—The customary troubles with steam traps are leaky valves, air-binding and plugging up with dirt or oil. The intermittent action of this trap and the metal used in the valves stop scoring and wire-drawing, the common sources of leakage. Air-binding is impossible because the air passes out of the bucket ahead of the steam through the

HOW THE ARMSTRONG STEAM TRAP WORKS

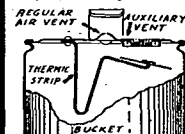
The Standard Trap



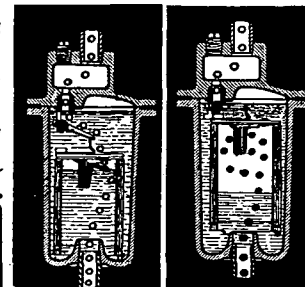
LEFT:
Standard trap (1) closed and (2) discharging.

BELOW:
Thermic bucket in blast trap.

RIGHT:
Blast trap (1) discharging air, (2) closed by steam.



The "Blast" Trap



vent at the top. When the trap is discharging, the flow of water under the bottom of the bucket prevents the accumulation of any dirt or sediment.

Simplicity—The Armstrong Steam Trap has only two moving parts—the valve lever assembly and the inverted bucket. Friction is practically eliminated in this mechanism. All wearing parts are made from nickel chrome steel except the discharge valve and seat which are made from a special chrome steel heat treated after machining to obtain maximum hardness and toughness. Many years of service without maintenance expense is the customary experience of Armstrong trap users.

Large Capacity—Discharge orifices

used in Armstrong traps are very large in proportion to the size of the pipe connections.

Armstrong trap capacity ratings are not theoretical but show actual test capacities when handling condensate at steam temperature. The effect of flash steam and pipe friction to and from the trap is thus automatically taken into consideration.

Service Organization—The satisfactory operation of all Armstrong traps is assured by 44 district representatives in the United States, Canada and Hawaii. Stocks of these traps are carried in nearly 100 leading cities. We will gladly put you in touch with your nearest representative and see that any traps you need are supplied promptly.

Sizes, Capacities and List Prices of Armstrong Traps

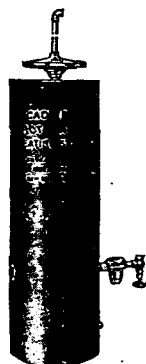
Trap Size	No. 200 and 201	No. 211	No. 212	No. 213	No. 214	No. 215	No. 216
Pipe Connections	1/2" or 3/4"	1/2"	1/2" or 3/4"	1/2" or 3/4"	1"	1" or 1 1/4"	1 1/2" or 2"
List Price (Regular)	\$7.00	\$9.25	\$15.00	\$20.75	\$29.00	\$38.00	\$55.00
List Price (Blast Trap)	\$8.50	\$10.75	\$17.00	\$22.75	\$31.50	\$40.50	\$60.00
Telegraph Code (Regular)	Acacanth Acacette Acacette	Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)	Acacanth Acacette Acacette	Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height	4"	6 3/4"	8"	10 1/4"	12 1/4"	14"	16 3/4"
Diameter	4 1/4"	4 1/4"	5"	6 3/4"	7 1/4"	8 3/4"	10 3/4"
Weight	3 1/4 Lb.	5 1/2 Lb.	10 1/2 Lb.	19 Lb.	32 Lb.	47 Lb.	76 Lb.
Maximum Pressure	125	200	200	250	250	250	250
Continuous discharge capacity in lbs. of water per hour at pressure indicated. For more complete information, see the Capacity Chart in the Armstrong Steam Trap Book.	5	450	840	1560	3000	4600	14,300
	10	560	1000	1900	3500	5600	17,200
	15	640	1080	2060	3900	6300	19,000
	20	690	890	1800	3100	5900	18,200
	25	460	940	1940	3390	6300	20,000
	30	500	970	2050	3600	6600	17,600
	40	550	780	1700	3450	5700	20,000
	50	600	840	1840	3750	6200	18,200
	60	635	900	1950	4050	6600	19,600
	70	660	940	2030	3700	6100	18,500
	80	690	800	1650	3920	6400	19,800
	90	640	840	1750	3220	6100	18,000
Lib. pressure	100	650	880	1840	3400	6300	18,500
	125	660	960	2040	3880	6700	20,400
	150	...	820	1530	3500	5900	18,600
	300	...	900	1680	3200	5400	17,400
	250	...	...	...	3500	5700	19,000

The Beaton & Cadwell Mfg. Company

Main Office and Factory: New Britain, Conn.

New York Office: 234 Water Street

CADWELL No. 45 SYSTEM FOR HOT WATER HEATING SYSTEMS



A complete unit—nothing else to buy.

A departure from the conventional type of equipment as used with tank in basement systems.

To keep the expansion pressure in the system within practical working limits even under sudden firing methods, as in oil burners.

Providing means for elastic pressure distribution within the system.

Keeping the system filled to any desired pressure, and above all—to positively protect the boiler.

All of this is achieved in a novel manner.

The combined pressure governing and relief feature—which is new:

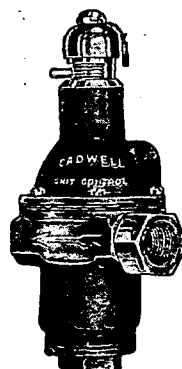
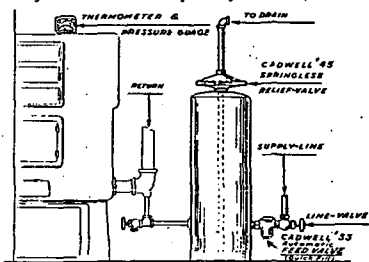
1. Has no springs.
2. Is governed entirely by physical laws.
3. It cannot increase the setting at any time.
4. It tests itself automatically as long as there is any water in system.
5. It is responsive to the slightest difference of pressure within the system.
6. Absolutely guaranteed to protect the boiler.

The filling arrangement is automatic and can be varied in pressure from 5 to 20 lb by adjusting screw.

A large strainer prevents foreign matter from entering system. The strainer, valve disc and seat can be cleaned at any time without losing water out of system.

Valve and tank supplied in Maroon color.

Layout shows simplicity of installation.



Cadwell No. 5 Control Unit

Cadwell No. 5 Control Unit

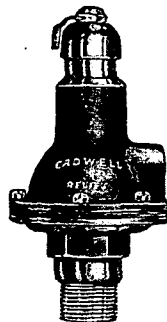
For Hot Water Heating System.

A Feed valve—Relief valve—Ball check—Screen and Back pressure valve all in one unit.

Closed by internal pressure not by heavy spring, eliminating sticking valve seats. Feeds to 15 lb, relieves at 30 lb.

Feed valve and Relief valve can be tested by pushing down on cap.

Feed valve and screen can be inspected and cleaned without drawing water out of system.

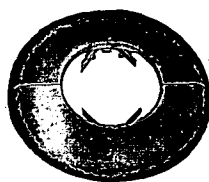


Cadwell No. 15 Relief Valve

Cadwell No. 15 Relief Valve

Built on same principle as No. 5 for relief of pressure only in Hot Water Heating systems.

Perfection Floor and Ceiling Plates



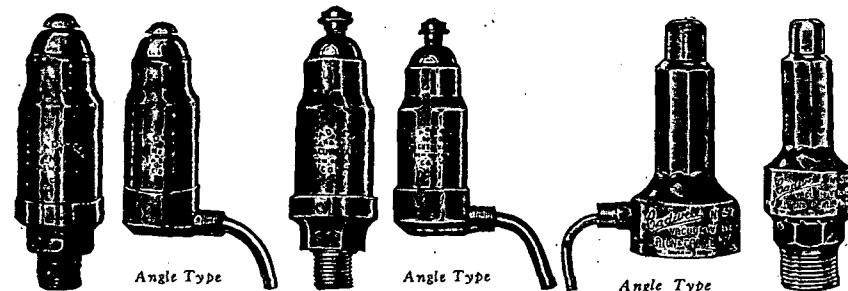
No. 10

Floor and ceiling plates to answer every requirement. Genuine Perfection No. 10, as illustrated—sheet steel or brass—sizes $\frac{1}{4}$ in. to 6 in. inclusive.

Other types available in cast iron or brass in

pipe sizes $\frac{1}{8}$ in. to 12 in. inclusive. Furnished in plain, nickel plated, chromium plated or desired finishes.

CADWELL THERMOSTATIC AIR VALVES FOR ONE PIPE STEAM AND VACUUM SYSTEMS

Bottom Outlet $\frac{1}{8}$ and $\frac{1}{4}$ -in. and $\frac{3}{8}$ x $\frac{3}{8}$ -in. sizes

No. 10 Non-Adjustable Thermostatic Valves

Bottom Outlet $\frac{1}{4}$ -in. and $\frac{3}{8}$ x $\frac{3}{8}$ -in. sizes

No. 20 Non-Adjustable Thermostatic Vacuum Valves

Angle Type

No. 50 Diaphragm Operated Thermostatic Vacuum Valves to "Equalize" One Pipe Steam Systems

No. 10—Regular, for radiators, non-vacuum, our best all metal syphon air valve.

No. 10 S. S.—Same as No. 10 except straight shank for venting return lines etc., made in the following sizes: $\frac{1}{8}$ in. I. P. S. $\frac{1}{4}$ in. I. P. S. $\frac{3}{8}$ in. Female Thread x $\frac{3}{4}$ in. Male Thread.

No. 20—Check operated vacuum air valve for radiators.

No. 20 S. S.—Same as No. 20 except straight shank for venting return lines, etc., made in $\frac{1}{4}$ in. I. P. S. and $\frac{3}{8}$ in. Female Thread x $\frac{3}{4}$ in. Male Thread.

No. 50—Diaphragm operated vacuum air valve, closes efficiently against the escape of steam or vacuum.

Large closing surface of diaphragm, through nickel silver pin forces check off seat with every closing of main valve by steam or vacuum, positively preventing the sticking of valve. Use this valve to "Equalize the System." Valve is constructed so that port vent can be increased in size to allow distant or sluggish radiators to heat, and equalized radiation results.

No. 50 S. S.—Same as No. 50 except straight shank for venting return lines, etc., made in $\frac{1}{4}$ in. I. P. S. and $\frac{3}{8}$ in. Female Thread x $\frac{3}{4}$ in. Male Thread.

RANGE BOILER OR TANK RELIEF VALVES

Cadwell No. 25 Temperature, Pressure and Vacuum Relief Valve for use on Range Boilers or Hot Water Storage Tanks



Cadwell No. 25 Relief Valve

The opening and closing of valve is governed entirely by internal pressure, not by a heavy spring forcing valve on to seat. Valve is self-closing, opens to allow the escape of temperature at about 210 deg, pressure 150 lb. Pressure setting can be supplied according to requirements of different cities. Vacuum release prevents collapse of boiler in case of break in supply line. Only one seat. Valve may be tested by pressing down on top of valve.

Cadwell No. 35 Pressure and Vacuum Relief Valve

This valve is similar to No. 25 except it does not operate by temperature.

Standard valve set to relieve pressure at 150 lb and slightest vacuum.



Cadwell No. 35 Pressure and Vacuum Relief Valve

C. A. Dunham Company

Administrative and General Offices

450 E. Ohio Street, Chicago, Ill.

Factories: MARSHALLTOWN, IOWA; MICHIGAN CITY, IND.; TORONTO, CANADA

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C. A. Dunham Co., Ltd.,
1523 Davenport Road,
TORONTO, ONT., CANADA

C. A. Dunham Co., Ltd.,
(of the United Kingdom)
Morden Road, LONDON, S.W. 19, ENGLAND

Over eighty sales offices in the United States, Canada and the United Kingdom bring Dunham Heating Service as close to you as your telephone. These representatives are available for engineering counsel in correct selection of Dunham Systems and Appliances for any type of building. The accumulated experience of the entire Dunham organization is put at the disposal of the Heating and Ventilating Engineer. This cooperation is available for *Modernization Work*, as well as for new construction in industrial, commercial and other projects.

DUNHAM DIFFERENTIAL VACUUM HEATING

The advantages of the Sub-Atmospheric principle which governs steam distribution in Dunham Differential Vacuum Heating Systems are utilized in three distinctive applications. Modifications of past standards of heating, such as the Vacuum Return Line systems and the Vapor systems, are definitely improved by relating certain Dunham equipment. The use of this equipment is explained in the Dunham Hand Book (No. 514).

- 1. The Dunham Differential Vacuum Heating System for High Duty**—This is a two-pipe system giving excellent room temperature control and winter air conditioning. The pressure, the temperature and the volume of steam in circulation are varied under a control which is a normal function of system operation. The wide range over which the vacuum and quality of steam in the radiators is regulated (from atmosphere to 25 in.) establishes correct rates of heat emission with radiators either complete or partially filled as required. A continuous valuation of heat requirements under positive temperature control may be secured through nine thermostats in a building, or zone of a building. The High Duty Differential System is a functioning part of the Dunham Average Temperature Control System.

Ventilation and Air Conditioning Application—Differential heating also gives effective service on blast heaters on ventilation and air conditioning systems. Differential control of steam temperatures and volume reduces the percentage of by-pass air and makes possible more accurate control of outlet air temperatures and humidities.

- 2. The Dunham Differential Vacuum Heating System for Low Duty**—The design of this system embraces equipment of the same general type as used in the High Duty System. Its fuel economy closely approaches that system but it does not claim the same preciseness of temperature control as characterizes the High Duty System. However, the principle of control is the same as in that system; the radiator temperatures are varied and radiators may be either completely or partially filled according to the heat loss requirements. The vacuum, however, is limited to fifteen inches. The Low Duty System can be very effectively related to existing buildings in changing

over ordinary vacuum return line systems to differential operation. One or more thermostats coupled with Dunham Heat Balancer may be used with a Low Duty System.

- 3. The Dunham One-Pipe Vacuum Heating System with Sub-Atmospheric Steam**—In this system the range of steam temperatures and pressures is ample to give great flexibility of operation in the lower range of vacuums. This system is designed primarily to enable owners of existing one-pipe systems to obtain the benefits of the Dunham Differential Vacuum principle of operation by rearranging the system to operate on that principle. The change-over can be made without extensive cutting of floors or walls. Systems having air line will usually require no such cutting.

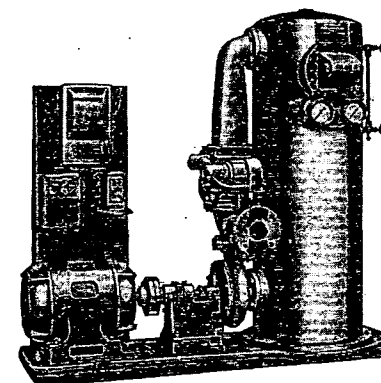
Dunham Line of Pumps for Differential Vacuum Heating Systems and for Vacuum Return Line Heating Service

The operation of these pumps is characterized by a quietness which is outstanding. Its use contributes to this essential of satisfactory heating—quietness—while supplying a vacuum producing means of great effectiveness and time proved in its ability to perform consistently over long periods of time with minimum of attention and maintenance.

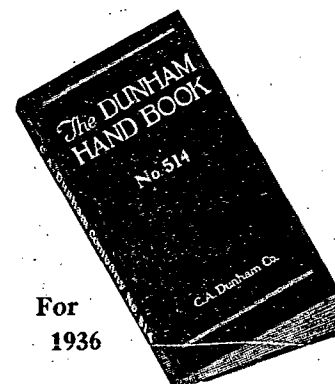
The unit combines added refinements of construction including ball bearing centrifugal pump of improved design, discharge valves operated by a powerful mechanism and other refinements.

The pumps are built in eleven sizes, ranging from capacity of 2,500 to 150,000 sq ft of radiation inclusive.

These pumps fulfill the adopted standard for the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and likewise the requirements of the Vacuum Return Line Heating Pump Manufacturer's Section of the *Hydraulic Institute*. Pumps to meet special requirements will be supplied.



"DV" Series Pump for Dunham Differential Vacuum Heating System
"VR" Series Pump for Dunham Vacuum Return Line Heating Service



For
1936

THE DUNHAM HAND BOOK

The Dunham Hand Book—No. 514. Comprehensive manual for architects and engineers dealing with the principles, designing, equipment and installation of heating systems which circulate steam under variable sub-atmospheric pressures, variable low pressures and variable volumes to give dynamically balanced temperature regulation in buildings. Contents include treatment of steam heating history, development of controlled heating, principles applied in the differential heating system, piping design and tables, equipment application with installation diagrams, ventilation, unit ventilators and air conditioning applications, unit heaters, concealed radiators, auxiliary steam service appliances, miscellaneous engineering and heating data and tables. 464 pp.

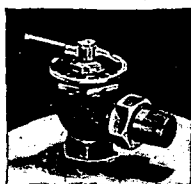
Barnes & Jones

129 Brookside Avenue, Jamaica Plain, Boston, Mass.

New York Office: 101 Park Avenue

Barnes & Jones Vapor and Vacuum Systems of Steam Heating; Modulation Valves; Packless Quick Opening Supply Valves; Metering Orifice Supply Valves; Thermostatic Radiator Traps; Thermostatic Traps for medium and high pressures; Condensators (Boiler Return Traps); Drip Traps; Heavy Duty Blast Traps; Vent Traps; Strainers; Damper Regulators; Gages; Proportionator Systems with Zone Control

Modulation Valves, Type K

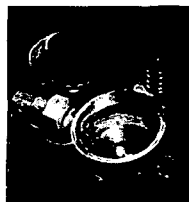


With indicating dial, non-rising stem, renewable disc seat. Tail piece extra heavy to prevent breakage, extra long to facilitate connection to radiator. Furnished with either lever or wheel handle.

Size.....	1/2 In.	3/4 In.	1 In.	1 1/4 In.
Cap. Sq Ft Rad...	30	60	100	180

Thermostatic Radiator Traps

The interchangeable control unit contains the thermostatic element, carries its own seat of special alloy and is a complete operating unit in itself. Calibrated under actual working pressure at the factory and locked in adjustment. Unit easily and quickly replaced without special tools; lift out the old unit and insert a new one. Made in low, medium, and high pressure types.



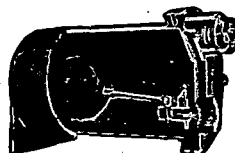
Symbol	120	12	124	134	13	14
Inlet Tapping.....	1/2"	1/2"	1/2"	3/4"	3/4"	1"
Outlet Tapping.....	1/2"	1/2"	1/2"	3/4"	3/4"	1"
Capacity, Sq Ft C. I. Rad...	200	200	400	400	700	1200

Float and Thermostatic Traps



Drip Type

For use on Unit Heaters, also drips from supply mains and risers and on returns from water heaters and indirect stacks. Float-controlled valve governs discharge of water; thermostatically-controlled valve allows passage of all air but prevents passage of steam. Made with 3/4 in., 1 in. and 1 1/4 in. tappings. Capacities 200 lb to 1200 lb of water per hour at 2 lb pressure differential.

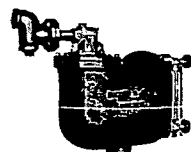


Heavy Duty Type

Combination float and thermostatic traps with an air and water capacity large enough to take care of the condensation from the largest vento stacks, dry kiln coils, hot water heaters and other units condensing large quantities of steam at low pressures. Made in sizes from 1 in. to 2 in. Capacities to 14,000 lb of water per hour at 2 lb pressure differential.

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 return side. Simple in construction, but positive in operation. All working parts are of best bronze metal.



No. 38 Condensator

No.	Capacity in Sq. Ft.
31	700
32	1,600
33	3,500
34	6,000
35	10,000
36	16,000
37	32,000

GRINNELL COMPANY INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: Providence, R. I.

National Distributors of Thermoflex Traps and Heating Specialties

For data on other Grinnell Products, see pages 919-921

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.

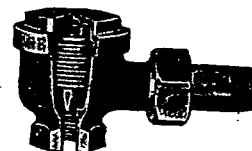
Thermoflex Traps have an exceptionally large orifice. This large orifice combined with high lift, insures fast action and freedom from clogging.

We supply Thermoflex Traps guaranteed for steam pressures up to 25 lb. and to 125 lb. Complete information and details of typical installations will be gladly sent on your request. Ask for Catalogue on Thermoflex Heating Specialties.

Valves, Traps, Gauges, Etc.

The Thermoflex line includes: Radiator Traps, Offset Traps, Blast Traps, Drip Traps, High Pressure Traps, Vent Traps, High-grade Packless Inlet Valves, and the Thermoflex Alternator, Thermoflex Compound Gauge, Thermoflex Damper Regulator.

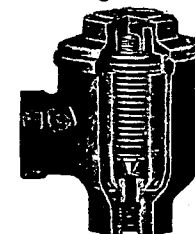
No. 12 Thermoflex Radiator Trap



The full eight-fold Thermoflex-Hydron Bellows is the best bellows ever made. Because of the Hydron-forming process every bellows is absolutely perfect. Body is heavy bronze construction throughout. Fully nickel-plated with highly polished trimmings. The No. 12 is made in angle and in corner patterns, with 1/2 in. inlet and 1/2 in. outlet tappings. The inlet neck is double thick to allow for expansion strains. Guaranteed for steam pressures up to 25 lb.

See also Pages 919-921

Thermoflex High Pressure Traps



The No. 100A Thermoflex Trap is guaranteed for steam pressures from 25-125 lb. Must not be used where the steam temperature exceeds 400 °F.

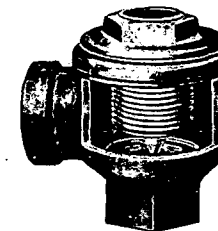
For use with all types of process work, Laundry Machinery, Kitchen Equipment, Hospital Sterilizers, Vulcanizers, Dry Kilns, Unit Heaters, Street Steam Service, etc., in fact any place that a trap is desired for service at the above pressures.

Small, compact and inexpensive in comparison to the usual float or bucket trap.

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, plain nickel finish. Can be furnished with unions, polished nickel or chromium plated at extra cost.

No. 4 Thermoflex Drip Traps



Used for dripping mains, risers, coils and unit heaters. Cast-iron body, bronze cap and inserted renewable bronze seat, angle pattern only, without unions. Can be used for any general purpose where a finished, nickel-plated trap is not necessary, and at a lower cost. Guaranteed for steam pressures up to 25 lb.

Hoffman Specialty Co., Inc.

General Sales Department

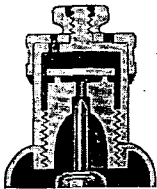
500 Fifth Avenue, New York, N. Y.

Main Office and Factory: Waterbury, Conn.

Sales Representatives in Principal Cities

Manufacturers of Adjustable Port Radiator Venting Valves, Quick Vents and Air Eliminators for One and Two Pipe Steam and Vacuum Systems; Hoffman Supply Valves, Traps and Basement Specialties for Controlled Heat Systems; Hoffman-Economy Vacuum and Condensation Pumps and the Hoffman-Tallmadge System of "Zoned Controlled Heat."

Hoffman offers a complete line of Radiator Air and Vacuum Valves and Quick Vents for every venting purpose on One or Two Pipe Steam Systems. The entire line of radiator venting valves is equipped with the Six-Speed Adjustable Orifice Venting Port illustrated at left), making it possible to balance the steam distribution in One-Pipe Steam Systems accurately, by increasing or decreasing the rate of venting of each radiator, which controls the flow of steam into that radiator.



SIPHON AIR VALVES

The Nos. 1, 70 and 71 are used for venting radiators on One or Two Pipe Steam Systems, and the Nos. 4, 5 and 75 are used in conjunction with these valves for venting steam mains, risers and other quick venting services.

OPERATION OF THE No. 1 VALVE

One of the vent ports is always open, regardless of the port adjustment. The larger ports allow a more rapid rate of venting than standard and the smaller ones a slower rate of venting and thereby control rate of steam flow into the radiators.

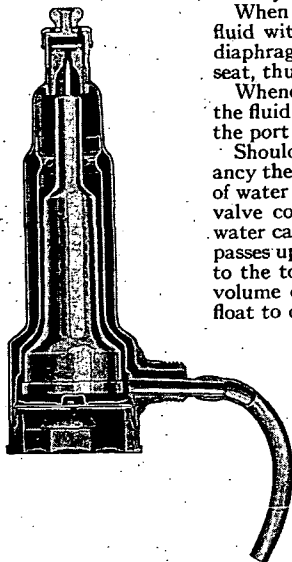
When steam reaches the float, vaporization of the heat sensitive fluid within the float expands, with "snap" action, the flexible diaphragm forming the float base and raises the valve pin to its seat, thus preventing the passing of steam.

Whenever air reaches the valve, its lower temperature reduces the fluid pressure within the float. The diaphragm contracts and the port is opened for the escape of the air.

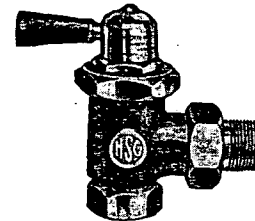
Should water surge into the valve, the float raises by its buoyancy thereby forcing the pin to its seat and preventing the escape of water through the vent port. As soon as water drops below the valve connection, air must enter the float chamber before the water can run out. Air enters through the valve connection and passes up through the air channels in the double shell construction to the top of the valve. As the air enters it displaces an equal volume of water through the siphon connection and allows the float to drop and open the vent port, allowing the air to escape.

VACUUM VALVES

The Nos. 2, 77 and 78 Vacuum Air Valves operate on a similar principle as described above, but in addition feature the Hoffman Double Air Lock consisting of the vacuum check and vacuum diaphragm. These valves are used on One Pipe Vacuum Systems; and for venting the ends of steam mains or heating risers, where it is also desired to prevent the return of air into the system, the Nos. 6, 16 or 76 Float Vacuum vents are used.



HOFFMAN CONTROLLED HEAT



No. 7 Modulating Valve

A Hoffman Controlled Heat System consists of the No. 7 Adjustable Orifice Modulating Valve on the supply end of the radiator, the No. 8-A Thermostatic Trap on the return end and either a Hoffman Differential Loop (for coal-fired installations operating at pressures up to 8 ounces), or a Boiler Return Trap where higher pressures are encountered, for returning the condensate to the boiler.

SUPPLY VALVES

Besides the No. 7 Adjustable Orifice Modulating Valve the Nos. 37 and 47 series represent a complete line of Packless Supply Valves that meet the exacting requirements of architects and engineers.

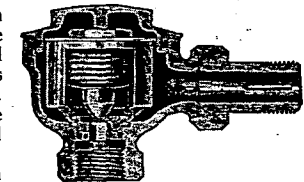
THERMOSTATIC TRAPS

The line of Bellows Type Thermostatic Traps, with hydraulically formed and tested bellows, consists of the Nos. 17-A, 18-A, 8-A and 9-A, and are principally used for low pressure steam or vapor systems. These traps have nominal capacities from 200 sq. ft. up to 600 sq. ft. of radiation. The Nos. 8-A and 9-A have renewable elements, which combine the thermostat, valve pin and renewable seat into a single unit.

The No. 10-A Hoffman Trap, which is equipped with waterhammer proof bellows, 1 in. connection, has a nominal capacity of 2800 sq. ft.

The Nos. 8 and 9 Traps have a thermostatic element consisting of three chambers each having a top and bottom diaphragm. These chambers are all joined together and the complete thermostatic member is housed in a cage and is not attached to the valve body or cap. This allows the thermostatic element and valve pin to be easily removed and replaced without adjustment. These traps range in sizes from 3/8 in. to 1 in., are medium pressure traps, and are recommended where pressures up to 50 lbs. are encountered.

The Nos. 20-A and 21-A High Pressure Traps are equipped with waterhammer proof bellows and integral strainer, for use on pressures up to 100 lbs. Available in 3/8 in. to 1 in. connection.



No. 8A—3/8 in.

DRIP AND HEAVY DUTY TRAPS

Where large amounts of condensation are encountered, it is recommended to use one of the float and thermostatic traps, which are available with or without the thermostatic element. These traps are available in large capacities and are mainly used for venting and dripping risers, steam mains, unit heaters, blast coils, etc.

VACUUM AND CONDENSATION PUMPS

The Hoffman-Economy line of Vacuum and Condensation Pumps offers a dependable method of economically returning the condensation from larger heating systems to the boiler. These pumps are made in single and duplex units, for varying capacities and pressures.

ZONED CONTROLLED HEAT

The Hoffman-Tallmadge System of Zoned Controlled Heat is a development for accurately controlling the temperatures of large buildings or portions thereof, with Precision Orifices and Zone Control Valves either manually, thermostatically or remotely controlled.

HOFFMAN SALES AND SERVICE

Hoffman Products are sold and stocked by leading wholesalers of heating and plumbing supplies everywhere. Hoffman representatives are available to assist in selection of suitable equipment for various services.

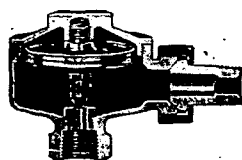
ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago



Branches and
Representatives
in Principal Cities

Illinois Thermo Radiator Traps

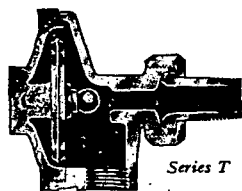


Series G

Flushes thoroughly 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. Will withstand 75 lb. steam pressure without damage although only low pressure service are required of them.

Made in three sizes from $\frac{1}{2}$ -in. to 2-in., in a variety of patterns. Furnished in either plain or nickel plated finish.

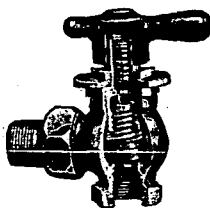
Illinois Thermo Radiator Trap



Series T

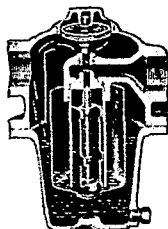
The Original Vertical Seat Trap. Self cleaning, non-adjustable. Positive and sensitive in operation. Thousands in use for over fifteen years without diaphragm replacements. Furnished in all sizes and patterns.

Illinois Modulating 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.

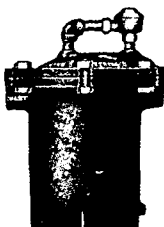
Illinois Combination F & T Traps



Series 7G

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 Combination Trap



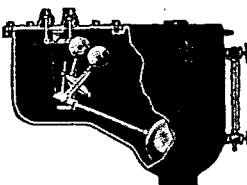
Series 3G

beyond the range of the 6G, 7G or 8G.

This is the heavy duty type similar in operation to the 6G, 7G, and 8G traps.

Series 3G traps are available in a complete range of sizes. They are equipped with separate thermostatic bypass and are furnished where specifications require this type, or where capacities are beyond the range of the 6G, 7G or 8G.

Illinois Return Trap



the air. Insures positive and complete circulation, and prevents cracked boiler sections.

Trap is self-contained, with no external working parts to be misadjusted, tampered with or injured. No stuffing boxes or packed joints, which insures continuous tightness against air or water leakage.

Write for Bulletins

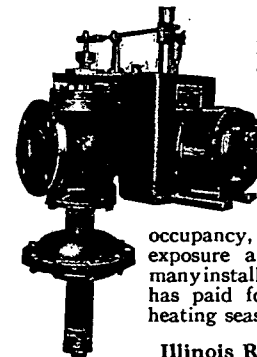
ILLINOIS ENGINEERING COMPANY

General Offices
and Factory:
Chicago



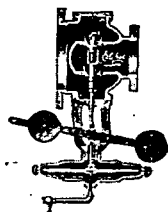
Branches and
Representatives
in Principal Cities

Illinois Thermal-Zone Control



Prevents over-heating and fuel waste in large buildings or groups of buildings heated from one central power plant. Buildings may be zoned as to occupancy, time, location, exposure and so on. In many installations this valve has paid for itself in one heating season.

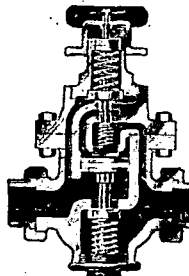
Illinois Reducing Valve



in sizes from $\frac{1}{4}$ in. to 12 in.

In general use on vacuum or low pressure heating systems. Will reduce to 4 oz. pressure from an initial pressure of 150 lb. The large diaphragm insures sensitive operation. Made in both straightway and expanded outlet bodies

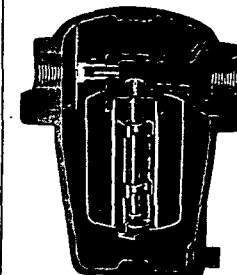
Master Type Pressure Regulator



For exacting requirements, such as tire and rubber vulcanizing, chemical process pressure control, and wherever high pressure steam must be accurately reduced in varying amount to any steady lower pressure not less than 10 lb.

It will reduce initial pressure up to 300 lb. down to any lower pressure not less than 10 lb., and does not build up pressure on a closed or dead end line. Made of bronze with monel metal valves and seats.

Illinois Steam Trap



Series 80

closed. No wire drawing or cutting of valve and seat which are of Monel metal.

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

Eclipse Spring Controlled Regulating Valve

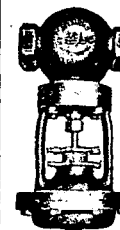


Fig. 121

Furnished in either single seated or double seated type as the service conditions require, for the control of steam, air or gas. Controlling spring is completely enclosed, protecting it from dirt and rust. Valves are furnished with the proper size diaphragm and the proper length spring to give satisfactory service under all operating conditions.

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.

Write for Bulletins

William S. Haines & Company

12th and Buttonwood Sts., Philadelphia, 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 Blast Traps, Air Eliminators, Heavy Duty Float Traps, High Pressure Traps, Boiler Return Traps and Packless Inlet Valves and Modulating Supply Valves.

All Haines Traps, whether designed for pressures below atmosphere or pressures in excess of 100 lb. per square inch, employ as their operating member a specially constructed Bourdon Tube—the principle that actuates the steam gage.

The tube is of tempered steel. It is charged with a volatile fluid and hermetically sealed. It is the expansion and contraction of the fluid, under varying temperatures, that furnishes the operating power.

The tube is mounted vertically on a horizontal valve motion. The end opposite the valve is anchored so that the travel of the tube either opens or closes the valve piece.

The thermostatic member is outboard the valve seat and closes the valve against the flow of steam. This arrangement prevents fouling of the trap due to scale or other foreign matter and permits a thorough draining of the unit to which it is attached.

THERMOSTATIC TRAPS

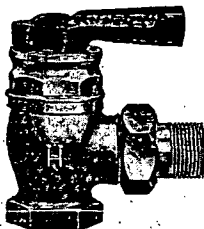
Haines Thermostatic Traps are constructed to endure, as well as to operate efficiently, in sizes ranging from $\frac{1}{2}$ to $1\frac{1}{2}$ in. They are thoroughly inspected and proved in our test laboratory under operating conditions to insure their serviceability.

MODULATING VALVES

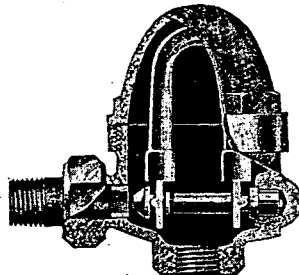
Haines Modulating Valves are permanently packed, furnished with a genuine Jenkins Bros. valve disc. Its modulating features permit varying the amount of steam admitted to the radiator.

They seat tightly and open full area on less than a complete turn. Furnished in Lever, Round Handles or Lock Shield type. The body is made of heavy brass, nickel plated with polished trimmings.

Made in sizes from $\frac{1}{2}$ to 2 in. in angle, straightway or corner patterns.



Haines Modulating Valve



Haines Thermostatic Trap

Kieley & Mueller, Inc.

Established 1879

Engineering Specialties for Pressure and Flow Control

34 West 13th Street, New York, N. Y.

Factory: NEWARK, N. J.

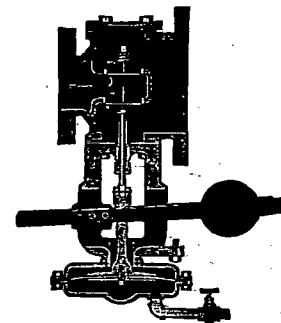
Agents in All Principal Cities

PRODUCTS—Valves: Altitude, Stop and Check, Pressure Regulating, Float, Pilot Reducing, Back Pressure, Tank Control.

Liquid Level Controllers, Water Feeders, Pump Governors, Steam Traps, Strainers.

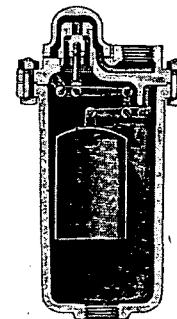
Also Damper Regulators, Hot Water Temperature Controllers, Oil Separators, Steam Separators, Return Traps, Water Columns, etc.

Catalogs sent upon request



Pressure Regulating Valve

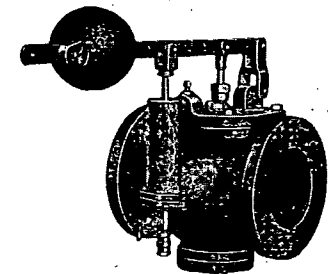
Spring and lever weighted valves for all services and for initial pressures up to 250 lb. and reduced pressures from 0 to three-quarters of the initial pressure. Single or double seated in sizes $\frac{3}{8}$ to 16 in. Suitable for steam, water, air, oil and gas. Controlled by a small feeler pipe connected from diaphragm to low pressure side.



Steam Traps

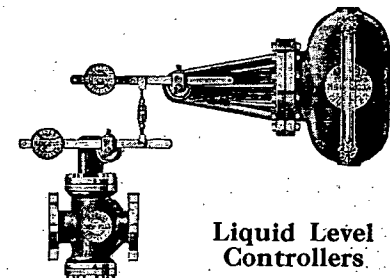
Large capacity, small sized inverted bucket traps; quick-acting, self-cleaning and non-air binding. Sizes $\frac{3}{8}$ to 2 in. Pressures up to 250 lb. Body and cover, semi-steel. Valve and seat, stainless steel. Removable cap allows inside inspection or replacement of valve parts without

disturbing pipe connections. (All parts are interchangeable).



Back Pressure and Atmospheric Relief Valve

For use where plant is operated either condensing or non-condensing. Outside air dash pot insures noiseless operation. Maintains exhaust line back pressure from 0 lb. to 25 lb. Made horizontal or vertical lever and weight or spring operated.



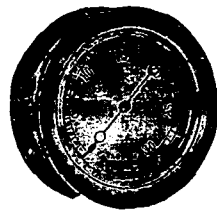
Liquid Level Controllers

For the accurate control of liquids in tanks or other vessels; suitable for use in industrial plants, gasoline plants, refineries, etc. Direct connected or remote control; ball bearing spindle and easy-to-pack stuffing box; rotary or sliding valve. Write for special bulletin C-3.

J. E. Lonergan Co.

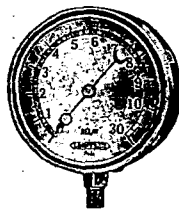
207 Florist Street, Philadelphia, Pa.

Pop Safety Valves; Relief Valves; Steam Gauges; Hydraulic Gauges; Air Gauges; Water Gauges; Pressure and Temperature Gauges; Test Gauges; Gauge Boards; Oil Gauges; Clocks; Counters; Gauge Cocks; Steam Gauge Syphons; Lubricating Specialties.



Model "GV"

Model GV—Vacuum Gauge. Gauges graduated to 30 in. vacuum. Ten sizes, $2\frac{1}{2}$ to 10 in. dial.



Model "BLGR"

Model BLGR—Gauge for indicating height of water in feet. Graduation 70 ft. Three sizes, $3\frac{1}{2}$, $4\frac{1}{2}$ and 5 in. dial.



Model "GLP"

Low pressure, Iron Body, Brass Mounted.

Set to blow off at 10, 15, 20, 25 or 30 lb.

Five sizes— $2\frac{1}{2}$ to $4\frac{1}{2}$ in.



Model "U"

Relief Valve. Snifter, Water or Cylinder—Bronze.

Recommended for steam engines, pumps, pipe lines, etc. Ten sizes, $\frac{1}{2}$ to 4 in.



Model "ORV"

Oil Relief Valve. Sizes $\frac{3}{8}$ to 2 in. For use on oil burning systems. Has large relieving capacity.



Model "VAK"

Special valve for vacuum breaking. Six sizes, $\frac{1}{2}$ to 2 in. Positive in its action.



Model "WRV"

Water Relief Valve for tank service. Sizes $\frac{3}{8}$, $\frac{1}{2}$ and $\frac{3}{4}$ in.

Pop Safety Valve A.S.M.E. "House Heating Boiler." Standard pressures, 5, 10 and 15 lb.



Model "HHD"

CATALOGUE

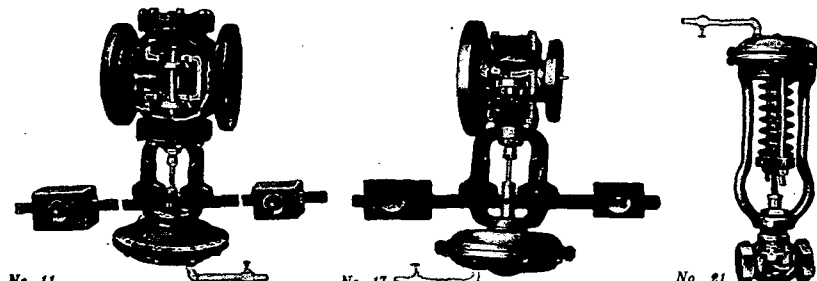
Write for our new 100-page catalogue, describing and illustrating the complete "Lonergan Line" or ask us about specialties in which you are interested.

Mueller Steam Specialty Co., Inc.

349-351 West 26th Street, New York City

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

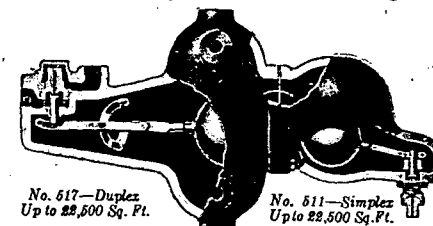
No. 11—For Vacuum, Vapor and Low Pressure Heating Systems. Initial Pressures, up to 200 lb.; Reduced Pressures, 0 to 10 lb.

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.

Automatic Water Feeders

With a powerful leverage to control the water line in steam boilers, etc. They supply make up water to compensate for evaporation, leaks, steam utilized in process work and condensation wasted. Where condensation held up in the system eventually returns in large quantities, our Duplex type protects the boiler against flooding. All working parts of non-corrosive metal, are accessible without breaking pipe connections. Provided with an integral strainer. For steam pressures up to 100 lb., water pressures up to 120 lb.



No. 617—Duplex
Up to 22,500 Sq. Ft.

No. 611—Simplex
Up to 22,500 Sq. Ft.

Steam Traps

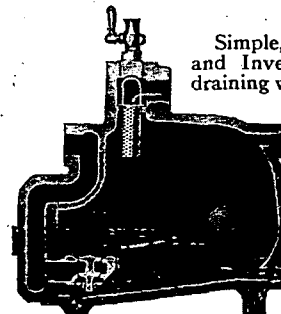
Simple, Sturdy and Compact Ball Float and Inverted Bucket Steam Traps for draining water of condensation from steam apparatus and steam mains.

Powerful leverage enables them to take care of large quantities of condensation.

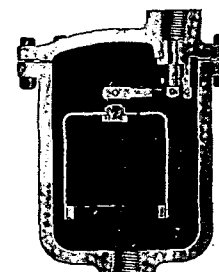
Ball Float Steam Traps equipped with integral strainer, water gages, air cocks, blow off and integral by-pass valve, when desired.

All working parts are accessible without disturbing any pipes.

Valves are sealed with several inches of water, making the escape of steam impossible.



Ball Float
No. 219—Up to 50 lb.
No. 221—Up to 150 lb.
Sizes $\frac{1}{2}$ to 3 in.



Inverted Bucket
No. 211—For Pressures
Up to 250 lb.
Sizes $\frac{1}{2}$ to 3 in.

CATALOGUE and BULLETINS covering our COMPLETE LINE gladly furnished on application.

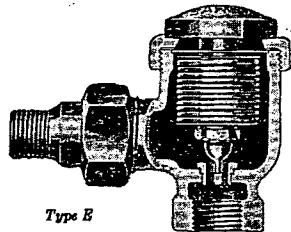
Sarco Company, Inc.

183 Madison Ave., New York, N. Y.

Branches in Principal Cities

SARCO CANADA LIMITED, FEDERAL BLDG., TORONTO, ONT.

PRODUCTS—A complete line of Specialties for Vapor, Vacuum and Gravity Steam Heating Systems, combined with a competent Engineering Service to architects and heating engineers to assist them in providing modern heating.

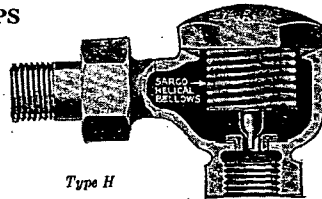


Type B

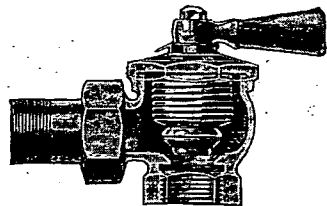
SARCO RADIATOR TRAPS

These efficient traps are equipped with bellows made from our own heavy wall, seamless, corrugated bronze tubing. They are exceptionally rugged, have long life

and high capacity. Available in angle, straightway and corner patterns. The body is heavy brass, nickel plated, with polished trimmings.



Type H



Sarco Bellows-Packless Valve

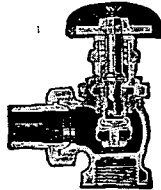
SARCO RADIATOR VALVES

For high vacuum heating systems, Sarco offers Sarco Bellows-Packless Valve. It is of the truly packless, modulating type. Leakage at the stem is impossible. The stem is sealed to the cap by Sarco seamless, corrugated bronze tubing.

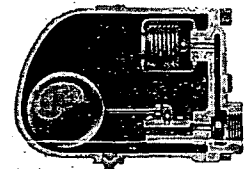
The Sarco-Marsh valve is of the commercial packless type in which the stem is sealed by a metal-to-metal cone seat, re-enforced by moulded packing held compressed by a spring.

Valve bodies of both types are cast brass, heavily nickel plated with polished trimmings.

Can be furnished with round or lever handles, or lock shield, in angle, straightway or corner patterns.



Sarco-Marsh Packless Valve



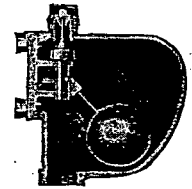
Sarco "FT" Trap

SARCO FLOAT—THERMOSTATIC TRAPS

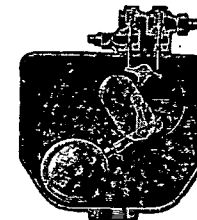
Used for dripping the ends of mains and risers. Also for stack or blast heaters, large unit heaters and hot water generators. Have automatic thermostatic air vents built in. Available in seven sizes with connections $\frac{3}{4}$ to 2 in. for pressures 0 to 200 lb.

SARCO AIR ELIMINATORS

For venting air from vapor systems at one central point in the basement. Available in two sizes: No. 6 for systems up to 2,000 sq. ft. and No. 12 for 15,000 sq. ft. Both are equipped with float valves to stop water escaping through the vent, and with check valves to prevent ingress of air when system is under vacuum.



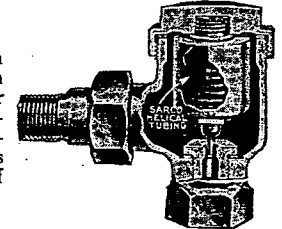
Air Eliminator No. 12



Alternating Receiver

SARCO ALTERNATING RECEIVER

A complete line of boiler return traps for vapor systems. Return water of condensation to boiler automatically, thereby assuring positive return of water under all pressure conditions. Made in six sizes for from 1,500 to 25,000 sq. ft. of radiation.



Type S-75 Steam Trap

SARCO MEDIUM PRESSURE TRAPS

These thermostatic steam traps are designed specially for kitchen fixtures and hospital equipment. Available in sizes $\frac{3}{8}$ to 1 in. in two types; S-65 for pressure 15 to 65 lbs. and S-75, 0 to 100 lb.



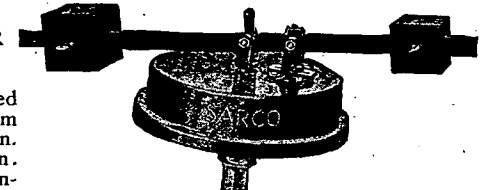
Type TR-21

SARCO TEMPERATURE REGULATOR

A self-contained automatic valve for controlling temperatures from 0 to 300 F. For hot water heaters, tanks, sterilizers and room control. Available in sizes $\frac{1}{2}$ to 6 in.

SARCO DAMPER REGULATOR

Specially designed for vapor and vacuum systems. Has 10-in. diaphragm. 18-in. lever and 1-in. connection.



Damper Regulator Type D-8



Compound Gauge

SARCO COMPOUND GAUGE

Recommended for vapor and vacuum systems in place of pressure gauge. Has $4\frac{1}{2}$ -in. dial, range 0 to 30 lb. pressure, 0 to 30-in. vacuum with extended scale 0 to 5 lb. and 0 to 10-in. for easy reading.

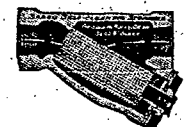
SARCO STRAINERS

Pipe line strainers are effective insurance against injury to traps, valves, meters, pumps, etc.

Sarco strainers are made with screens for steam, water, oil, gas, brine or ammonia.

CATALOG

For full information on all Heating Specialties write for Catalog HV-45.



Strainer

Sterling Engineering Company

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MANUFACTURERS OF STEAM AND WATER HEATING EQUIPMENT

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PRODUCTS

The 1936 models of Sterling products are the result of twenty years' experience in the manufacture of heating control devices. They are designed to provide uninterrupted service for the life of the building. They are guaranteed against mechanical defects and provide the highest degree of attainable efficiency.

The products of the Sterling Engineering Company are divided into four groups:

Group 1—Products in this group are described in Bulletin No. 234 of the Sterling Engineering Company's loose leaf catalog. They consist of:

Diaphragm type thermostatic radiator traps, bellows type thermostatic radiator traps, float and thermostatic traps, blast traps, boiler return traps, bellows packless quick-opening radiator valves, spring packed quick-opening radiator valves, vents and air eliminators, strainers, damper regulators, orifices, air line valves and pressure-vacuum gauges.

Group 2—Products in this group are described in Bulletin No. 334 of the Sterling Engineering Company's loose leaf catalog. They consist of:

Return line vacuum heating pumps, condensation pump and receiver units and air line vacuum pumps.

Group 3—Products in this group are described in Bulletin No. 352 of the Sterling Engineering Company's loose leaf catalog. They consist of:

Type E No. 100 self-contained thermostatic radiator control valves, Types G No. 101 and G No. 102 remote control valves, Type F No. 103 remote control valves, Type J No. 104 unit heater control valves, No. 110 tank control valves and Nos. 18, 19, 20 and 25 motor operated valves.

Group 4—Products in this group are described in Bulletin No. 434 of the Sterling Engineering Company's loose leaf catalog. They consist of:

Sterling Compellers for circulating water in closed circuits, Sterling Instant Water Heaters which heat water instantaneously, using the water from the heating boiler as a heat transfer medium and Sterling No. 21 automatic valves which control the flow of hot water to the heating system. The combination of these products, with the necessary electrical controls and thermostats, constitutes the Sterling Instant System of Water Heating.

The Sterling Engineering Company maintains direct company representatives in the cities listed above. The representatives are engineers schooled in the technique of steam and water heating and are prepared to give technical advice and cooperation to architects, engineers and contractors on the application and use of Sterling equipment.

Write the Company at Milwaukee or communicate with our representative for a copy of our complete loose leaf catalog describing the above mentioned products in detail and giving complete information regarding their use, application and installation.

The Trane Company

La Crosse, Wis.

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TRANE PRODUCTS

The many applications of Trane heating, cooling, ventilating, drying, processing and air conditioning equipment make it impossible to discuss them all here. Because of this we are listing for the convenience of the engineer, the architect, and the contractor a list of available catalogs. Any or all of these catalogs will be sent on request to interested parties.

HEATING SPECIALTIES

Complete catalog on steam and vacuum heating specialties which includes complete data on the Trane 14-Corrugation bellows traps and Hermetic Valves.

Bulletin on orifice steam heating systems for residences and buildings requiring up to 800 sq. ft. of radiation.

TEMPERATURE CONTROL

Bulletin describing the Trane Temperature Control Valve with remote control and balanced pressure for controlling room temperatures.

CONVECTION HEATERS

Technical data on Trane Convection Heaters.

Type and model bulletin—Convection Heater Capacities and Roughing-in Dimensions Trane Convection Heaters.

UNIT HEATERS

Trane Floor Line Spread Unit Heater bulletin containing complete Engineering Data with capacities and Roughing-in Dimensions of all sizes of Propeller Fan Type Units.

Trane Torridor bulletin—containing complete engineering data with capacities and roughing-in dimensions of all sizes of Blower Type Units.

EXTENDED SURFACE

Bulletin containing complete information on the application of Trane extended heat transfer surface to fan systems of heating, drying, and process applications. Complete data.

Four Bulletins:

1. Direct Expansion Cooling.
2. Low Pressure Steam.
3. High Pressure Steam.
4. Water Heating and Cooling.

COOLING

Trane Comfort Coolers.
Trane High Velocity Coolers.
Trane Product Coolers.

CLIMATE CHANGERS

A manual and complete bulletin on the residential Air Conditioning Unit for use with steam or hot water boilers.

A manual and complete bulletin on the Direct Fired Climate Changer.

A bulletin on the Climate Changer for humidification only.

HUMIDIFIERS

Trane Cascade Humidifier for use with Convection Heater systems. Trane Climate Changer Humidifier for hot water and steam heating systems.

VENTILATION

The Trane Air-o-Lizer a complete heating, ventilating unit for schoolroom ventilation.

PUMPS

Trane small Centrifugal Pumps for condensate return and circulating work.

RAILWAY AIR-CONDITIONING

A complete manual on railway air conditioning equipment and application of air conditioning units for ice, steam, jet, and mechanical refrigeration systems.

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

since 1888
Webster
Systems of
Steam Heating

Main Office and Factory:
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PRODUCTS AND SERVICES

Improved Webster Systems of Steam Heating including Vacuum and Type "R" (vapor).

Webster Central Control Systems including HYLO and MODERATOR. 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.

Webster Series "78" Traps for use at process pressures (10 to 125 lb. per sq. in.)

IMPROVED WEBSTER SYSTEMS

The Improved 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. Improved 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.

CENTRAL CONTROL SYSTEMS

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 Hylo Systems may be provided for manual control, or if desired, may be semi-automatic by incorporation of inside thermostat or thermostat and schedule clock. The Moderator Systems employ an automatic Outdoor Thermostat sup-

plemented by a manual Variator.

The latter is used for quick heating-up, night load, and unusual weather or occupancy conditions. Use of Webster Central Control Systems results in (1) increased comfort because over-heating and underheating are minimized and (2) lower fuel or steam costs.

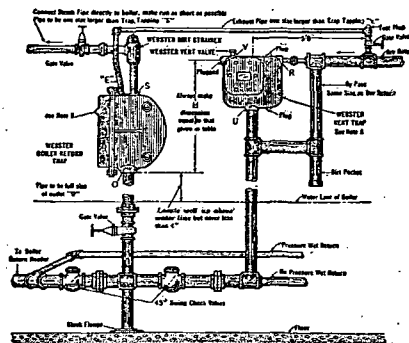


Fig. 1. Conventional arrangement of piping around Webster Basement Equipment for the Webster Type "R" System

WEBSTER SYSTEM RADIATION

Concealed, non-ferrous type for use exclusively with Improved Webster 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

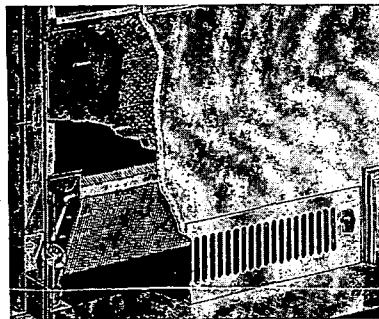


Fig. 2. Webster System Radiation

and return piping connections. 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.

RADIATOR SUPPLY VALVES

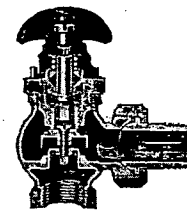


Fig. 3. Webster Type "W" Valve

Made in angle, right-corner, left-corner, straightway single union and double union models with either wheel, lever, lockshield, chain wheel or extended stem handles. Sizes: $\frac{1}{2}$, $\frac{3}{4}$, 1 and $1\frac{1}{4}$ in.

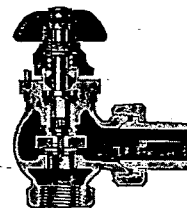


Fig. 4. Webster Three-Point Valve

normal-excess-type. Excess setting gives 140 per cent of normal steam flow. Otherwise similar in design to Type "W" Valve. Sizes: $\frac{3}{4}$ and 1 in.

Sylphon Packless—Positively packless having flexible laminated Sylphon bellows completely enclosing stem. Quick-

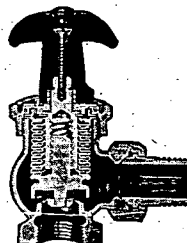


Fig. 5. Webster Packless Valve

Type "W"—Standard supply valve of highest quality. Slotted sleeve gives "modulation" of steam flow provided proper pressure is maintained on system. Quick-opening, non-rising stem type. Uses molded ring packing. Valve disc is renewable.

"Three-Point"—Has sleeve orifice fitting into seat opening. Especially desirable for hospitals, hotels, etc., where quick heating-up or higher temperatures are required in different rooms. Is normally an open-shut type but may be easily converted to shut-

Type "B"—Quick-opening, non-rising stem, packless type. Special gland takes up wear of molded ring packing. Renewable composition disc. Made in angle, right-corner, left-corner and straightway single-union and double-union models with wheel, lockshield, chain wheel and extended stem handles. Sizes: $\frac{1}{2}$, $\frac{3}{4}$, 1, $1\frac{1}{4}$, $1\frac{1}{2}$ and 2 in.

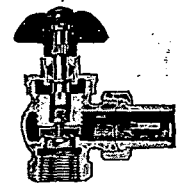


Fig. 6. Webster Type "B" Valve

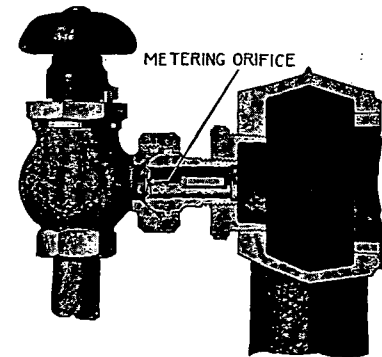


Fig. 7. Metering Orifice Inserted in Union Connection of a Webster Supply Valve

Metering Orifices—Accurately sized and made of heavy gage Monel Metal to resist erosion and corrosion, amply thick to be free from vibration and shaped for quiet operation. Available in a number of types to fit both new and installed radiator supply valves of Webster or other make.

RETURN TRAPS

Sylphon—Original and highly perfected type of low pressure, thermostatic bellows trap. Rugged in construction. Renewable seat. Factory adjusted. Made in angle, right-corner, left-corner and straightway bodies. Sizes: $\frac{1}{2}$, $\frac{3}{4}$ and 1 in. Normal operating pressures up to 5 lb. per sq. in. Maximum occasional pressure 10 lb. per sq. in.

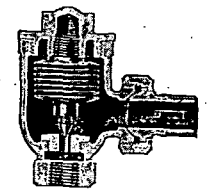


Fig. 8. Webster 518 Sylphon Trap

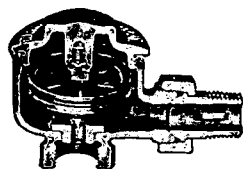


Fig. 9. Webster
Size 708 Trap

Series "7"—Perfected diaphragm-type thermostatic trap. Unusually strong in construction. Renewable seat. Factory-adjusted, phosphor-bronze diaphragm. Made in angle, right-corner, left-corner and straightway bodies. Sizes: $\frac{1}{2}$, $\frac{3}{4}$ and 1 in. Normal operating pressures up to 5 lb. per sq. in.; maximum occasional pressure, 10 lb. per sq. in.

Series "7-M"—Similar in design to Series 7 but built for normal operating pressures up to 15 lb. per sq. in. Maximum occasional pressure is 25 lb. per sq. in. Uses Monel Metal diaphragm, valve piece and seat insert.

Series "26"—A heavy duty trap for drips of mains, blast radiation, unit heaters and similar applications. A rugged float-type trap available with and without thermostatic air vent. Made in three sizes: 200, 700 and 1200 lb. water per hour at 2 lb. pressure difference. 700 lb. size has integral dirt strainer; other sizes do not. Maximum working pressure is 15 lb. per sq. in.

Series "19" (not illustrated)—A float-type, heavy duty trap for large quantities of water as in hot water generators, fin heater coils and steam mains. Has thermostatic air vent. Five sizes handling up to 11,700 lb. water per hour at 2 lb. pressure difference. Maximum working pressure is 15 lb. per sq. in.

Series "16" (not illustrated)—Similar to Series 19 but without thermostatic air vent. For heavy duty applications where concentration of air does not occur.

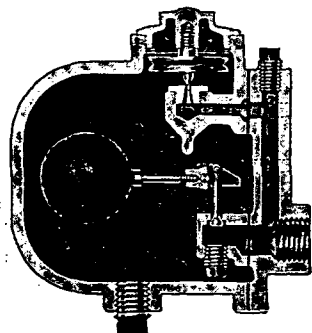


Fig. 10. The Webster Size 00089-T Drip Trap is Rated 800 lb. Water per Hour at 2 lb. Pressure Difference

Series "78"

—thermostatic trap built for process steam pressures (10 to 125 lb. per sq. in.). Monel Metal diaphragm. Stainless Steel valve piece and seat insert. Angle model only. Sizes: $\frac{3}{8}$, $\frac{1}{2}$, $\frac{3}{4}$ and 1 in. Extensively used with laundry, cooking, sterilizing and other process-steam-using equipment.

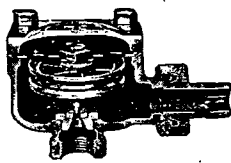


Fig. 11. Webster
Series "78" Trap

DIRT STRAINERS AND POCKETS

Placed in return lines of steam heating systems to prevent dirt, rust and scale from impairing tightness of traps.

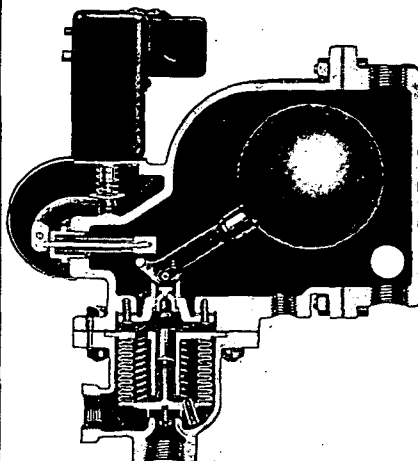


Fig. 12. Size 34-C Webster Boiler Protector with Low Water Electrical Cut-out Switch. Size 34 has no Cut-out Switch

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 gauge 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 in two sizes, 34 and 030, both with $\frac{3}{4}$ in. connections. Size 34 can be furnished with electrical cut-out switch.

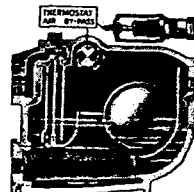
Wright-Austin Co.

317 West Woodbridge St., Detroit, Mich.

PRODUCTS—Steam Traps, Strainers, Air Traps, Steam and Oil Separators, Compressed Air Purifiers, Exhaust Heads, Boiler Feeders, Alarm Water Columns, Water Gauges, Trycocks.

"Combination" Steam Trap

Float Type with internal thermostatic air bypass and strainer. A modernly designed and very successful trap for vacuum and pressure heating.



"Airxpel" Bucket Type Steam Traps

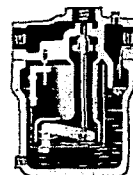
Are "double duty" traps, because they automatically discharge both air and condensate.



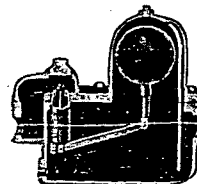
At left is illustrated the "Baby" Airxpel, a new streamline body design, which adds attractiveness to the installation. Has the handiest of all pipe connections for radiation, etc.

The Cub sizes are made in $\frac{1}{2}$ in., $\frac{3}{4}$ in., 1 in. Especially suitable for individual unit drainage on heating and process equipment.

Also, three "Master" sizes $\frac{1}{2}$ in. to 2 in., for general service.



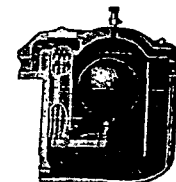
"Victor" Low Pressure Steam Trap



For heavy volumes of condensation at low pressures.

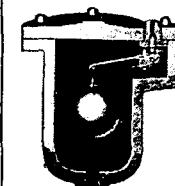
"Emergency" Float Type Steam Trap

Three valve trap with large capacity at high pressures. An exceptionally reliable trap for use in inaccessible places.



Air Relief Trap

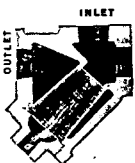
For relieving air from forced circulation hot water heating systems, water supply lines, closed tanks, receivers, pumps, etc.



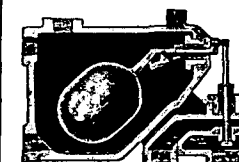
"Tuway" Strainer

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.

For cleaning, flush through blow-off connection, or remove screen by unscrewing bottom plug.



Boiler Feeder

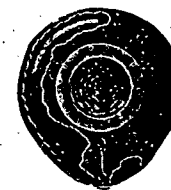
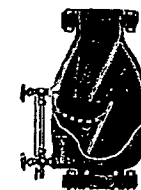


An iron watchman that automatically maintains uniform water level in low pressure boilers.

Separators—Steam and Oil

Type "A" Vertical Steam

Type "S" Horizontal Oil



We make separators of every type. Send for descriptive Bulletins on any of the items listed on this page.

Detroit Stoker Company

Sales and Engineering Offices
General Motors Bldg., Detroit, Mich.

Main Office and Works at
Monroe, Mich.

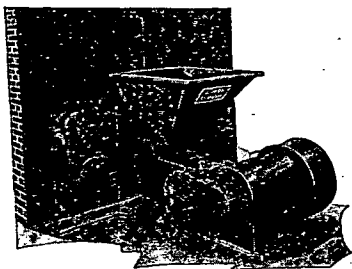


Since 1898

District Offices in
Principal Cities

Built in
Canada at London, Ont.

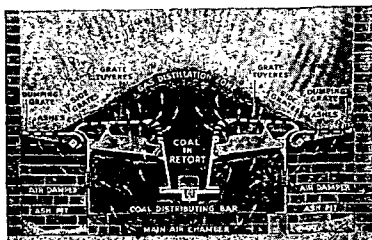
A Detroit Stoker for Every Service. Each installation, regardless of size is carefully studied from an engineering standpoint to insure best results. Sold and installed by our own organization.



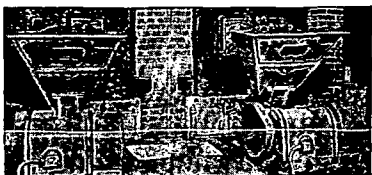
Detroit LoStoker, showing Motor Driven Blower and Stoker, One Compact Unit at the Front

Detroit LoStoker

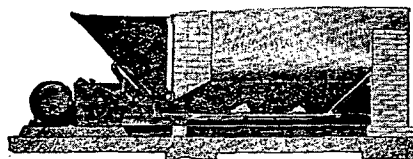
Built in various widths and lengths to fit furnaces of all types of boilers. Compact, easily installed, responsive and automatic. Savings, due to increased efficiency, combined with the ability to successfully burn less expensive grades of coal, make Detroit Stokers pay a handsome return on the investment. Write for Bulletin 369.



Large Active Fuel Bed, with Provision for Admitting Air under the Dumping Grates at each side to burn off the Combustible Prior to Dumping Ashes



Detroit LoStokers with Firebox Boilers



Detroit LoStoker
Adjustable Plunger Feed
Built in various sizes to fit the furnace

Detroit LoStoker Advantages include:

Agitator in coal hopper for positive coal feed. Cannot stick or jam with wet coal.

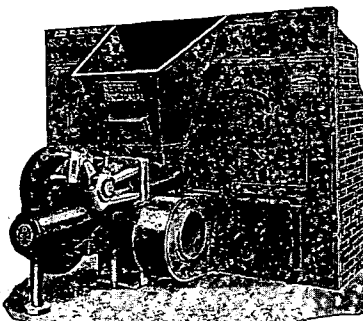
Adjustable Plunger Feed for the control of the quantity of coal.

Heavy Mechanical Drive of simple design. Little power is required for operation.

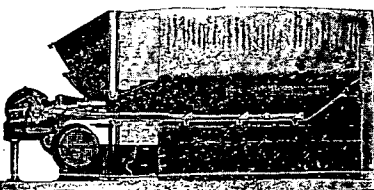
Side Cleaning with dumping grates. Ash doors are provided. No hand cleaning.

Automatically Controlled. Motor or turbine driven, controlled from steam pressure, water temperature or thermostat.

Detroit UniStoker



Detroit UniStoker for Large Heating Boilers.
Each Boiler and Stoker, an Independent Unit



Detroit UniStoker, Plunger Feed, Side Cleaning, Motor or Steam Turbine Driven. (Arrows Indicate Flow of Air to all parts of the Large Active Fuel Bed)

Whiting Corporation

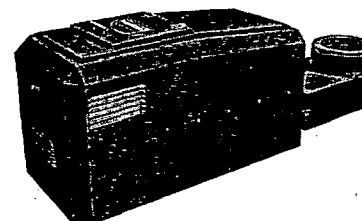
15620 Halsted Street, Harvey, Ill.

(Chicago Suburb)

Representatives in All Principal Cities

PRODUCTS—Complete line of Stokers and Pulverizing Equipment for firing boilers (up to 5,000 hp) including Horizontal Compression Feed Stokers, Underfeed Stokers for small boilers and furnaces, Impact and Table Roller Pulverizers, Pulverized Coal Conveying and Feeding Equipment, Burners, etc.

UNDERFEED STOKERS FOR AUTOMATIC HOME HEATING

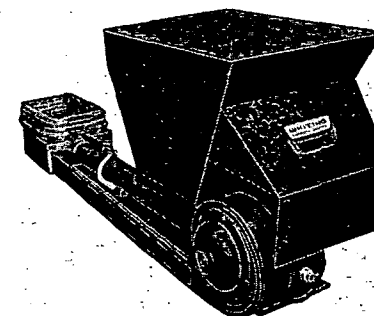


Standard models range in capacity from 20 to 90 lb. of bituminous coal per hour and fit practically any solid-fuel boiler or furnace, old or new. Dual Draft Burner permits efficient combustion of lowest-priced grades of coal. Further economies result from the efficiency with which fuel-feeding is regulated. Hopper is only 25 in. from floor to facilitate filling, each day or so. Except for setting upstairs thermostat, operation is then entirely automatic. The Whiting Stoker is practically noiseless and odorless. There is no back-firing—no ashes (only a few clean, hard clinkers). Operating cost is substantially less than any other form of automatic heating—a clean, safe stoker which may always be hand-fired in a storm or other emergency, when electric wires may be down.

UNDERFEED STOKERS FOR COMMERCIAL USE

For either Refractory or Dead Plate Setting—ranging in capacity from 90 to 1250 lb. of bituminous coal per hour. They are designed to fit all types of boilers—cast iron, locomotive, horizontal return tube, vertical down draft, and brick set. Like the Domestic Models, they are equipped with Dual Draft Burner, permitting the efficient combustion of lowest-priced grades of coal.

Whiting Underfeed Stokers are designed for small industrial plants—warehouses and stores—laundries—dairies—garages—greenhouses—hotels and apartments—hospitals—public buildings—public and private institutions—schools, colleges and churches. Their installation reduces fuel expense and operating costs. Less labor is required, and the boiler room kept cleaner.



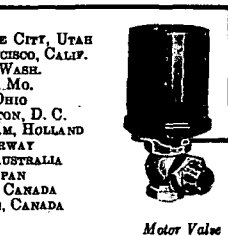
Barber-Colman Company

Rockford, Illinois



BARBER-COLMAN ELECTRIC SYSTEM OF TEMPERATURE AND HUMIDITY CONTROL AND UNI-FLO GRILLES AND REGISTERS

Branches and Representatives

 Thermostat ATLANTA, GA. BUFFALO, N. Y. CHICAGO, ILL. CINCINNATI, OHIO CLEVELAND, OHIO DENVER, COLO. DETROIT, MICH. DULUTH, MINN. FRAMINGHAM, MASS. GREENVILLE, S. C. INDIANAPOLIS, IND. KANSAS CITY, MO.	LOS ANGELES, CALIF. MILWAUKEE, WIS. MINNEAPOLIS, MINN. MOLINE, ILL. NEW ORLEANS, LA. NEW YORK, N. Y. OMAHA, NEBR. PHILADELPHIA, PA. PHOENIX, ARIZ. PITTSBURGH, PA. PORTLAND, ORE. RICHMOND, VA.	 Motor Valve SALT LAKE CITY, UTAH SAN FRANCISCO, CALIF. SEATTLE, WASH. ST. LOUIS, MO. TOLEDO, OHIO WASHINGTON, D. C. AMSTERDAM, HOLLAND OSLO, NORWAY SYDNEY, AUSTRALIA TOKYO, JAPAN TORONTO, CANADA WINNIPEG, CANADA
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FEATURES

The Barber-Colman Electric System of Temperature and Humidity Control can be applied to any kind of heating, ventilating, or air conditioning installation. Accurate, automatic controls can be furnished for a single room or an entire building, for an ordinary residence or an ornate city hall, for one radiator or a large and complicated blast system. Suitable apparatus can be provided to handle steam, hot water, warm air, air conditioning, humidification, cooling—anything that up-to-date engineering requires. Barber-Colman apparatus comprises fifteen different types of thermostats, twelve different types and hundreds of sizes of motor-operated valves, ten different types of motor-operated damper controllers, as well as hygrometers and other special equipment and accessories. Barber-Colman control systems are installed and operating successfully in every section of the country. Communications from heating engineers regarding control problems will be welcomed.

All-Electric—All units operated by electricity, all connections electrical.

Low-Voltage—Power used 25-volt a.c. Smaller size wire and less insulation required. Facilitates installation in new or existing buildings.

Motor - Driven—Operating units powered by small electric motors. No actuation by gases or liquids.

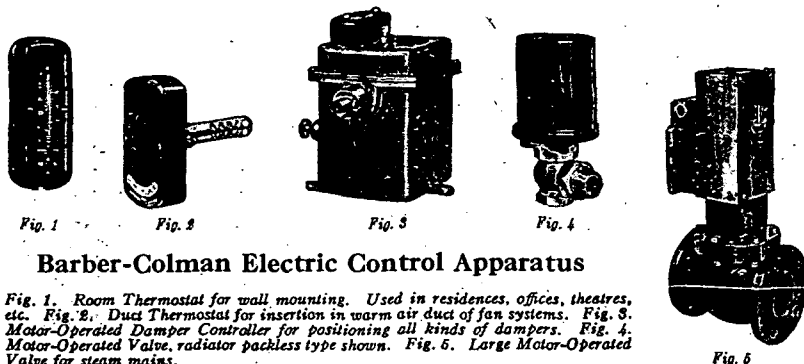
Minimum Auxiliaries—Few required by all-electric equipment.

Appearance—Neatness and good appearance stressed in design. Units occupy minimum space.

Variety and Adaptability—Sufficient equipment available to provide proper controls for any heating, ventilating, or air conditioning system.

Low Maintenance Cost—Minimum upkeep work required. Apparatus will perform dependably without attention.

Underwriters' Approval—The Barber-Colman Electric System of Temperature and Humidity Control is listed as standard by the Underwriters' Laboratories.



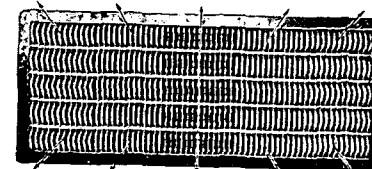
Barber-Colman Electric Control Apparatus

Fig. 1. Room Thermostat for wall mounting. Used in residences, offices, theatres, etc. Fig. 2. Duct Thermostat for insertion in warm air duct of fan systems. Fig. 3. Motor-Operated Damper Controller for positioning all kinds of dampers. Fig. 4. Motor-Operated Valve, radiator packless type shown. Fig. 5. Large Motor-Operated Valve for steam mains.

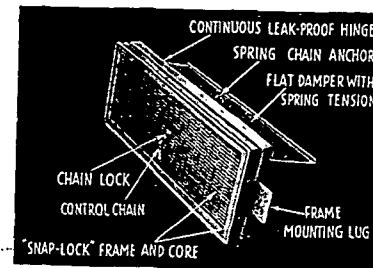
GRILLES **UNI-FLO** REGISTERS

ENGINEERED AIR DISTRIBUTION OUTLETS

Featuring—Aspiration, Diffusion, Directional Flow



Grilles—For sidewall, panel, and base-board applications. SNAP-LOCK construction. Frame is roughed in after studding is in place. Fin core snaps into frame when decoration is completed. Directional Fins furnished as required.

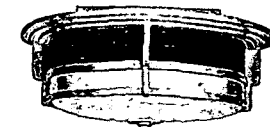


Registers—Damper under spring tension. Continuous leak-proof piano hinge. Flexible control chain with positive lock. Spring tension on damper when closed. Tamper-proof key operation optional. SNAP-LOCK construction.

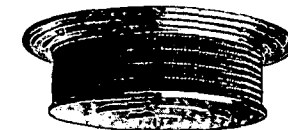
Aspiration—An unique effect caused by turbulence of UNI-FLO air stream. Entrains surrounding air to improve circulation.

Diffusion—Turbulence of UNI-FLO air stream causes thorough mixture with room air and rapid rise of air stream temperature.

Directional Flow—Deflecting fins can be arranged to direct all or any parts of air stream at various angles up, down or sideways as required by location of outlet.

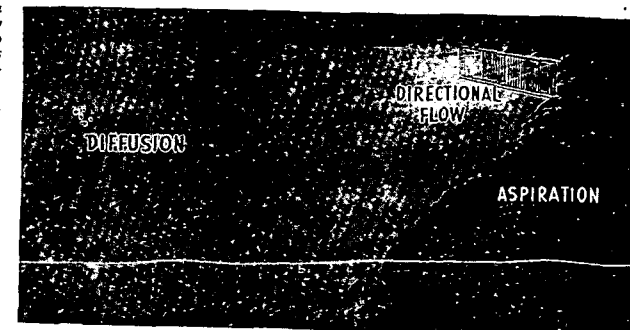


AIR-LITE—Ceiling unit combining air outlet and lighting fixture. Uniform diffused air flow in all directions. Two sizes.



Ceiling Grilles—Air outlet only. UNI-FLO diffusing fins deliver soft blanket of air. Adjustable volume control. Three sizes.

UNI-FLO Fins—Multiplicity of thin fins have a number of valuable and unique features. Diffusers on edges of fins produce turbulent, rolling air stream that mixes rapidly with room air. Permissible low supply air temperatures and high discharge velocities result in small duct and grille sizes. "Aspiration"—definitely a UNI-FLO characteristic—induces thorough circulation, uniform temperature throughout conditioned space. Long throw of air possible. Deflecting fins provide accurate distribution of air in room.



Detroit Lubricator Company

Detroit, Michigan, U. S. A.

NEW YORK, N. Y., 40 West 40th Street

CHICAGO, ILL., 816 S. Michigan Avenue LOS ANGELES, CALIF., 3251 Wilshire Blvd.

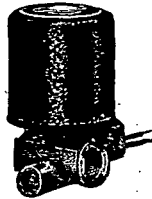
Canadian Representative:

RAILWAY AND ENGINEERING SPECIALTIES LIMITED, Montreal, Toronto, Winnipeg

Division of American Radiator & Standard Sanitary Corporation

AIR CONDITIONING CONTROLS

Solenoid Valve

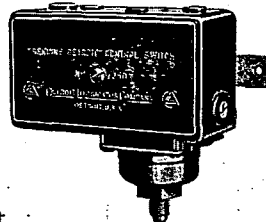


No. 673-R

A dependable shut-off for liquid or suction lines in both air-conditioning and refrigeration. Draws approximately 12 watts during operating period. Furnished with either $\frac{3}{16}$, $\frac{1}{32}$ or $\frac{1}{4}$ in. orifice. See Bulletin 34.

Pressure Control (Model RB-3)

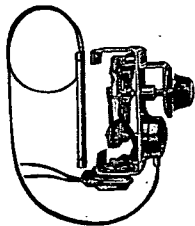
Accurately controls low side pressure. Range 20 in. vacuum to 40 lb pressure. Differential adjustable from 5 to 25 lb. Available in other models to control temperatures. See Bulletin 60.



No. 250

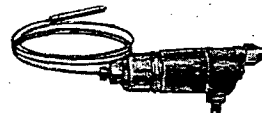
Cabinet Thermostat

A high voltage control entirely concealed except for the knob on the outside of the cabinet. Permits integral wiring of a portable cabinet heater or cooler. Positively insulated against both radiant and conducted heat. See Bulletin 67.



Automatic Expansion Valve

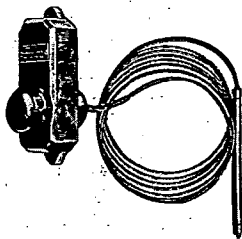
Either gas or liquid charged—sensitive, efficient and dependable on either low or high temperatures. Thermostatic element provides automatic operation. Assures full flow and maximum tonnage. Capacities up to 2 tons sulphur dioxide or Freon. Other Thermostatic Expansion Valves for capacities up to 21 tons on Freon.



No. 673

Differential Thermostat

Can be furnished in either Northern or Southern range to assure maximum comfort. Designed primarily for temperature control with room cooling apparatus. It operates on differences in temperature inside and outside. Write for Bulletin 74.



No. 691

ZONE CONTROL

For proper control of forced air systems, zoning is necessary. The "Genuine Detroit" Duct Damper Motor No. 431 may be mounted directly on the duct without risk of objectionable noise. The No. 431 controls both fuel and air supply. It is neat in appearance and easily installed.

OTHER CONTROLS

This Company can also supply you with a full line of Boiler and Furnace Limit Controls—Room thermostats both high and low voltage for both heating and cooling—Fan controls—Humidity and Stoker Controls.

Julien P. Friez & Sons, Inc.,

(A Subsidiary of the Bendix Aviation Corporation)

Baltimore, Maryland, U. S. A.

Established 1876

Manufacturers of a Complete Line of Automatic Electric Controls for Industrial and Comfort Applications. Also a Complete Range of Recording and Accurate Measuring Instruments for Indoor and Outdoor Applications



Humidistats—Human hair elemented, pre-eminent in their field. See Bulletin #A.

Thermostats—Precision instruments, beautiful in appearance, high quality. See Bulletin #T.

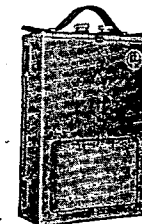
Effectistats—Providing Effective Temperature Control with simple heating (or cooling) equipment only. See Bulletin #E.



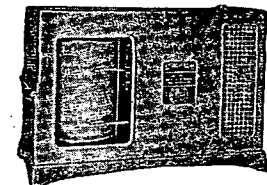
Indicators—Human hair elemented, accurate and reliable, moderate in price. See Bulletin #D.

Comfortrols—Providing complete Effective Temperature Control where complete year round air conditioning equipment is available. See Bulletin #C.

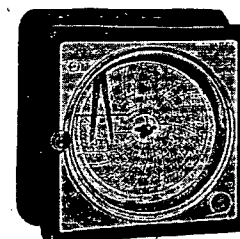
Hytherstats—Humidistat and Thermostat incorporated into a single case. See Bulletin #C.



Portable Recorders—Recording relative humidity, temperature, and running time. Rugged, reliable, accurate, and moderate in price. See Bulletin #G.

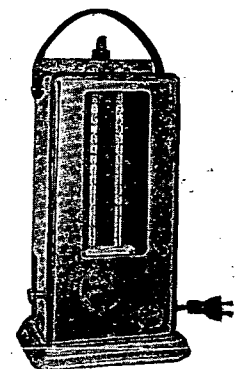


Hythergraphs—Recording relative humidity and temperature. Precision instruments designed for permanent or semi-permanent installations. See Bulletin #H.



Round Chart Recorders—Recording relative humidity and temperature for wall mounting and for local or remote recording. See Bulletin #R.

Thermo-Shielded Electrically-Aspirated Psychrometers—A most accurate instrument for precise measurement of Wet and Dry Bulb conditions. Represents 60 years' experience in construction of accurate instruments.



Anemometers—Biram and Direct-Reading Types.

Write for full details of the complete Friez line of indicating, recording, and measuring instruments for indoor and outdoor applications.

The Fulton Sylphon Company

SYLPHON AUTOMATIC TEMPERATURE CONTROLLING INSTRUMENTS

Knoxville,
Tennessee



Sales Representatives
in Principal Cities

PRODUCTS

Automatic Radiator Valves for Room Temperature Control; Space Temperature Regulators; Unit Heater Regulators; Duct Temperature Regulators; Damper Regulators; Hot Water Supply Temperature Regulators; Regulators for the Control of Refrigeration Temperatures, Air Conditioning Systems, Drinking Water Systems, etc.; Packless Expansion Joints for Heating and Plumbing Risers; Pressure Reducing and Regulating Valves.

GENERAL INFORMATION

The Fulton Sylphon Products, listed above and described in the following paragraphs, depend upon the famous Sylphon Metal Bellows for their remarkable, trouble-free service and long life.

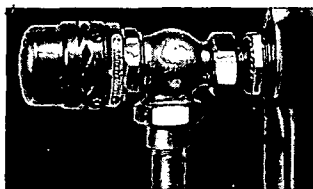
By continuous engineering study and intimate contact with heating and refrigeration problems, this Company has used this efficient, practically indestructible bellows in the development of a line of temperature control and heating specialties known for their outstanding performance qualities.

In attaining and maintaining this enviable position, THE FULTON SYLPHON COMPANY's plant has been extended until today it is one of the largest in the world specializing in the manufacture of this type of equipment.

Every phase of the quality production of Sylphon products from raw materials to the finished product is scientifically controlled in this plant. At this plant, a unique testing laboratory is maintained as a proving ground where every Sylphon product is subjected to conditions duplicating the most strenuous actual service in the field before approval for shipment.

It is with the knowledge of these resources behind the Fulton Sylphon line that engineers prefer, whenever possible, to specify Fulton Sylphon equipment. For they know by experience that permanent satisfaction will always be assured.

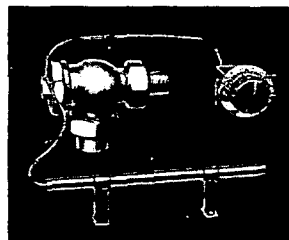
SYLPHON AUTOMATIC RADIATOR VALVES



No. 885 Sylphon Automatic Radiator Valve for exposed radiation. (Globe Type No. 886)

Used in place of ordinary radiator valves to provide individual room temperature comfort in both new and existing buildings. Entirely self-contained, self-powered and self-reliant, they require no complicated piping or auxiliary equipment.

Types for both exposed and concealed radiation are available.

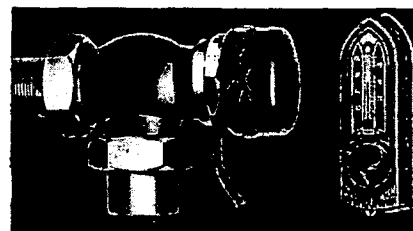


Sylphon Automatic Radiator Valve for Concealed Radiation. Angle Valve, No. 895. Globe Type, No. 896

Sylphon Valves for enclosed radiators provide the balanced heat control made necessary by the very nature of enclosed radiation. These valves are exclusive in that they provide control at two points, by two bulbs—first, at the cool air inlet at the bottom and, second, at the heated air outlet at the top of the enclosure.

Small, neat in appearance, finely finished, conveniently adjustable to the room temperature desired, and faithful in their maintenance of temperature at the set point—these valves answer the demand for an inexpensive means of providing accurate, dependable space temperature control in individual rooms, sections or throughout large buildings. Bulletin HVG-255.

SYLPHON ELECTRIC CONTROL VALVES

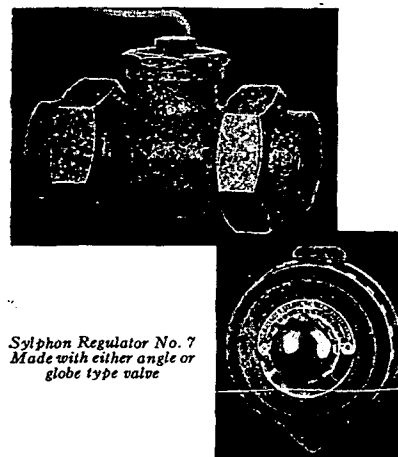


No. 890 Sylphon Electric Control Valve. (Globe Type No. 891)

Similar in appearance and action to Sylphon Automatic Valves, but operated by an electric wall thermostat. These valves utilize as their motor element a "hot bulb" containing a volatile liquid which expands, causing liquid pressure to throttle the valve, when an electric heater coil which surrounds the bulb is energized by the closing of the electric thermostat circuit.

This arrangement provides radiator valve control from a remote location, permits the regulation of several radiators from a single thermostat, and permits the inclusion of an electric time switch to govern maximum and minimum temperatures supplied to a building during working and idle hours, offering zone control of large buildings at a fraction of the cost of conventional motor-operated valve systems. Bulletin HVG-519.

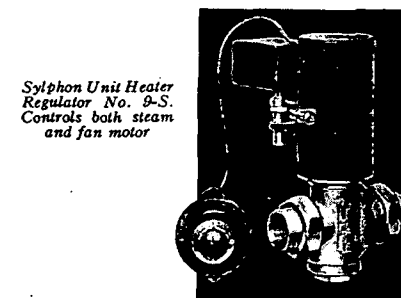
SYLPHON REGULATORS FOR UNIT HEATERS OR DIRECT RADIATION IN INDUSTRIAL AREAS



Sylphon Regulator No. 7 Made with either angle or globe type valve

This instrument is entirely mechanical and utilizes pressure of an expansible liquid in a Sylphon Bellows thermostat element transmitted through flexible tubing to another Sylphon Bellows in the valve, to actuate the packless valve and throttle the steam supply to the room or section of the building, according to temperature desired.

No. 7 Regulator may also be installed with a thermostatic switch in the steam supply line between the regulator and the unit heater or heaters when used to control this type of equipment. This switch will then act to cut off the fan motor whenever steam supply is cut off, and will start the motor again when steam is turned on. Bulletin HVG-252.



Sylphon Unit Heater Regulator No. 9-S. Controls both steam and fan motor

Sylphon Unit Heater Regulators
Nos. 9, 9-S and 9-E are similar controls for use with higher steam pressures. No. 9 controls steam supply only. No. 9-E controls steam supply and operates in connection with a room thermostat. It may be connected with a thermostatic switch in the steam line to control the fan motor as well. No. 9-S combines within the regulator itself means of controlling both steam supply and fan motor. Bulletin HVG-50.

SYLPHON PACKLESS EXPANSION JOINTS

The Sylphon Packless Expansion Joint was designed to eliminate useless building height, expensive contraction and non-revenue producing space. It prevents costly leaks and repairs, requires no repacking, and because it is always tight, it allows the heating system to operate at full efficiency. Two types, for steam (No. 110), and for water (No. 111). Thousands are in use in many prominent buildings. Write for Bulletin HVG-300.



No. 110 Sylphon Expansion Joint

Johnson Service Company

TEMPERATURE AND HUMIDITY CONTROL

General Offices and Factory

Milwaukee, Wis.

Branch Offices in all Large Cities

JOHNSON TEMPERATURE REGULATING CO. OF CANADA, LTD., 97 JARVIS STREET, TORONTO, ONT.
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. A single nation-wide organization devoted to Design, Manufacture and Installation for more than 50 years.

Temperature and Humidity Control for every range required in manufacturing and industrial processes.

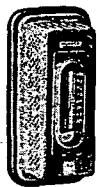
Room temperature control applied to radiators, unit ventilators, and heat delivery ducts. Johnson "Duo-Stats" to maintain the proper relationship between outdoor and radiator temperatures for groups of radiators or "heating zones."

A complete line of devices for automatic control of air conditioning systems, heating, cooling, humidifying, dehumidifying. Automatic seasonal shifting of control cycles.

Periodic Flush Systems for intermittent flushing in various sections of a building, reducing load on piping system and insuring economy in use of water.

Special bulletins and catalogues on request. Sales Engineers at branch offices in all principal cities.

Johnson All-Metal Thermostats



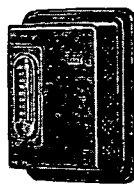
Room Thermostat

Johnson thermostats, room, insertion, and immersion types, are all-metal throughout, no soft or hard rubber parts to deteriorate and become inoperative. Every thermostat precise in construction and thoroughly tested for accuracy, efficiency, and durability.

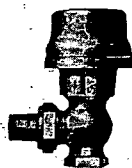
The Johnson intermediate action thermostat gives a true graduated motion to mixing dampers and valves. It holds them in an intermediate position to maintain the temperature of the room accurately within one deg. above or below the setting of the thermostat, if desired.

Johnson "Dual" or Two-Temperature Thermostats

The Dual, two-temperature, room thermostat especially adapted for use where various rooms or groups of rooms are occupied when the remainder of the building is not in use. Separate steam mains avoided. The shifting from "day" or occupancy temperature to an economy temperature for "non-occupancy" conditions, accomplished by a switch or Johnson program clock at a central point. Push buttons on the thermostat are provided when "occupant control" is desired. Dual thermostats are all-metal and operate valves and dampers gradually to maintain temperature accurately within one degree.



Dual Thermostat



Radiator Valve

Johnson Valves

Johnson diaphragm valves are simple and rugged. Seamless metal bellows and heavy spring operate the valve stem. No complicated moving parts. Made in all standard sizes and patterns. Direct acting (normally open) or reverse acting (normally closed). Three-way mixing and three-way bypass valves.

For steam, water, brine, and Freon service. Johnson valves are available, if desired, with diaphragms of special moulded rubber, super-aged and heat-resistant.

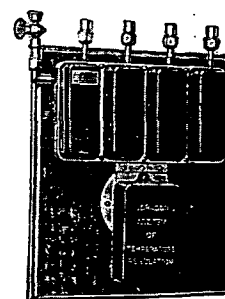


Room Humidostat

Humidostats and Humidifiers

The Johnson Humidostat automatically controls the supply of moisture delivered to the air by a humidifier or air washer and maintains a constant percentage of relative humidity. Available in both room and insertion patterns, and with various types of elements as determined by requirements, controlling within 1 per cent at relative humidity of 95 per cent and 100 deg Fahr if desired.

Johnson humidifiers are furnished in steam "grid" type or pan type with copper evaporating pan, brass heating coil, and float control.



Four-Point Insertion Thermostat

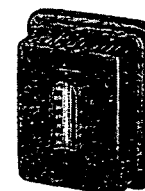
Air Conditioning Control

Summer-Winter room thermostats for operation of valves and dampers in reverse sequence for cooling and heating. Insertion and immersion thermostats in one, two, three, and four-point patterns for operating valves and dampers successively at different temperatures.

Remote readjustable thermostats, reset from a distant point by pilot or differential thermostat or by pressure switch. Differential thermostats to maintain desired temperature differences between two points, such as outdoors and treated space.

Solenoid air switches, manual switches, static pressure regulators, velocity regulators, operating dampers to regulate flow of air in ducts.

The Johnson sensitivity adjustment is an important development in the field of 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, with respect to the capacity of the conditioning apparatus.



Summer-Winter Thermostat

Zone Control

Johnson "Duo-Stats" to regulate the flow of heat in a group of radiators constituting a "heating zone" by maintaining the proper relationship between outdoor and radiator temperatures.

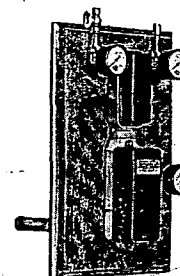
Process Control

Calibrated insertion thermostats for controlling temperature of liquids, air and gases. Mercury extended tube thermostat for remote location of sensitive element.

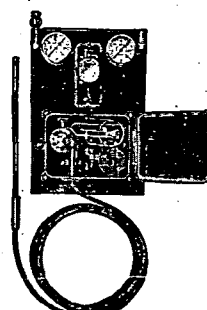
Wet-bulb thermostats for close regulation of humidity. "Record-O-Stats," combination instruments to record and control temperatures.



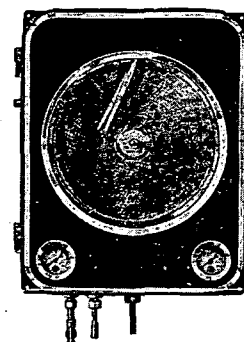
Program Clock



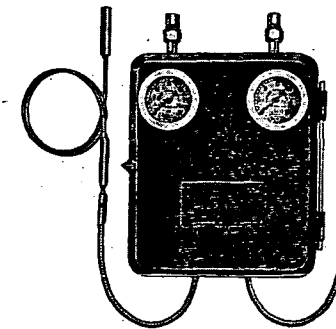
Remote Readjustable Thermostat



Mercury Extended Tube Thermostat



Record-O-Stat



Johnson "Duo-Stat"

GENERAL CONTROLS

1505 BROADWAY
CLEVELAND, OHIO



1368 HARRISON ST.
SAN FRANCISCO, CALIF.

FOR DEPENDABLE CONTROL OF TEMPERATURE - PRESSURE - FLOW

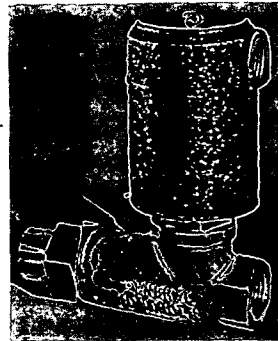
Gas - Oil - Air - Water - Steam - Refrigerants

Your installations will operate at high efficiency, with dependability and safety, if controlled by GENERAL Automatic Thermostats, Solenoid Valves and Switches. Their fundamental designs are right from an engineering standpoint. Their construction is simple—few moving parts—no pawls or catches to fail—no parts to oil—no corrosive materials to rust or jam. They are manufactured by methods of precision. The result has been an enviable record of field performance.

The GENERAL Solenoid

The quiet A. C. Solenoid is a GENERAL CONTROLS feature. Its design and simplicity of action have given the solenoid a reputation for dependable year-after-year field service.

The solenoid is current-failure, incorporating impact-type plungers. Special metal plunger-tube is rolled and sealed; no threaded joints or gaskets used. Water-proof coils are detachable and terminal connections may be rotated in any direction for ease of wiring.

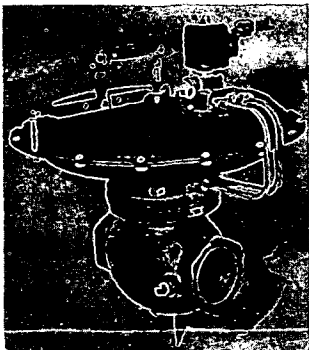


K-10 Magnetic Valve

For Domestic or Industrial Applications

Illustrated on this page are the K-10 and the B-50 magnetic gas valves widely used on industrial installations. Among the 19 other GENERAL products you will find the right control for practically any application. The General Controls Co. will gladly help solve any specific problem. Your correspondence is invited.

K-10 Magnetic Valve—For controlling oil, water, air, gas. A two-wire, current-failure magnetic valve. Its lever action closing with the flow, insures a tight shut-off indefinitely. Will operate at high pressures on either fluids or gases. As in all other GENERAL CONTROLS, the K-10 is entirely free from A.C. hum.

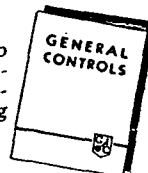


B-50 Slow Opening Valve

B-50 Slow-Opening Gas Valve—Electrically operated diaphragm valve having an adjustable opening time of from 5 to 60 seconds. Ample damper power even at low pressures. Uniflow valve guide provides uniform increase in gas flow. Current consumption 8 watts.

Send for New Catalogue

This catalog will be valuable to every engineer. Showing the complete line of GENERAL CONTROLS, it also gives engineering data and wiring diagrams. Write us. Your copy will be sent by return mail.



The Mercoid Corporation

SOLE MANUFACTURERS OF THE MERCOID SWITCH

CHICAGO, ILL.,
4201 Belmont Avenue.

NEW YORK, N. Y.,
90 West Street.

SAN FRANCISCO, CALIF.,
1129 Folsom Street.

Distributed and Stocked in Many Cities by the Graybar Electric Co., Inc.

COMPLETE LINE OF AUTOMATIC CONTROLS

Mercoid Controls have a wide application in both the domestic and industrial fields. They are generally employed for the control of motor-driven automatic heating, refrigeration and air conditioning equipment, electric space or tank heaters, and for numerous industrial applications. Equipped throughout with Mercoid sealed mercury contact switches.

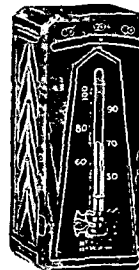
Complete Catalog Sent Upon Request

SENSATHERM

A sensitive air type thermostat for low or high voltage requirements. Used for controlling automatic heating, refrigeration, or air conditioning equipment.

Furnished standard for low voltage to operate with the Mercoid Transformer - Relay. It can directly handle 20 watts or less on 110 or 220 volts without a relay. The standard range is 55 to 85 deg F. Other ranges are available. Total differential, 1 deg F (plus or minus 1/2 deg).

Dual Sensatherms available for day and night temperature regulation, also for heating and cooling operations.



TYPE V TRANSFORMER-RELAY

A low voltage mercury contact relay, which also acts as a transformer inducing low voltage (24 volts) on the pilot circuit. Its construction does away with all hum and chatter.

Available for 110 or 220 volts, 60, 50 or 25 cycle. Can also be furnished as a repulsion-relay for 110 or 220 volts d-c.

MERCOID COMBINED PRESSURE AND LOW WATER CONTROL

TYPE DA-121

The Type DA-121 Mercoid Low Water and Pressure Control with the new double



adjustment feature, is designed primarily for the protection of automatically fired steam boilers, to prevent the hazard of firing into dry boilers and also guard against building up excessive steam pressure.

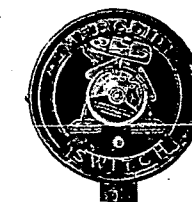
MERCOID STOKATHERM

The Stokatherm System operates the stoker for only the very shortest periods necessary to maintain a minimum fire while the thermostat is in the off position. The firing periods are not at definitely timed intervals or governed by fixed stack temperatures, but are automatically regulated to meet the variable conditions encountered on each installation. If the fuel delivered to the stoker should have a higher or lower burning rate than that formerly used, the Stokatherm System automatically compensates for its firing period accordingly. Should the fuel become exhausted or the fire go out, the Stokatherm System automatically stops the stoker. This prevents operating an empty stoker and guards against filling the combustion chamber with unburned fuel.

Warm Air Furnace Controls—Type M-51 is used to protect warm air furnaces from overheating where automatic heating equipment is used. Type M-53 is to control fans or blowers to prevent operation unless the furnace is warm enough to deliver heat.

Adjustable mounting flange and a graduated temperature dial is provided. Also furnished with long bi-metal stems for bake-ovens and industrial applications.

Adjustable mounting flange and a graduated temperature dial is provided. Also furnished with long bi-metal stems for bake-ovens and industrial applications.



Minneapolis-Honeywell Regulator Company

Executive Offices: 2711 Fourth Ave. So., Minneapolis

Factories: MINNEAPOLIS MINN., AND WABASH, IND.

Branch and Distributing Offices:

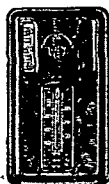
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ALBANY	DALLAS	INDIANAPOLIS*	OKLAHOMA CITY	SCRANTON
ATLANTA*	DENVER	JACKSON, MICH.	OMAHA*	SEATTLE
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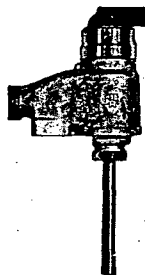
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THE MODUTROL SYSTEM OF AUTOMATIC CONTROL

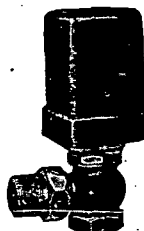
for Heating, Ventilating and Air Conditioning



Room Type Thermostat



The Modulator



Electric Motor Radiator Valve

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. Controllers of various types actuated by positive means and, in turn, connected to electric power units governing valves, dampers, etc., present a new conception of dependability and ease of installation not heretofore possible. In the **Modutrol System** the well-known dependability of electric switches and motors, the permanency of electric wiring, the flexibility of self-contained units and the accessibility of each and every portion of the equipment represent just a few of many advantages offered in this modern means of air conditioning control.

PROVIDES TRUE MODULATION

Viewed from the performance standpoint, the **Modutrol System** offers true modulation of the temperature, air flow and humidity. True modulation means automatic proportioning of heat, cold, air flow or humidity in exactly the required amount to offset changes in the heat loss or relative humidity.

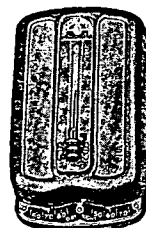
INCLUDES SUPPLEMENTARY EQUIPMENT

Supplementing the modulating controls is a wide variety of on-and-off or two-position motors, controllers and valves, thus making the **Modutrol System** extremely flexible as to the selection of control equipment to produce the desired results.

APPLICABLE TO ANY FORM OF AIR CONDITIONING OR HEATING

Any form of air conditioning or heating, from systems for the home to systems for the skyscraper can be controlled by the **Modutrol System**. While the principle of the **Modutrol System** in its use of electrical or self-contained units differs from the principles in vogue in years past, there is nothing untried or unproven in either design or application.

See also Page 930



Twin Thermostat

RESULT OF A HALF CENTURY'S EXPERIENCE

Fifty years of concentration upon temperature control problems has built up within the Minneapolis-Honeywell organization the knowledge, technique and engineering ability which have produced the **Modutrol System** as well as the comparatively simple domestic control systems.

REDUCES COST OF HEATING OR AIR CONDITIONING SYSTEM OPERATION

In new buildings or old, a lowering of operating costs, and an immense improvement in the comfort delivered, can be accomplished by considering the control system as an integral part of the heating system. In fact, automatic control systems, today, cannot be considered as accessories to the heating systems.

ENGINEERING SERVICE

The Minneapolis-Honeywell Automatic Control engineer is at your service at all times. He will be glad to furnish you with recommended control layouts and cost estimates.

He is trained to recommend control results before installation of equipment and to produce control results after the installation of equipment has been completed.

COMPLETE INSTALLATION SERVICE

Minneapolis-Honeywell Branch Offices and Distributors are equipped to make the complete installation of the **Modutrol System** for control of Air Conditioning or Heating installations. Thoroughly trained men are also available to supervise, adjust, or service the control equipment if the purchaser prefers to make his own installation of controls.

RESPONSIBILITY FOR ENTIRE CONTROL SYSTEM

The **Modutrol System** and its supplementary equipment is so complete that the Minneapolis-Honeywell Regulator Co. is equipped to assume the entire responsibility for any control installation, thereby eliminating the difficulties and misunderstandings which the division of responsibility may create.

TYPICAL SPECIFICATIONS

A complete set of typical specifications covering automatic control systems for use in heating, ventilating, and air conditioning is available for use by engineers and architects.

DESCRIPTIVE LITERATURE

Catalogs arranged according to the following subdivisions are available upon request:

1. The **Modutrol System**.
2. Air Conditioning Controls.
3. Oil Burner Controls.
4. Gas Heating Controls.
5. Stoker Controls.
6. Industrial Regulator Controls.
7. Refrigeration Controls.
8. Complete Condensed Catalog.

In addition to this literature, complete data sheets are available, including technical information on all equipment in the **Modutrol System** which is of prime interest to the engineer. These pages will be furnished on request.

TYPICAL MODUTROL SYSTEMS

The Modutrol System is composed of many individual units of control which may be used separately or in any combination properly coordinated to produce the required results.

The Modutrol Systems outlined on the succeeding pages are only a few of the many types available but will serve to illustrate the extreme flexibility of the Modutrol System and its adaptability to the needs of the individual installation.

ZONE CONTROL OF HEATING SYSTEMS

It is widely recognized that proper temperature control in larger commercial buildings, stores, office buildings, churches, schools, factories and warehouses, and similar structures must be based on heating needs of various parts of the building, and that these needs vary due to different conditions in parts of the same structure. North exposures require more heat than south exposures—similarly those parts of the building subject to the wind effect need more heat than those not so exposed—and where the sun shines on a part of the building less heat is needed than is required in parts not subject to the heat given off by the sun.

The Modutrol System makes provision for temperature control which takes these factors and others into proper consideration. Thus each building may be divided into suitable zones based on exposure, occupancy and time requirement and the delivery of heat is adjusted to the requirements of each zone. Modutrol Zone Control prevents temperature fluctuations and thus prevents wasteful delivery of steam where and when it is not needed yet maintains comfort conditions for the occupants. Modutrol System Control installed in either new or old buildings throughout the nation have a notable record in the reduction of heating costs—many installations saving in excess of 20 per cent of the annual fuel bill.

THE WEATHERSTAT

The combined effect of all four outside weather factors which are temperature, wind direction, wind velocity, and solar radiation determines the heat loss and consequently the heat requirements for any building.

The Weatherstat is the outside control which responds to all of these four weather factors and the reaction of the Weatherstat to these outside conditions enables it to immediately compensate for outside changes by supplying more or less heat as required to the building or zone it controls. This results in high efficiency and economy of operation.

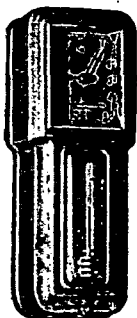
The fuel savings it effects make the Weatherstat an ideal medium for modernization as well as for new buildings because an investment in the Weatherstat is self-liquidating, and insures the benefits of tenant satisfaction.

The Weatherstat consists of a mass of iron shaped to contain a thermostatic element with the necessary electric contacts, and is arranged to house a small electrical heating element within the mass.

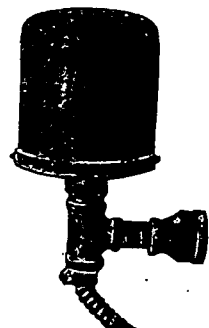
The heat-release from the electrical heating element is designed to, and by an adjustable resistance can be made to, bear the same relation to heat losses from the control housing as the heat release from the heating fixtures within the building bears to heat losses of the building.

Being subject to the same weather influences, and the same relative rate of heating and cooling, the temperature throughout the zone or building and the temperature within the control housing will maintain a fixed relationship, and the temperature in the control housing can be used as the pilot temperature for the zone or building.

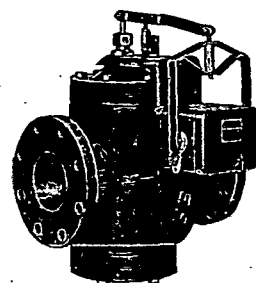
In reality, therefore, the Weatherstat is equivalent to a room out-of-doors, with the same heat loss ratio as the zone or building which it controls.



Chronotherm



The Weatherstat



Motorized Zone Valve

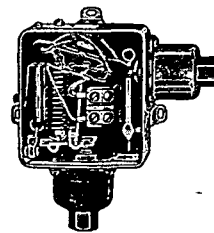
UNIT HEATER CONTROL

The nature of the unit heater method of delivering heat makes automatic controls an absolute essential if any degree of satisfaction and economy of operation is to be obtained.

Minneapolis-Honeywell controls meet the requirements of all types of installations. The simpler of these is satisfied by the mercury switch type of line voltage thermostat, while complete and precise control can be obtained through the use of low voltage thermostats, limiting devices, motor valves, and relays. These complete systems of control so govern the unit heater as to make it not only completely automatic, but highly satisfactory and decidedly economical in operation.



Line Voltage Thermostat



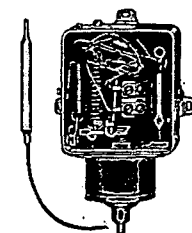
Refrigeration Pressure Control



Unifan Control

REFRIGERATION CONTROLS

A complete line of refrigeration controls of pressure and temperature type are available for the control of refrigeration equipment. Thermostats of the self-contained and remote type with a wide variety of temperature ranges offer solution to any refrigeration temperature control problem. The scope of pressure ranges in the pressure controller embraces any problem of refrigeration pressure control.



Refrigeration Temperature Control

UNIT VENTILATOR CONTROL

The desirable heating and ventilating properties of the unit ventilator are greatly enhanced by the Modutrol System of automatic control applied to the heating coils, mixing and/or fresh air dampers. All makes of unit ventilators can be purchased with Modutrol equipment incorporated or this system of control may be applied to new or old installations in the field. Complete sets of Modutrol Motors, temperature controllers and damper linkage plus steam valves where necessary are available. The economy of operation under the Modutrol System is a distinct advantage in addition to the healthful comfortable temperature control result which it produces. Engineering assistance in the selection and installation of the Modutrol System to unit ventilators is available at all Minneapolis-Honeywell offices.

The typical unit ventilator shown at the right is provided with control for final room temperature through the modulation of mixing dampers actuated by a Modutrol Motor under influence of a modulating room thermostat. The intake damper actuated by a two position Modutrol Motor opens to admit outdoor air on starting of the unit ventilator fan and moves to the closed or recirculation position on interruption of current to the fan motor. Often the intake dampers of a multiple unit ventilator installation are controlled in groups from manual switches on a central control panel either simultaneously with or independent of fan motor operation.

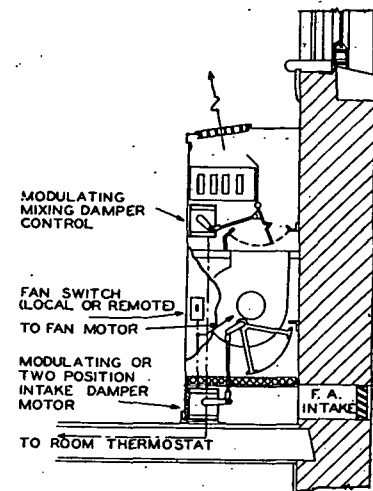


FIGURE 1

CONTROL OF CENTRAL FAN SYSTEMS

Modutrol System Controls can be selected for and applied to any of the innumerable arrangements of central fan systems from the simple system used only for ventilation to the most complex air conditioning system involving all varieties of preheaters, reheaters and booster heaters, fresh air, recirculated air and mixed air ducts and dampers, face and by-pass dampers, humidifiers and dehumidifiers, coil or spray coolers, etc. Control may be applied in almost any degree of automatic operation from various controllers of the thermostatic room or duct type, humidity sensitive type or those actuated by pressure or time.

In these systems electric motorized or Modutrol Valves or Modutrol Motors actuating dampers respond to the demands of the controllers to which they are electrically wired. Each motorized valve or damper motor is selected for its particular function and is a self-contained unit. The wide selection of equipment in the **Modutrol System** insures the ability of the central fan performance in a satisfactory manner and yet provides extreme flexibility and minimum cost in both installation and maintenance while providing the inherent dependability of electrical equipment. In either new construction or old the inter-connection of the controllers and the controlled equipment is simplified to the running of electrical wiring the majority of which is low voltage.

CONTROL OF AIR CONDITIONING SYSTEMS

Whether air conditioning is accomplished in unit conditioners or central fan conditioning systems, the **Modutrol System** will provide an accurate and reliable control.

When required the **Modutrol System** provides for automatically adjusting the temperature of the cooler area in relation to outdoor temperature, and for the modulation of outdoor air and return air dampers so as to use outdoor air for either heating or cooling whenever possible.

The following typical examples will illustrate the flexibility of the **Modutrol System** in its application to air conditioning systems.

Figure 2 illustrates a system of controls for a typical air conditioning system. Operation of the controls is as follows:

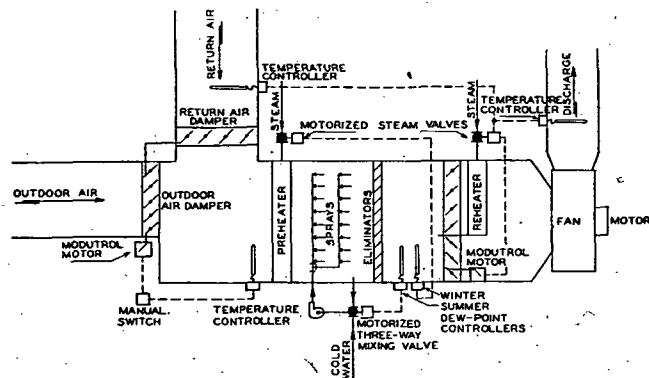


FIGURE 2

1. In winter the temperature controller at the intake of the preheater is set at some temperature control point above freezing and regulates the **Modutrol Motor** which in turn determines the relative positions of the outdoor and return air dampers so as to deliver air to the conditioner at this predetermined temperature. A manual switch provides for setting the dampers at any desired position in summer and also in winter if preferred.

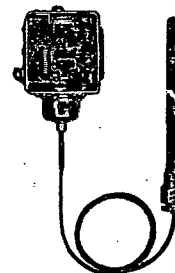
2. The winter dew point controller regulates the flow of steam to the preheater through the operation of the **Modutrol Motorized Valve** so as to deliver air from the sprays at a predetermined winter dew point temperature. In winter the **Motorized Three-Way Mixing Valve** will be automatically placed in the position which will provide all recirculated water and no cold water to the sprays.

3. The summer dew point controller is set at a higher temperature control point than the winter dew point controller. This arrangement, therefore, provides for automatic change-over as the load changes from heating to cooling, or from cooling to heating.

The summer dew point controller regulates the proportions of recirculated and cold water to the sprays through the operation of the **Motorized Three-Way Mixing Valve** so as to deliver air from the sprays at the predetermined summer dew point temperature.

4. The temperature controller in the return air regulates the position of the reheater face and by-pass dampers delivering more or less air through the reheater coils as the heat requirement increases or decreases. The steam flow to the reheater coil is modulated by a **Modutrol Motor** operated by a dual control switch on the **Modutrol Motor** operating the face and by-pass dampers. This combination provides an accurate and reliable control of the return air temperature and the temperature controller in the delivered air prevents the air at this point from falling below a predetermined temperature.

COMPENSATED AUTOMATIC CONTROL



Compensating Controller

The demand for increased efficiency in the operation of Air Conditioning and Heating Systems has imposed new functions upon the component parts of these systems. Compensated control of temperature, relative humidity, pressure, and combinations of these factors provide these new functions, and the ease and accuracy with which it is accomplished by the **Modutrol System** has made its use a practical necessity.

The following examples of compensated controls illustrate a few of the many possible variations:

1. Compensated Dry-Bulb Control.

The outdoor temperature is measured and automatically determines the inside dry-bulb temperature control according to any predetermined schedule.

Compensated dry-bulb temperature control is accomplished by the use of an outdoor compensator and an inside return air temperature controller or room thermostat which in turn operates the conditioning equipment.

2. Effective Temperature Control.

The indoor relative humidity is measured and automatically determines the inside dry-bulb temperature control point according to a predetermined schedule which will result in a fixed effective temperature.

Effective temperature control is accomplished by the use of an inside relative humidity compensator and an inside return air temperature controller or room thermostat which in turn operates the conditioning equipment.

3. Compensated Effective Temperature Control.

The outdoor temperature is measured and automatically determines the inside effective temperature control point. The inside relative humidity is then measured and automatically determines the inside dry-bulb temperature control point. Both of these determinations are made according to any predetermined schedule.

Compensated effective temperature control is accomplished by the use of an outdoor temperature compensator, an inside relative humidity compensator and a return air temperature controller or room thermostat which in turn operates the conditioning equipment.

4. Compensated Relative Humidity Control.

The outdoor temperature is measured and automatically determines the inside relative humidity control point in order to prevent frosting on the windows according to any predetermined schedule.

Compensated relative humidity control is accomplished by the use of an outdoor compensator and an inside relative humidity controller which in turn operates the means for adding moisture to the controlled space.

CONTROLS FOR RESIDENCES

The need for reliable and accurate temperature control systems for residential heating plants is greater today than ever before. This type of heating plant has developed from the simple system of yesterday to the complex one of today, involving automatic firing and in many cases all year air conditioning equipment. Successful performance of such a heating system depends upon the coordination of its several functions into a definite program of operation. Occupancy comfort, therefore, depends upon properly synchronized control that the home owner has neither the skill nor the patience to give. The home owner must of necessity depend more and more upon an adequate control system for safe and economical operation of his heating system.

For the small manually fired coal heating plants, there are simple damper control sets of thermostat, limit control and damper motor. For the more elaborate heating systems, there are control systems that coordinate functions of heating, air circulation, humidification and cooling into a definite program of operation. For the large heating systems there are control systems for zone control, humidity control and compensated temperature control by which the inside temperature factors are matched against the outside weather variables. Supplementary controls are available for existing control systems that increase "user satisfaction" and operate the heating equipment with increased economy, safety and convenience.

THE POWERS REGULATOR CO.

45 Years of Temperature and Humidity Control

Offices in 43 Cities—See Your Phone Directory

General Offices and Factory: 2719 Greenview Ave., CHICAGO;

General Eastern Office: 231 East 46th Street, NEW YORK;

1808 W. Eighth Street, LOS ANGELES;

The Canadian Powers Regulator Co., 106 Lombard St., TORONTO, ONT.

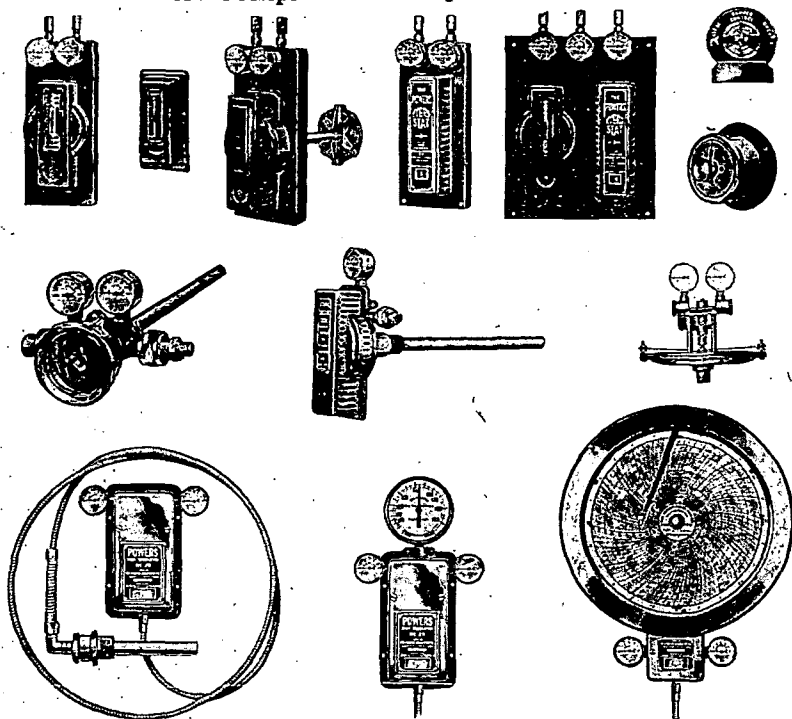
PRODUCTS—A very complete line of compressed air operated and self-operating temperature, humidity and air flow controls for automatically regulating heating, cooling, ventilating and air conditioning systems and industrial processes.

A complete line of self-operating and compressed air operated valves and regulators made for: Controlling

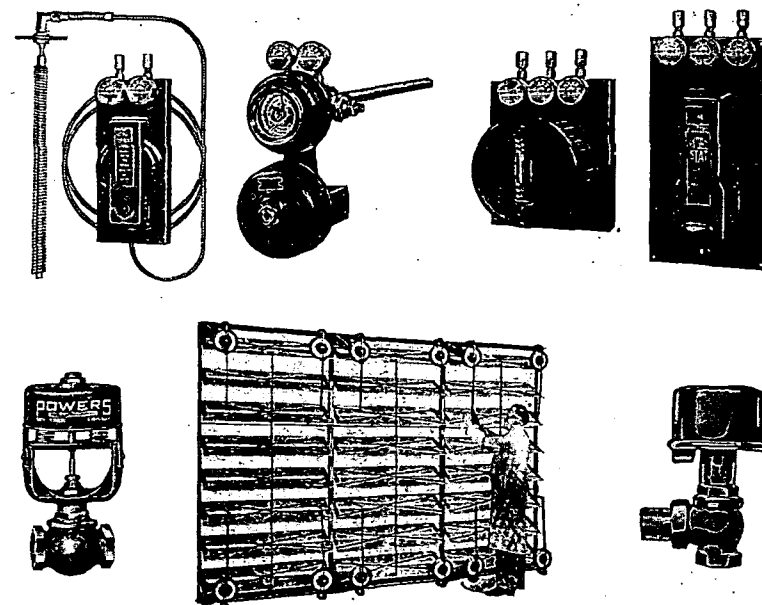
steam heated hot water heaters, and submerged type heaters; and for automatically mixing hot and cold water or steam and cold water delivering a mixture at a predetermined temperature.

Dial Indicating and Recording Thermometers. High pressure steam traps and pressure reducing valves.

Powers Compressed Air Operated Apparatus



Powers Compressed Air Operated Apparatus



POWERS ENGINEERING SERVICE

As the accurate performance of a heating, cooling, ventilating or an air conditioning system, or an industrial process is so dependent upon its automatic control equipment, and as the cost of such control is but a fraction of the entire system, the use of the proper type of regulation is always sound economy.

To secure the maximum return on the investment in automatic control equipment, it is exceedingly important that proper selection of control apparatus be made

when each installation is being planned.

Forty-five years of experience in furnishing and installing temperature and humidity control for every conceivable purpose in all types of buildings has 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.

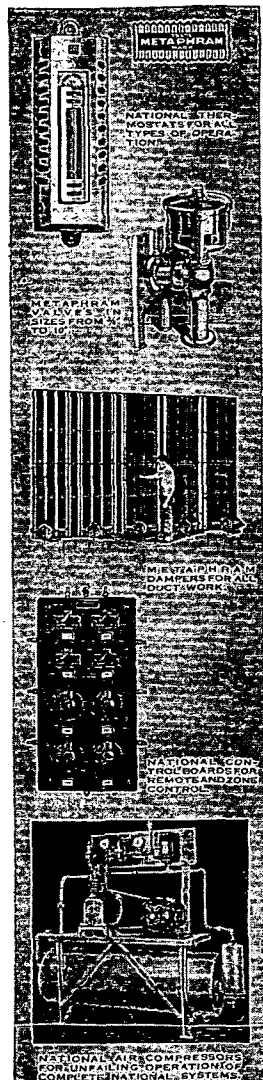
National Regulator Co.

2311 Knox Avenue

Chicago, Illinois

Manufacturers and Contractors

Complete Systems for Control of Temperature, Humidity, Ventilation and Air Conditioning. Metaphram Damper Regulators for Domestic Heating Boilers and Tank Heaters. A-Jacks High Pressure Steam Damper and Combustion Control.



A National Control System for temperature, humidity, ventilation or air conditioning comprises coordinated equipment easily understood by local operating engineers. Each system is planned to meet the requirements of the individual building to assure proper regulation.

National Thermostats (Room Type)—Air operated; simple, two-temperature and compound types for direct radiation, direct radiation and unit ventilating machines, straight blast heating, ventilating fan units and dampers.

National Thermostats (Duct Type)—Air operated; for ventilating duct or blast heating control and hot water tanks.

National Thermostats (Industrial) for control of industrial and process temperatures.

Metaphram Valves—Air operated; for direct radiation, steam lines, hot water tanks, humidifiers and accumulator control.

Metaphram Dampers—Air operated by Metaphram motors; for accurate automatic control of ventilation and blast heating. Built in round, double or louver types, of black or galvanized steel or special metals.

National Control Boards—Located in engineers room or central place for remote or zone control of pneumatic or electric-pneumatic switches in connection with two-temperature and zone control systems under manual or time-clock operation; operating valves or steam lines, radiation, or ventilating fan units and dampers.

National Air Compressors—Self-contained, automatic units for unfailling operation of National apparatus and equipment.

Metaphram Damper Regulators for all domestic hot water or low pressure heating boilers, gas, oil or coal fired.

A-Jacks Control for high pressure boilers (15 lb to 300 lb pressure) giving synchronized control to boiler pressure and fuel consumption.

Catalogs and Bulletins—Thoroughly illustrated bulletins are available on all products. Engineering assistance will be rendered without obligation.

The Dole Valve Company

Main Office and Factory: 1901-1933 Carroll Avenue, Chicago, Ill.

Branch Offices and Representatives

ALBANY, N. Y.
ATLANTA, GA.
BALTIMORE, MD.
BOSTON, MASS.
BUFFALO, N. Y.
BUTTE, MONT.
CINCINNATI, OHIO

CLEVELAND, OHIO
DALLAS, TEXAS
DENVER, COLO.
DETROIT, MICH.
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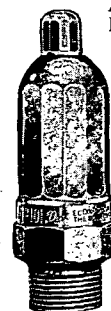
MILWAUKEE, WIS.
MINNEAPOLIS, MINN.
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ST. LOUIS, MO.
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TORONTO, ONT., CANADA
WILKES-BARRE, PA.
WINNIPEG, MAN., CANADA

Dole

DOLE AIR VALVES AND BRASS PRODUCTS
A Complete line of thermostatic and hot water key air valves
Patents have been applied for on all thermostatic type valves



No. 101

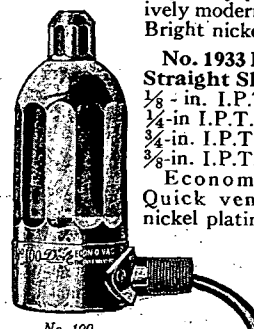
No. 101 Dole Econ-O-Vac Valve: Straight Shank— $\frac{3}{4}$ -in. I. P. T. Male, $\frac{3}{8}$ -in. I. P. T. Female. Bellows operated vacuum seal. Quick venting. Chrome plated—popular price.

No. 100 Dole Econ-O-Vac Valve—Bellows operated vacuum seal. Chrome plated—popular priced.

No. 1933 Dole Air Valve: Angle Type—Popularly priced. Attractively modern in appearance. Bright nickel finish.

No. 1933 Dole Air Valve: Straight Shank— $\frac{1}{2}$ -in. I. P. T. $\frac{3}{4}$ -in. I. P. T. Male, $\frac{3}{8}$ -in. I. P. T. Female.

Economical in cost. Quick venting. Bright nickel plating.



No. 100

No. 1 Dole Syphon Air Valves—Our leading syphon air valves. Sturdy construction. Modern design. Vents freely. Chrome plated.



No. 1933—Angle

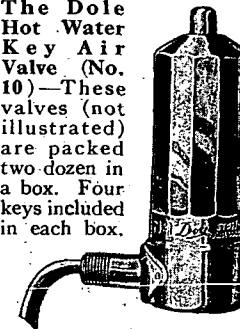


No. 1933—Straight Shank



No. 3C

The Dole Hot Water Key Air Valve (No. 10)—These valves (not illustrated) are packed two dozen in a box. Four keys included in each box.



No. 1



No. 6B

No. 6B Dole Vacuum Valve: Straight Shank— $\frac{3}{4}$ -in. I. P. T. Male, $\frac{3}{8}$ -in. I. P. T. Female.

Our leading quick venting vacuum valve. Bellows operated vacuum seal. Extra quick venting. Chrome plated.

No. 2B Dole Vacuum Valve—Our leading vacuum valve. Fool proof construction. Non-adjustable. Bellows operated vacuum seal. Chrome plated.

No. 3A Dole Quick Vent Air Valve: Straight Shank— $\frac{1}{2}$ -in. I. P. T.

No. 3B Dole Quick Vent Air Valve: Straight Shank— $\frac{1}{4}$ -in. I. P. T.

No. 3C Dole Quick Vent: Straight Shank— $\frac{3}{4}$ -in. I. P. T. Male, $\frac{3}{8}$ -in. I. P. T. Female.

Our leading quick vent valves. Extra large venting ports. Chrome plated. Reasonable prices.

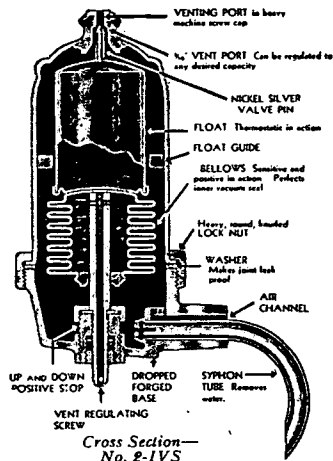
Write for Prices and Discounts

Anderson Manufacturing Co.

Cambridge, Mass.

Manufacturers of VENT-RITE Air Valves
Pioneers of Balanced Radiation—Since 1932

VENT-RITE RADIATOR AIR VALVES



VENT-RITE Valves can be regulated to meet the venting requirements of any radiator . . . By means of an *exclusive type* of variable venting orifice, which *provides an unlimited selection of Vent Ports*, the distribution of steam in one-pipe systems, is *positively controlled*.

VENT-RITE Valves will balance the resistance opposing distribution of steam in a one-pipe job so that there are no lagging radiators. The venting capacity can also be restricted at desired points, to prevent overheating in spots.

When installed on a thermostatically controlled heating system VENT-RITES can be so set as to allow all radiators to receive the desired amount of heat before the thermostat shuts off.

VENT-RITE Valves can be taken apart with the use of an ordinary strap wrench and cleaned. All valves have a conventional union coupling with a heavy gasket which makes the joint leak-proof. The adjustment feature is not conspicuous and will not be tinkered with. A wire-like tool is required to set venting rates, and once set by the heating contractor, they stay set.

VENT-RITE Valves are made in both vacuum and non-vacuum types. The construction of the vacuum type does not depend upon a ball or disc check to maintain vacuum in a system. This is accomplished by a sensitive and positive acting bellows, internally subjected at all times to atmospheric pressure which elongates the bellows, and closes the vent when the internal pressure in the valve is less than the atmospheric pressure. This pressure differential causes the valve to close, thereby perfecting an inner vacuum seal.

VENT-RITE Radiator Air Valves are noiseless in operation, positive in action and seal by float action against water. They close thermostatically under temperature and will not leak or sputter. All parts are made of the best material suitable for valves of this type. The valve pins are of nickel silver. The base is made of a brass drop forging. They are attractively finished in chromium plate.

VENT-RITE Valves are guaranteed against defective workmanship or material.

VENT-RITE BALANCER

The VENT-RITE Balancer is designed for installation on automatically fired steam heating systems equipped with VENT-RITE Vacuum Valves (No. 2-I.V.S.). . . . Used in conjunction with these air valves, it provides a means for controlling the distribution of steam at every cycle . . . It also makes possible the utilization of the advantages of a non-vacuum system plus controlled distribution and the advantages of a vacuum system—and eliminates the objectionable features of both. In the operation of one-pipe systems this "balancing unit" brings about a far greater heating efficiency and fuel economy than has heretofore been possible . . . Essentially, the VENT-RITE Balancer creates an entirely new steam heating system and insures **Balanced Radiation** on every heating cycle.

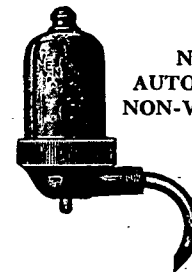
The VENT-RITE Balancer is installed on one of the mains and is electrically connected with the unit operating the steam supply and automatically operates in conjunction with the heat control.

Anderson Manufacturing Co.

Cambridge, Mass.

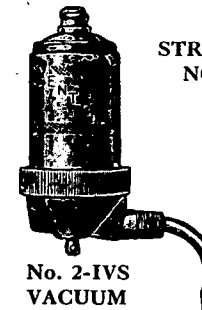
Manufacturers of VENT-RITE Air Valves
Pioneers of Balanced Radiation—Since 1932

THE VENT-RITE LINE



No. 1
AUTOMATIC
NON-VACUUM

1/8 in. side vent for radiators. Variable venting capacity. Come-apart construction. Float seal against water. Closes thermostatically under temperature.



No. 2-IVS
VACUUM

1/8 in. side vent for radiators. Inner Vacuum Seal. Variable venting capacity. Come-apart construction. Float seal against water. Closes thermostatically under temperature.



No. 3
STRAIGHT SHANK
NON-VACUUM

1/4 in. bottom outlet. For risers, mains and convection type radiators. Float seal against water. Variable venting capacity. Come-apart construction. Closes thermostatically under temperature.



No. 4-IVS
VACUUM

1/4 in. bottom outlet. For risers, mains and convection type radiators. Inner Vacuum Seal. Variable venting capacity. Come-apart construction. Float seal against water. Closes thermostatically under temperature.



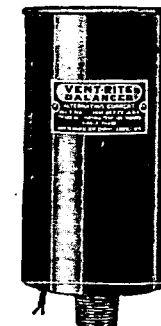
No. 5
STRAIGHT SHANK
NON-VACUUM

1/2 in. bottom outlet. For risers, mains and convection type radiators. Float seal against water. Maximum venting capacity 1/8 in. diameter hole. Come-apart construction. Closes thermostatically under temperature.



No. 6-IVS
VACUUM

1/2 in. bottom outlet for risers, mains and convection type radiators. Inner vacuum seal. Maximum venting capacity 1/8 in. diameter hole. Come-apart construction. Float seal against water. Closes thermostatically under temperature.



VENT-RITE
BALANCER

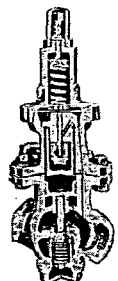
A solenoid operated valve for automatically controlled steam heating systems . . . Provides controlled distribution of heat on every cycle . . . Provides quicker venting on mains . . . Provides complete Btu transfer from heat supply unit to radiators . . . 3/4 in. bottom outlet. For use only on 110 Volts—60 Cycles A.C.

Foster Engineering Co.

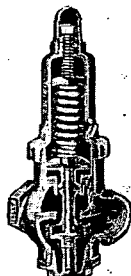
109-113 Monroe Street

Newark, N. J.

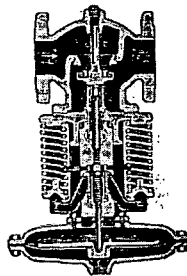
Fifty-six Years Experience in Designing, Manufacturing and Applying Automatic Valves to meet all Classes of Control Service on Steam, Liquids and Gases. Standard and Special Construction



Class 34G
Pressure Regulator



Type 34U
Pressure Regulator



Class 35A1
Pressure Regulator

Class 34G Pressure Regulator—Single bevel-seated (strainer protected), pilot operated, for single-stage reduction on steam or air for initial pressures up to 900 lbs, 750° F T.T. Maintains a constant delivery regardless of fluctuations of initial pressure or volume of flow and is adapted for deadend service. Type 34G8 constructed with large diaphragms for delivery pressure 0 to 15 lbs. Sizes ½ in. to 12 in. Bronze, semi and cast-steel.

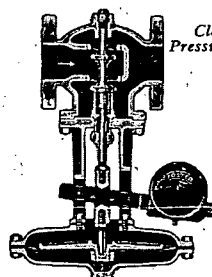
Type 34U Pressure Regulator—Direct Acting Piston Operated for variable initial pressures up to 300 lbs, 160° F T.T. on water or air; adapted for deadend service. Special construction to meet higher pressure. Several types of pilot or direct-acting diaphragm actuated for low delivery pressure and intermittent service. All types have renewable seat discs. Sizes ½ in. to 12 in. Bronze and semi-steel.

Class 35A1 Pressure Regulator—For general service on steam with fairly constant initial pressure up to 250 lbs and for delivery pressure between 1 and 150 lbs. For deadend service. Maintains constant delivery pressure. Sizes ½ in. to 10 in. Bronze, semi or cast-steel body. Note accessibility of springs.

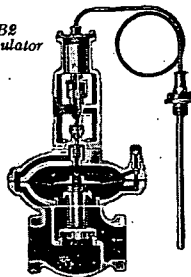
Class 35B2 Pressure Regulator—For service on steam in apartments, office buildings, etc. Initial pressures up to 250 lbs and constant delivery pressures 0 to 50 lbs. Direct acting, double-seated type (not adapted for deadend service), for variable initial pressures and volume. Sizes ½ in. to 10 in. Bronze, semi or cast-steel body.

Temperature Regulator—Pilot operated, self-contained, single-seated regulator for deadend service. Maintains temperatures of gases or liquids within 1° F ±. Pressures 10 to 200 lbs. Sizes ½ in. to 3 in. Larger sizes to 8 in., double-seated. Type 34T2, direct-acting, double-seated (not adapted for deadend service) for 0 to 125 lbs. Sizes ½ in. to 6 in. Bronze and semi-steel body.

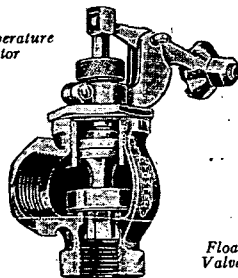
Float Valves—Very sensitive valve for hot or cold water service, being actuated by the pressure in the supply line. Sizes range from ½ in. to 8 in. Angle and Globe. Several types made with piston and stem loosely coupled, adding direct-acting feature. Bronze and semi-steel body.



Class 36B8
Pressure Regulator



34T Temperature
Regulator



Float
Valves

Jenkins Bros

BRONZE - IRON - STEEL VALVES - SINCE 1864

Mechanical Rubber Goods

Principal Stores and Offices: 80 White Street, New York, N. Y.; 524 Atlantic Avenue, Boston, Mass.; 133 N. Seventh Street, Philadelphia, Pa.; 822 Washington Boulevard, Chicago, Ill.; 510 Main Street, Bridgeport, Conn.; (Office and Factory). JENKINS BROS., LIMITED: MONTREAL, QUE., (Works, Head Office); LONDON, W.C. 2



Fig. 106
A Bronze Globe,
Renewable Comp. Disc



Fig. 950
Bronze Globe,
Re grind-Renew



Fig. 148
Iron Body Globe



Fig. 570
Bronze Gate



Fig. 525
Iron Body Gate

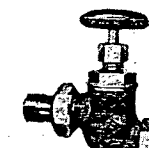


Fig. 850
Radiator Offset
Globe

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 300 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.

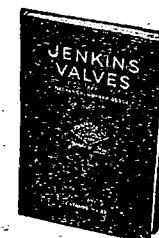
General Classifications of Jenkins Valves Include—Bronze Valves fitted with Jenkins renewable composition disc. Bronze Re grind-Renew Valves with bevel and plug type seats. Bronze Gate Valves. Iron Body Valves fitted with Jenkins renewable composition disc. Iron Body Re grinding Valves. Iron Body Gate Valves with solid wedge and double disc parallel seats. All-Iron Valves. Cast Steel Gate Valves and Swing Check Valves. Electrically and Hydraulically Operated Gate Valves. Radiator Valves.

Fire Line Valves. Quick-opening and Self-closing Valves, Needle Valves, Y Valves.

Other Jenkins Products Are—Colored Valve Wheels with or without service markings molded in relief letters. Composition Valve Discs exactly suited to service conditions. Sheet Packing. Gaskets. Moncrieff Scotch Gage Glasses.

CONSULT THIS HELPFUL BOOK

This 307 page Jenkins Catalog not only gives complete details on over 500 Jenkins Valves, but also it has a large section of engineering data and practical information about valves and layouts. Make sure you have a copy, including the new Supplement "B."



JENKINS VALVES ARE SOLD BY MOST GOOD SUPPLY HOUSES

New York Air Valve Corporation

NYAVCO

CHICAGO
BOSTON

Since 1898
DETROIT
ST. LOUIS

CLEVELAND
PITTSBURGH

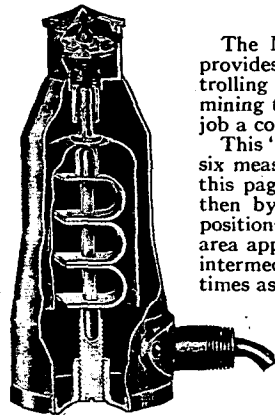
NYAVCO

Orifice Control
Air Valves

611-621 Broadway, New York

Vacuum Type Orifice
Control Air Valves

"CONTROL BY VENTING"



The NYAVCO Orifice Control Air Valve herewith illustrated provides through metered orifice disc an accurate medium for controlling the speed of air elimination from radiators—thus determining the volume of entering steam; making any one pipe steam job a controlled heat job.

This "Control by Venting" feature is secured by using any of the six measured positions on the disc—as indicated in the sketch on this page—from the full opening as noted on the 0.096 position, then by graduations of 10/1000ths of an inch through 5 other positions to 45/1000ths. The full opening or 0.096 position has an area approximately 5 times as great as the 0.045 position and the intermediate sizes have areas which are roughly 4-3-2 and 1½ times as great.

On intermittent heating (Oil Burner, Stoker, Gas fired Boiler) installations the valve is virtually a necessity as it assures a proportionately equal steam distribution regardless of size of radiator or its distance from the boiler.

With proper setting of the control disc the largest or most distant and the smallest and nearest radiator will heat simultaneously. The valve cap locks the disc in after adjustment is secured—positively preventing tampering.

No sealed floats—Note the bimetallic construction of the operating parts which preclude the possibilities of injury to the factory adjustment. The open inner float prevents ejection of water into room in case of flooding.

NYAVCO, distinct in its own field—does not have a sealed float and the highest pressure possible cannot injure the working parts of the valve.

Due to its shock proof construction and its "control by venting" feature it is invaluable on convector type radiators.

Made in both venting and vacuum types.

All NYAVCO Valves are also Made in

Vacuum Type

NYAVCO is Also Made in

½ in. Straight Male; ¼ in. Straight Male for risers, coils, direct and indirect radiation.
All ¾ in. valves have ¾ in. female tapping.

½ in. Straight Male for convector radiation, unit heaters, blast heaters, etc.
¾ in. Straight Male for quick vents, unit heaters, blast heaters, etc.



DISC SHOWING 6 DIFFERENT POSITIONS, FROM .45/1000THS PARTIAL OPENING, TO .96/1000THS COMPLETE OPENING

The Burt Mfg. Co.

VENTILATORS - OIL FILTERS - EXHAUST HEADS
AKRON, OHIO, U. S. A.
Representatives in All Principal Cities

Burt "Forst-Aire" Fan Ventilators



Dimensions — Weights — Capacities

Size Inches	Ga. Galv.	Oz. Copper	Net Wt. Lbs.	Overall Dia. Inches	Overall Height In.*	Motor		Free Air Capacity C.F.M.
						H.P.	R.P.M.	
12	22	16	82	21	31 1/4	1/10	1725	690
14	22	16	99	25 1/2	35	1/10	1725	1200
16	22	16	104	27 1/2	36	1/10	1725	1850
18	20	18	125	30	38 1/2	1/10	1725	2440
20	20	18	151	34 1/2	43	1/6	1140	2990
24	20	18	170	39	50	1/6	1140	4300
30	18	20	347	52 1/2	58	1/3	860	6350
36	18	24	483	60	64	1/2	860	10600
42	18	24	625	69	72	3/4	675	11130
48	18	24	798	78 1/2	80	3/4	675	14500

*Exclusive of base.

Head—High Efficiency, Low Resistance.

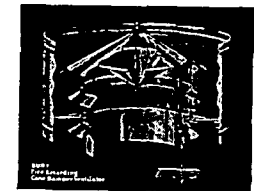
Motor Mounting—Rubber Cradled—Secure.

Types—Standard—Heat Resisting—Acid Resisting—Static Pressure.

Motors—Totally Enclosed—Fan Duty—Vertical, Ball Bearing—Class 1, Group D optional, Single or Multi-Speed, 110 V., 1 Ph., Standard on ½ and ¾ H.P.—220 V., 3 Ph. on larger sizes. Other Power Characteristics optional at slight delivery delay.

Burt High Efficiency Cone Damper Ventilator

An examination of the following typical capacities will indicate that this unit type gravity ventilator has been engineered to move large quantities of air under all conditions. Large, unrestricted outlet openings in the head, the diameter and height of windband and the inverted Cone Camper are factors contributing to its efficiency. When used with patented Spring Clip and Fusible Link is approved by Underwriter's Laboratories, Inc.



Dimensions — Weights — Capacities

Size—Inches	8	10	12	14	16	18	20	24	30	36	42	48	54	60	66	72
Ga. Galv.	22	22	22	22	22	20	20	20	18	18	18	18	18	16	16	16
Oz. Copper	16	16	16	16	16	18	18	18	20	24	24	24	28	28	28	32
Net Wt.—Lbs.	11	15	22	28	37	51	60	74	116	218	290	385	472	680	865	1002
O.A. Dia.—In.	14 1/2	18 1/2	21	25 1/2	27 1/2	30	34 1/2	39	52 1/2	60	69	78 1/2	89	99	109 1/2	118 1/2
O.A. Ht.—In.	14 1/2	16 1/2	19 1/2	23	24	26 1/2	28	35	40	46	54	62	71	80	88	94
Cap.—C.F.M.*	255	392	575	774	1015	1270	1540	2180	3280	4440	5700	6950	8460	9260	11250	13400

*Temp. diff. 20 deg. F. Stack ht. 40 ft. Wind velocity 10 m.p.h. From tests of actual operating conditions by Prof. G. L. Tuve, Case School of Applied Science, Cleveland, Ohio.

Other Burt Ventilators

Sliding Sleeve Damper Unit—Metal or Glass Top—a moderate priced gravity unit of exceptional quality.

Ball Bearing Revolving Ventilator—Accurately balanced, well braced, double bearing unit of high capacity.

Monovent Continuous Ridge Ventilator—designed to "take the roof off." Ten foot sections fit together perfectly to give as long an outlet as desired. Vertical operating damper with fully protected lifting mechanism.

Burt Round Louvre Dampers

Can be fitted into any Burt Ventilator where an extremely close fitting damper is needed. Edges felted if required. Well balanced and easy operating in either hand

or damper motor control. Latter available for manual or thermostatic operation to suit requirements of particular job.

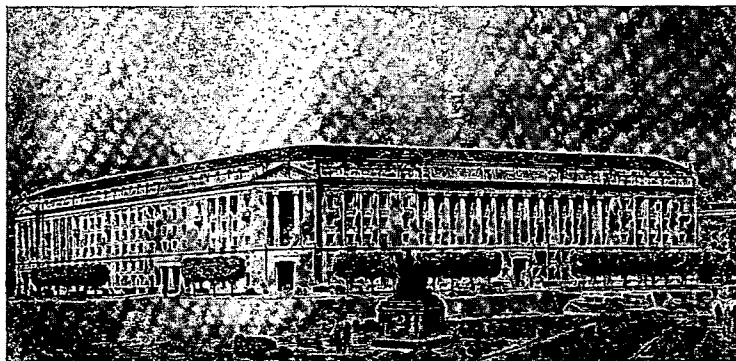
Write for Burt Handbook of Ventilation Data

Ferro-Nil Corporation

500 Fifth Avenue

New York City

WATER CONDITIONING FOR AIR CONDITIONING



New Department of Justice Building, Washington, D. C.

All water used for washing air picks up corrosive matter from the air. The corrosive matter may be oxygen, carbon dioxide, sulphur tri-oxide or sulphur di-oxide. In air conditioning systems where water is recirculated, the amount of acidic material contained in the water may reach very high levels. This acid water will corrode metallic surfaces with which it comes in contact. In severe cases equipment has been known to fail after three months operation due to corrosive action of water used.

Ferro-Nil Service by maintaining water in a non-corrosive condition, prevents corrosion. Corrosion¹ causes high rates of depreciation. Ferro-Nil Service by maintaining metallic surfaces free from rust and scale affords optimum heat transfer efficiencies at all times.

We number among our clients, nationally known corporations such as:—

Rockefeller Center
National Broadcasting Company
Columbia Broadcasting Company
Runkel Chocolate Company
Metropolitan Life Insurance Company
U. S. Government

SUBSTANTIAL DOLLAR SAVINGS GUARANTEED

Let us submit estimate of cost of Ferro-Nil Service for your air conditioning equipment.

The illustration depicts one of the many fine buildings where Ferro-Nil Service is used.

¹See discussion of Corrosion Chapter 34/

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 United States Radiator Corporation, 874
 Utica Radiator Corporation, 1000
 Warren Webster & Company, 1028-1030
 Young Radiator Co., 914

COOLING EQUIPMENT, Air

Aeriet Air Conditioner Company, 799
 Aerofin Corporation, 915-917
 Air Controls, Inc., 834
 Airtemp Incorporated, 800

American Blower Corporation, 802-803
 Baker Ice Machine Co., Inc., 823
 Betz Unit Air Cooler Co., 804
 Buckeye Blower Company, 922-923
 Buffalo Forge Company, 901
 E. K. Campbell Heating Co., 909
 Canadian Sirocco Co., Ltd., 802-803
 Carbondale Machine Corporation, 824
 Carrier Engineering Corporation, 825

Clarge Fan Company, 805
 Fedders Manufacturing Co., 911
 Frick Company (Incorporated), 827
 Frigidaire Corporation, 806-807
 General Electric Company, 838-839, 968-969
 Henry Furnace & Foundry Co., 843, 896
 Ilg Electric Ventilating Company, 903

Ingersoll-Rand Company, 828-829
 Kelvinator Corporation, 808-811
 McQuay, Incorporated, 812
 Modine Manufacturing Co., 912
 John J. Nesbitt, Inc., 922-923
 Niagara Blower Company, 814-815
 Norge Division, Borg-Warner Corporation, 830, 847, 892
 Savage Arms Corporation, 816
 Servel, Inc., 817
 Thermal Units Manufacturing Company, 818
 Trane Company, The, 1027
 Unit Heater and Cooler Co., The, 913

Vilter Manufacturing Company, The, 832
 Westinghouse Electric & Manufacturing Co., 819, 970
 Wittenmeier Machinery Company, 833
 York Ice Machinery Corporation, 820
 Young Radiator Company, 914

COOLING EQUIPMENT, Oil

Aerofin Corporation, 915-917
 Carbondale Machine Corporation, 824
 Frick Company (Incorporated), 827
 G & O Manufacturing Company, The, 918
 Niagara Blower Company, 814-815
 Unit Heater and Cooler Co., The, 913
 Williams Oil-O-Matic Heating Corporation, 848, 893
 York Ice Machinery Corporation, 820
 Young Radiator Company, 914

COOLING EQUIPMENT, Water

Aeriet Air Conditioner Company, 799
 Aerofin Corporation, 915-917
 American Blower Corporation, 802-803
 Baker Ice Machine Co., Inc., 823
 Buffalo Forge Company, 901
 Canadian Sirocco Co., Ltd., 802-803
 Carbondale Machine Corporation, 824
 Frick Company (Incorporated), 827
 Frigidaire Corporation, 806-807
 McQuay, Incorporated, 812
 Niagara Blower Company, 814-815
 Norge Division, Borg-Warner Corporation, 830, 847, 892
 Research Corporation, 821
 Thermal Units Manufacturing Company, 818
 Trane Company, The, 1027
 Unit Heater and Cooler Co., The, 913

Vilter Manufacturing Company, The, 832
 Wittenmeier Machinery Company, 833
 York Ice Machinery Corporation, 820
 Young Radiator Company, 914

CORROSION, Treatment of Aquatic Chemical & Metallurgical Engineers, 860

Ferro-Nil Corporation, 1060

COVERING, Ammonia Pipe

Armstrong Cork Products Co., 940
 Baker Ice Machine Co., Inc., 823
 Eagle-Picher Lead Company, The, 941
 Ehret Magnesia Manufacturing Co., 946
 Johns-Manville, 950-951
 Mundet Cork Corp., 952
 Ruberoid Co., The, 956-957
 Zonolite Corporation, 960-961

COVERING, Pipe

Alfot Insulation Co., 942-943
 Armstrong Cork Products Company, 940
 Baker Ice Machine Co., Inc., 823
 Eagle-Picher Lead Company, The, 941
 Ehret Magnesia Manufacturing Co., 946
 Frick Company (Incorporated), 827
 Johns-Manville, 950-951
 Mundet Cork Corp., 952
 Owens-Illinois Glass Company, 854
 Ric-wil Company, The, 963
 Ruberoid Co., The, 956-957
 Standard Lime & Stone Co., The, 958
 Wilson & Co., 962
 Zonolite Corporation, 960-961

COVERING, Surfaces

Alfot Insulation Co., 942-943
 Armstrong Cork Products Company, 940
 Baker Ice Machine Co., Inc., 823
 Ehret Magnesia Manufacturing Co., 946
 Johns-Manville, 950-951
 Mundet Cork Corp., 952
 Owens-Illinois Glass Company, 854
 Ruberoid Co., The, 956-957
 Wilson & Co., 962
 Zonolite Corporation, 960-961

CUTTING AND WELDING APPARATUS (See Welding and Cutting Apparatus)**DAMPER REGULATORS, Boiler (See also Regulators)**

American Radiator Company, 801, 861-865
 Barber-Colman Company, 1034-1035
 Detroit Lubricator Company, 1036
 C. A. Dunham Company, 1012-1013
 Foster Engineering Co., 1056
 Fulton Syphon Co., 1038-1039
 General Controls, 1042
 Hart & Cooley Manufacturing Co., 1002-1003
 Hays Corporation, 932
 Henry Furnace & Foundry Co., 843, 896
 Hoffman Specialty Co., Inc., 1016-1017
 Illinois Engineering Co., 1018-1019
 Leeds & Northrup Company, 934
 Minneapolis-Honeywell Regulator Company, 1044-1049

National Regulator Co., 1052
 Powers Regulator Co., The, 1050-1051
 Sarco Company, Inc., 1024-1025
 H. A. Thrush & Co., 926-927
 Warren Webster & Company, 1028-1030
 Westinghouse Electric & Manufacturing Co., 819-970

DAMPER REGULATORS, Furnace

Barber-Colman Company, 1034-1035
 Detroit Lubricator Company, 1036
 Foster Engineering Co., 1056
 Fulton Syphon Co., 1038-1039
 General Controls, 1042
 Hart & Cooley Manufacturing Co., 1002-1003
 Henry Furnace & Foundry Co., 843, 896
 Leeds & Northrup Co., 934
 Minneapolis-Honeywell Regulator Company, 1044-1049
 National Regulator Co., 1052
 Powers Regulator Co., The, 1050-1051
 Sarco Company, Inc., 1024-1025
 Tuttle & Bailey, Inc., 1004-1005

DAMPERS, Air Volume Control

Air Controls, Inc., 834
 Auer Register Co., The, 1001
 Hart & Cooley Manufacturing Co., 1002-1003
 Independent Register & Mfg. Co., 1006
 Tuttle & Bailey, Inc., 1004-1005
 Waterloo Register Company, The, 1007
 Young Ventilating Co., The, 849

DAMPERS, Flue

E. K. Campbell Heating Co., 909
 Henry Furnace & Foundry Co., 843, 896
 Tuttle & Bailey, Inc., 1004-1005
 Young Ventilating Company, The, 849

DAMPERS, Mechanical

Air Controls, Inc., 834
 Buffalo Forge Company, 901
 Carrier Engineering Corporation, 825
 Fulton Syphon Co., 1038-1039
 Hart & Cooley Manufacturing Co., 1002-1003
 Henry Furnace & Foundry Co., 843, 896
 Johnson Service Company, 1040-1041

National Regulator Co., 1052

Powers Regulator Co., The, 1050-1051
 Young Ventilating Company, The, 849

DEAERATORS

American Radiator Company, 801, 861-865

DIFFUSERS (See Ventilators, Floor and Wall)**DRAFT APPARATUS (See Blowers, Forced Draft)****DRYING EQUIPMENT**

Aeriet Air Conditioner Company, 799
 American Radiator Company, 801, 861-865
 Betz Unit Air Cooler Co., 804
 Buffalo Forge Company, 901

Carrier Engineering Corporation, 825
Clara Fan Company, 805
G & O Manufacturing Co., The, 918
McQuay, Incorporated, 812
B. F. Sturtevant Co., 908
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
L. J. Wing Mfg. Co., 906-907

DUST COLLECTING EQUIPMENT

American Air Filter Company, Inc., 850-851
American Blower Corporation, 802-803
American Foundry Equipment Co., The, 910
Buffalo Forge Company, 901
Canadian Sirocco Co., Ltd., 802-803
Clara Fan Company, 805
Owens-Illinois Glass Company, 854
Research Corporation, 821
B. F. Sturtevant Co., 908

DUST COLLECTORS, Cloth Type

American Air Filter Company, Inc., 850-851
American Blower Corporation, 802-803
American Foundry Equipment Co., The, 910
Canadian Sirocco Co., Ltd., 802-803
Coppus Engineering Corporation, 852

DUST SEPARATORS (See Separators, Dust)

EVAPORATORS

Aerofin Corporation, 915-917
Baker Ice Machine Co., Inc., 823
Carrier Engineering Corporation, 825
Curtis Refrigerating Machine Company, Division Curtis Manufacturing Company, 826
Feddors Manufacturing Co., 911
Frick Company (Incorporated), 827
Frigidaire Corporation, 806-807
General Electric Company, 838-839, 968-969
McQuay, Incorporated, 812
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
Vilter Manufacturing Company, The, 832
Westinghouse Electric & Manufacturing Co., 819, 970
Wittenmeier Machinery Company, 833
York Ice Machinery Corporation, 820
Young Radiator Company, 914

EXHAUST HEADS (See Heads, Exhaust)

EXPANSION JOINTS

American District Steam Company, 897
E. B. Badger & Sons Co., 898
Carrier Engineering Corporation, 825
Crane Co., 868-869
Fulton Syphon Co., 1038-1039
Illinois Engineering Company, 1018-1019
Underground Steam Construction Co., 964

EXPOSITIONS

International Exposition Co., 889

FAN BLADES

American Blower Corporation, 802-803
Buffalo Forge Company, 901
Canadian Sirocco Co., Ltd., 802-803
Torrington Mfg. Co., The, 904-905
Westinghouse Electric & Manufacturing Co., 819, 970

FANS, Centrifugal

Air Controls, Inc., 834
American Blower Corporation, 802-803
Bayley Blower Company, 900
Buckeye Blower Company, 922-923
Buffalo Forge Company, 901
E. K. Campbell Heating Co., 909
Canadian Sirocco Co., Ltd., 802-803
Clara Fan Company, 805
Coppus Engineering Corporation, 852
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
Ilg Electric Ventilating Company, 903
Meyer Furnace Company, The, 846
John J. Nesbitt, Inc., 922-923
Niagara Blower Company, 814-815
B. F. Sturtevant Co., 908
Torrington Mfg. Co., The, 904-905
Trane Company, The, 1027

FANS, Electric

Barber-Colman Company, 1034-1035
Buffalo Forge Company, 901
Century Electric Company, 967
Coppus Engineering Corporation, 852
Delco Appliance Corporation, 835, 876, 890
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
Ilg Electric Ventilating Company, 903
Westinghouse Electric & Manufacturing Co., 819, 970
L. J. Wing Mfg. Co., 906-907

FANS, Furnace

Air Controls, Inc., 834
American Blower Corporation, 802-803
Buffalo Forge Company, 901
E. K. Campbell Heating Co., 909
Canadian Sirocco Co., Ltd., 802-803
Champion Blower & Forge Co., 902
Clara Fan Company, 805
Coppus Engineering Corporation, 852
Henry Furnace & Foundry Co., 843, 896
Ilg Electric Ventilating Company, 903
Meyer Furnace Company, The, 846
L. J. Mueller Furnace Co., 844-845
Trane Company, The, 1027
L. J. Wing Mfg. Co., 906-907

FANS, Portable

American Blower Corporation, 802-803
Barber-Colman Company, 1034-1035
Bayley Blower Company, 900
Buffalo Forge Company, 901
E. K. Campbell Heating Co., 909

Canadian Sirocco Co., Ltd., 802-803
Champion Blower & Forge Co., 902
Coppus Engineering Corporation, 852
Delco Appliance Corporation, 835, 876, 890
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
Ilg Electric Ventilating Company, 903
Westinghouse Electric & Manufacturing Co., 819, 970
L. J. Wing Mfg. Co., 906-907

FANS, Propeller

Air Controls, Inc., 834
American Blower Corporation, 802-803
American Machine and Metals Manufacturing Corp., DeBothezat Division, 899
Buffalo Forge Company, 901
Canadian Sirocco Co., Ltd., 802-803
Clara Fan Company, 805
Coppus Engineering Corporation, 852
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
Ilg Electric Ventilating Company, 903
B. F. Sturtevant Co., 908
Thermal Units Manufacturing Company, 818
Torrington Mfg. Co., The, 904-905
Trane Company, The, 1027
Westinghouse Electric & Manufacturing Co., 819, 970
L. J. Wing Mfg. Co., 906-907

FANS, Supply and Exhaust

Air Controls, Inc., 834
American Blower Corporation, 802-803
American Machine and Metals Manufacturing Corp., DeBothezat Division, 899
Bayley Blower Company, 900
Buckeye Blower Company, 922-923
Buffalo Forge Company, 901
E. K. Campbell Heating Co., 909
Canadian Sirocco Co., Ltd., 802-803
Champion Blower & Forge Co., 902
Clara Fan Company, 805
Coppus Engineering Corporation, 852
Delco Appliance Corporation, 835, 876, 890
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
Ilg Electric Ventilating Company, 903
L. J. Mueller Furnace Co., 844-845
John J. Nesbitt, Inc., 922-923
Niagara Blower Company, 814-815
B. F. Sturtevant Co., 908
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
Westinghouse Electric & Manufacturing Co., 819, 970
L. J. Wing Mfg. Co., 906-907

FEED WATER HEATERS (See Heaters, Feed Water)

FEED WATER REGULATORS (See Regulators, Feed Water)

FEEDERS, Boiler

Crane Co., 868-869
General Controls, 1042
Kieley & Mueller, Inc., 1021
McDonnell & Miller, 858-859
John H. McGowan Company, The, 997
Mueller Steam Specialty Co., Inc., 1023
Warren Webster & Company (Safety), 1028-1030
Westinghouse Electric & Manufacturing Co., 819, 970
Wright-Austin Co., 1031

FEEDERS, Water

Decatur Pump Company, 996
General Controls, 1042
Kieley & Mueller, Inc., 1021
McDonnell & Miller, 858-859
Mueller Steam Specialty Co., Inc., 1023
Wright-Austin Co., 1031

FILTERS, Air (See Air Cleaning Equipment)

FITTINGS, Pipe, Flanged

American Brass Co., The, 972-973
American Radiator Company, 801, 861-865
Chase Brass & Copper Co., Incorporated, 974-975, 999
Crane Co., 868-869
Frick Company (Incorporated), 827
Grinnell Company, Inc., 919-921, 1015
Vilter Manufacturing Company, The, 832
York Ice Machinery Corporation, 820

FITTINGS, Pipe, Screwed

Chase Brass & Copper Co., Incorporated, 974-975, 999
Crane Co., 868-869
Frick Company (Incorporated), 827
Grinnell Company, Inc., 919-921, 1015
Vilter Manufacturing Company, The, 832
York Ice Machinery Corporation, 820

FITTINGS, Pipe, Sweat

American Brass Co., The, 972-973
Chase Brass & Copper Co., Incorporated, 974-975, 999
Crane Co., 868-869
Detroit Lubricator Company, 1036
Mueller Brass Co., 976-977
Revere Copper and Brass Incorporated, 979
Streamline Pipe and Fittings Co., 976-977

FITTINGS, Welding

Crane Co., 868-869
Grinnell Company, Inc., 919-921, 1015
Westinghouse Electric & Manufacturing Co., 819, 970

FLOOR AND CEILING PLATES

Beaton & Cadwell Mfg. Company, The, 1010-1011
Crane Co., 868-869

FLUE GAS ANALYSIS

Hays Corporation, 932
Leeds & Northrup Company, 934
Minneapolis-Honeywell Regulator Company, 1044-1049

FURNACE-BURNER

Delco Appliance Corporation, 835, 876, 890
Gar Wood Industries, Inc., 836-837
Gilbert & Barker Mfg. Co., 840-842
Henry Furnace & Foundry Co., 843, 896
Meyer Furnace Company, The, 846
L. J. Mueller Furnace Co., 844-845
Norge Division, Borg-Warner Corporation, 830, 847, 892

FURNACE HEATING SYSTEMS (See Heating Systems, Furnace)

FURNACE REGULATORS (See Regulators, Furnace)

FURNACES, Electric

General Electric Company, 838-839, 968-969
Westinghouse Electric & Manufacturing Co., 819, 970

FURNACES, Warm Air

American Radiator Company, 801, 861-865
E. K. Campbell Heating Co., 909
Carrier Engineering Corporation, 825
Delco Appliance Corporation, 835, 876, 890
Electrol Incorporated, 891
General Electric Company, 838-839, 968-969
Gilbert & Barker Mfg. Co., 840-842
Henry Furnace & Foundry Co., 843, 896
Lookout Boiler & Mfg. Company, 886-887
Meyer Furnace Company, The, 846
L. J. Mueller Furnace Co., 844-845
Norge Division, Borg-Warner Corporation, 830, 847, 892
Trane Company, The, 1027

GAGE BOARDS

Bristol Company, The, 929
Consolidated Ashcroft Hancock Co., Inc., 931
Julien P. Friez & Sons, Inc., 1037
Hays Corporation, 932
J. E. Loneragan Co., 1022
Minneapolis-Honeywell Regulator Company, 1044-1049

GAGE GLASSES

American Radiator Company, 801, 861-865
Hays Corporation, 932
Jenkins Bros., 1057

GAGES, Altitude

American Radiator Company, 801, 861-865
Bell and Gossett Company, 924-925
Bristol Company, The, 929
Consolidated Ashcroft Hancock Co., Inc., 931
Julien P. Friez & Sons, Inc., 1037
Hays Corporation, 932
J. E. Loneragan Co., 1022
Mercoid Corporation, The, 1043
New York Air Valve Corporation, 1058
Taylor Instrument Companies, 936-937
United States Gauge Co., 938
Warren Webster & Company, 1028-1030

GAGES, Ammonia

Baker Ice Machine Co., Inc., 823
Bristol Company, The, 929
Consolidated Ashcroft Hancock Co., Inc., 931

J. E. Loneragan Co., 1022
Mercoid Corporation, The, 1043
United States Gauge Co., 938
Vilter Manufacturing Company, The, 832

GAGES, Compound

Hoffman Specialty Co., Inc., 1016-1017
Sarco Company, Inc., 1024-1025
United States Gauge Co., 938
Warren Webster & Company, 1028-1030

GAGES, HOT Water

Bell and Gossett Company, 924-925
Bristol Company, The, 929
Consolidated Ashcroft Hancock Co., Inc., 931
Julien P. Friez & Sons, Inc., 1037
Mercoid Corporation, The, 1043
Minneapolis-Honeywell Regulator Company, 1044-1049
New York Air Valve Corporation, 1058
H. A. Thrush & Co., 926-927
United States Gauge Co., 938

GAGES, Pressure

American Radiator Company, 801, 861-865
Baker Ice Machine Co., Inc., 823
Bristol Company, The, 929
Brown Instrument Company, 930
Consolidated Ashcroft Hancock Co., Inc., 931
Hays Corporation, 932
J. E. Loneragan Co., 1022
Mercoid Corporation, The, 1043
Minneapolis-Honeywell Regulator Co., 1044-1049
New York Air Valve Corporation, 1058
Taylor Instrument Companies, 936-937
H. A. Thrush & Co., 926-927
United States Gauge Co., 938

GAGES, Steam

American Radiator Company, 801, 861-865
Anderson Manufacturing Co., 1054-1055
Bristol Company, The, 929
Consolidated Ashcroft Hancock Co., Inc., 931
C. A. Dunham Company, 1012-1013
Hoffman Specialty Co., Inc., 1016-1017
J. E. Loneragan Co., 1022
Mercoid Corporation, The, 1043
Minneapolis-Honeywell Regulator Company, 1044-1049
New York Air Valve Corporation, 1058
Taylor Instrument Companies, 936-937
United States Gauge Co., 938
Warren Webster & Company, 1028-1030

GAGES, Vacuum

American Radiator Company, 801, 861-865
Anderson Manufacturing Co., 1054-1055
Bristol Company, The, 929
Brown Instrument Company, 930
Consolidated Ashcroft Hancock Co., Inc., 931
C. A. Dunham Company, 1012-1013
Julien P. Friez & Sons, Inc., 1037
Hays Corporation, 932

Hoffman Specialty Co., Inc., 1016-1017
J. E. Lonergan Co., 1022
Mercoird Corporation, The, 1043
Minneapolis-Honeywell Regulator Company, 1044-1049
New York Air Valve Corporation, 1058
Taylor Instrument Companies, 936-937
United States Gauge Co., 938
Warren Webster & Company, 1028-1030

GAGES, Vapor

American Radiator Company, 801, 861-865
Bristol Company, The, 929
Consolidated Ashcroft Hancock Co., Inc., 931
C. A. Dunham Company, 1012-1013
Hoffman Specialty Co., Inc., 1016-1017
Mercoird Corporation, The, 1043
Minneapolis-Honeywell Regulator Company, 1044-1049
New York Air Valve Corporation, 1058
United States Gauge Co., 938
Warren Webster & Company, 1028-1030

GAGES, Water

American Radiator Company, 801, 861-865
Baker Ice Machine Co., Inc., 823
Bristol Company, The, 929
Brown Instrument Company, 930
Consolidated Ashcroft Hancock Co., Inc., 931
Detroit Lubricator Company, 1036
Julien P. Friez & Sons, Inc., 1037
J. E. Lonergan Co., 1022
Mercoird Corporation, The, 1043
Minneapolis-Honeywell Regulator Co., 1044-1049
New York Air Valve Corporation, 1058
Wright-Austin Co., 1031

GAS BURNERS

Babcock & Wilcox Company, The, 877
Delco Appliance Corporation, 835, 876, 890
Henry Furnace & Foundry Co., 843, 896
Spencer Heater Company, 872-873

GAS HEATERS' (See Heaters, Gas)

GASKETS, Asbestos
Crane Co., 868-869
Ehret Magnesia Manufacturing Co., 946
Jenkins Bros., 1057
Johns-Manville, 950-951
Ruberoide Co., The, 956-957

GASKETS, Cork

Armstrong Cork Products Company, 940
Johns-Manville, 950-951
Mundet Cork Corp., 952

GASKETS, Rubber

Crane Co., 868-869
Ehret Magnesia Manufacturing Co., 946
Frick Company (Incorporated), 827
Jenkins Bros., 1057
Johns-Manville, 950-951

GOVERNORS, Pump

Crane Co., 868-869
C. A. Dunham Company, 1012-1013
Foster Engineering Co., 1056
Illinois Engineering Company, 1018-1019
Kieley & Mueller, Inc., 1021
Mueller Steam Specialty Co., Inc., 1023
Warren Webster & Company, 1028-1030
Wright-Austin Co., 1031

GRATES FOR BOILERS AND FURNACES

American Radiator Company, 801, 861-865
Combustion Engineering Company, Inc., 879
E. Keeler Company, 883
Lookout Boiler & Mfg. Company, 886-887
L. J. Mueller Furnace Co., 844-845
Unit Heater and Cooler Co., The, 913

GRILLES, REGISTERS AND ORNAMENTAL METAL WORK

American Blower Corporation, 802-803
Auer Register Co., The, 1001
Barber-Colman Company, 1034-1035
E. K. Campbell Heating Co., 909
Canadian Sirocco Co., Ltd., 802-803
Hart & Cooley Manufacturing Co., 1002-1003
Independent Register & Mfg. Co., 1006
L. J. Mueller Furnace Co., 844-845
Trane Company, The, 1027
Tuttle & Bailey, Inc., 1004-1005
Waterloo Register Company, The, 1007

HANGERS, Pipe

American Brass Company, The, 972-973
Beaton and Cadwell Mfg. Company, The, 1010-1011
Chase Brass & Copper Co., Incorporated, 974-975, 999
Crane Co., 868-869
Frick Company (Incorporated), 827
Grinnell Company, Inc., 919-921, 1015
Ric-wil Company, The, 963
Vilter Manufacturing Company, The, 832

HANGERS, Radiator

American Radiator Company, 801, 861-865
Burnham Boiler Corporation, 866-867
Crane Co., 868-869
Grinnell Company, Inc., 919-921, 1015
National Radiator Corporation, 870-871

HEADS, Exhaust

Burt Mfg. Co., The, 1059
Crane Co., 868-869
Illinois Engineering Company, 1018-1019
Wright-Austin Co., 1031

HEADS, Sprinkler

Grinnell Company, Inc., 919-921, 1015

Mueller Brass Co., 976-977
Streamline Pipe and Fittings Co., 976-977

HEAT CABINETS (See Heaters, Cabinet)

HEAT SURFACE, Fan System

Aerofin Corporation, 915-917
Buckeye Blower Company, 922-923
Buffalo Forge Company, 901
E. K. Campbell Heating Co., 909
G. O. Manufacturing Company, The, 918
General Electric Company, 838-839, 968-969
John J. Nesbitt, Inc., 922-923
Niagara Blower Company, 814-815
B. F. Sturtevant Co., 908
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
Westinghouse Electric & Manufacturing Co., 819, 970
York Ice Machinery Corporation, 820

HEATERS, Air

Aeriet Air Conditioner Company, 799
Aerofin Corporation, 915-917
American Blower Corporation, 802-803
Buckeye Blower Company, 922-923
Buffalo Forge Company, 901
E. K. Campbell Heating Co., 909
Canadian Sirocco Co., Ltd., 802-803
Carrier Engineering Corporation, 825
Clarage Fan Company, 805
Combustion Engineering Company, Inc., 879
Electric Air Heater Company, 910
Electrol Incorporated, 891
Gar Wood Industries, Inc., 836-837
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
Ilg Electric Ventilating Company, 903
McQuay, Incorporated, 812
Meyer Furnace Company, The, 846
John J. Nesbitt, Inc., 922-923
Niagara Blower Co., 814-815
Norge Division, Borg-Warner Corporation, 830, 847, 892
B. F. Sturtevant Co., 908
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
Unit Heater and Cooler Co., The, 913
Westinghouse Electric & Manufacturing Co., 819, 970
York Ice Machinery Corporation, 820

HEATERS, Automatic Hot Water, Domestic

American Radiator Company, 801, 861-865
Davis Engineering Corporation, 928
Electrol Incorporated, 891
Gar Wood Industries, Inc., 836-837
Henry Furnace & Foundry Co., 843, 896
Sterling Engineering Company, 1026
Williams Oil-O-Matic Heating Corporation, 848, 893
Westinghouse Electric & Manufacturing Co., 819, 970

HEATERS, Blast

Aeriet Air Conditioner Company, 799
Aerofin Corporation, 915-917
American Blower Corporation, 802-803
Bayley Blower Company, 900
Buckeye Blower Company, 922-923
Buffalo Forge Company, 901
E. K. Campbell Heating Co., 909
Canadian Sirocco Co., Ltd., 802-803
Carrier Engineering Corporation, 825
Clarage Fan Company, 805
Fedders Manufacturing Co., 911
McQuay, Incorporated, 812
Modine Manufacturing Co., 912
John J. Nesbitt, Inc., 922-923
Niagara Blower Company, 814-815
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
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Aeriet Air Conditioner Company, 799
American Radiator Company, 801, 861-865
Burnham Boiler Corporation, 866-867
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
McQuay, Incorporated, 812
Modine Manufacturing Co., 912
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
Young Radiator Company, 914

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Aeriet Air Conditioner Company, 799
American Radiator Company, 801, 861-865
Electric Air Heater Company, 910
General Electric Company, 838-839, 968-969
Ilg Electric Ventilating Company, 903
B. F. Sturtevant Co., 908
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
Westinghouse Electric & Manufacturing Co., 819, 970
Young Radiator Company, 914

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Bell and Gossett Company, 924-925
Carbondale Machine Corporation, 824
Davis Engineering Corporation, 928
General Electric Company, 838-839, 968-969
Westinghouse Electric & Manufacturing Co., 819, 970

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Bell and Gossett Company, 924-925
Davis Engineering Corporation, 928
General Electric Company, 838-839, 968-969
Williams Oil-O-Matic Heating Corporation, 848, 893

HEATERS, Gas

American Radiator Company, 801, 861-865
Carrier Engineering Corporation, 825

Kewanee Boiler Corporation, 884-885
Meyer Furnace Company, The, 846

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American Radiator Company, 801, 861-865
Bell and Gossett Company, 924-925
Burnham Boiler Corporation, 866-867
Crane Co., 868-869
Electrol Incorporated, 891
Henry Furnace & Foundry Co., 843, 896
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Lookout Boiler & Mfg. Company, 886-887
L. J. Mueller Furnace Co., 844-845
Spencer Heater Company, 872-873
Sterling Engineering Company, 1026
Thermal Units Manufacturing Company, 818
Waterfilm Boilers Incorporated, 888
Williams Oil-O-Matic Heating Corporation, 848, 893
Westinghouse Electric & Manufacturing Co., 819, 970

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American District Steam Company, 897
American Radiator Company, 801, 861-865
Bell and Gossett Company, 924-925
Betz Unit Air Cooler Co., 804
E. K. Campbell Heating Co., 909
Chase Brass & Copper Co., Incorporated, 974-975, 999
Davis Engineering Corporation, 928
Electrol Incorporated, 891
McQuay, Incorporated, 812
National Radiator Corporation, 870-871
Sterling Engineering Company, 1026
Thermal Units Manufacturing Company, 818

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American Radiator Company, 801, 861-865
Kewanee Boiler Corporation, 884-885
Lookout Boiler & Mfg. Company, 886-887
L. J. Mueller Furnace Co., 844-845

HEATERS, Storage

American District Steam Company, 897
Bell and Gossett Company, 924-925
Burnham Boiler Corporation, 866-867
Davis Engineering Corporation, 928
General Electric Company, 838-839, 968-969

HEATERS, Tank

Bell and Gossett Company, 924-925
Burnham Boiler Corporation, 866-867
Grinnell Company, Inc., 919-921, 1015
Kewanee Boiler Corporation, 884-885
Lookout Boiler & Mfg. Company, 886-887
L. J. Mueller Furnace Co., 844-845

National Radiator Corporation, 870-871
Spencer Heater Company, 872-873
Sterling Engineering Company, 1026
Westinghouse Electric & Manufacturing Co., 819, 970

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Aeriet Air Conditioner Company, 799
American Blower Corporation, 802-803
Bayley Blower Company, 900
Buckeye Blower Company, 922-923
Buffalo Forge Company, 901
Burnham Boiler Corporation, 866-867
E. K. Campbell Heating Co., 909
Canadian Sirocco Co., Ltd., 802-803
Carrier Engineering Corporation, 825
Champion Blower & Forge Co., 902
Clarage Fan Company, 805
C. A. Dunham Company, 1012-1013
Electric Air Heater Company, 910
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Ilg Electric Ventilating Company, 903
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John J. Nesbitt, Inc., 922-923
Niagara Blower Company, 814-815
B. F. Sturtevant Co., 908
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
Unit Heater and Cooler Co., The, 913
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American Radiator Company, 801, 861-865
Betz Unit Air Cooler Co., 804
Buffalo Forge Company, 901
E. K. Campbell Heating Co., 909
L. J. Mueller Furnace Co., 844-845
Trane Company, The, 1027

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Aeriet Air Conditioner Company, 799
Airtemp Incorporated, 800
American Blower Corporation, 802-803
American Radiator Company, 801, 861-865
Betz Unit Air Cooler Co., 804
Buckeye Blower Company, 922-923
Buffalo Forge Company, 901
Canadian Sirocco Co., Ltd., 802-803
Carrier Engineering Corporation, 825
Clarage Fan Company, 805
Delco Appliance Corporation, 835, 876, 890
Electric Air Heater Company, 910
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Gar Wood Industries, Inc., 836-837
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
Ilg Electric Ventilating Company, 903
Kelvinator Corporation, 808-811
Lookout Boiler & Mfg. Company, 886-887

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Unit Heater and Cooler Co., The, 913
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L. J. Wing Mfg. Co., 906-907
Young Radiator Company, 914

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Airtemp Incorporated, 800
Burnham Boiler Corporation, 866-867
Delco Appliance Corporation, 835, 876, 890
Electrol Incorporated, 891
Gar Wood Industries, Inc., 836-837
General Electric Company, 838-839, 968-969
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Kelvinator Corporation, 808-811
Meyer Furnace Company, The, 846
L. J. Mueller Furnace Co., 844-845
Norge Division, Borg-Warner Corporation, 830, 847, 892
Williams Oil-O-Matic Heating Corporation, 848, 893

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American Radiator Company, 801, 861-865
E. K. Campbell Heating Co., 909
Carrier Engineering Corporation, 825
Delco Appliance Corporation, 835, 876, 890
Electrol Incorporated, 891
Gar Wood Industries, Inc., 836-837
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Kelvinator Corporation, 808-811
Lookout Boiler & Mfg. Company, 886-887
Meyer Furnace Company, The, 846
L. J. Mueller Furnace Co., 844-845
Norge Division, Borg-Warner Corporation, 830, 847, 892

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Airtemp Incorporated, 800
American Blower Corporation, 802-803
American Radiator Company, 801, 861-865
Burnham Boiler Corporation, 866-867
Canadian Sirocco Co., Ltd., 802-803
Carrier Engineering Corporation, 825
Delco Appliance Corporation, 835, 876, 890
General Electric Company, 838-839, 968-969
Henry Furnace & Foundry Co., 843, 896
Lookout Boiler & Mfg. Company, 886-887
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L. J. Mueller Furnace Co., 844-845
National Radiator Corporation, 870-871
Trane Company, The, 1027

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Aeriet Air Conditioner Company, 799
American Blower Corporation, 802-803
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Beaton and Cadwell Mfg. Company, The, 1010-1011
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Electrol Incorporated, 891
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General Electric Company, 838-839, 968-969
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Henry Furnace & Foundry Co., 843, 896
Lookout Boiler & Mfg. Company, 886-887
L. J. Mueller Furnace Co., 844-845
National Radiator Corporation, 870-871
Parks-Cramer Company, 813
Sterling Engineering Corporation, 1026
H. A. Thrush & Co., 926-927
Trane Company, The, 1027
Utica Radiator Corporation, 1000
L. J. Wing Mfg. Co., 906-907

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Airtemp Incorporated, 800
American Blower Corporation, 802-803
Bell and Gossett Company, 924-925
Canadian Sirocco Co., Ltd., 802-803
Crane Co., 868-869
Delco Appliance Corporation, 835, 876, 890
Electrol Incorporated, 891
Gar Wood Industries, Inc., 836-837
General Electric Company, 838-839, 968-969
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National Radiator Corporation, 870-871
Norge Division, Borg-Warner Corporation, 830, 847, 892
Trane Company, The, 1027
Utica Radiator Corporation, 1000

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Aeriet Air Conditioner Company, 799
American District Steam Company, 897
American Radiator Company, 801, 861-865
Anderson Manufacturing Co., 1054-1055
Barnes & Jones, Incorporated, 1014
Bell and Gossett Company, 924-925
Burnham Boiler Corporation, 866-867
E. K. Campbell Heating Co., 909
Carrier Engineering Corporation, 825
Crane Co., 868-869
Delco Appliance Corporation, 835, 876, 890
C. A. Dunham Company, 1012-1013
Electrol Incorporated, 891
Gar Wood Industries, Inc., 836-837
General Electric Company, 838-839, 968-969

Gilbert & Barker Mfg. Co., 840-842
Hoffman Specialty Co., Inc., 1016-1017
Illinois Engineering Company, 1018-1019
Lookout Boiler & Mfg. Company, 886-887
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National Radiator Corporation, 870-871
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Sarco Company, Inc., 1024-1025
Sterling Engineering Company, 1026
Trane Company, The, 1027
Utica Radiator Corporation, 1000
Warren Webster & Company, 1028-1030
L. J. Wing Mfg. Co., 906-907

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Aeriet Air Conditioner Company, 799
American Radiator Company, 801, 861-865
Barnes & Jones, Incorporated, 1014
Bell and Gossett Company, 924-925
Burnham Boiler Corporation, 866-867
Crane Co., 868-869
Delco Appliance Corporation, 835, 876, 890
C. A. Dunham Company, 1012-1013
Electrol Incorporated, 891
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Illinois Engineering Company, 1018-1019
Lookout Boiler & Mfg. Company, 886-887
L. J. Mueller Furnace Co., 844-845
National Radiator Corporation, 870-871
New York Air Valve Corporation, 1058
Sarco Company, Inc., 1024-1025
Sterling Engineering Company, 1026
Trane Company, The, 1027
Utica Radiator Corporation, 1000
Warren Webster & Company, 1028-1030

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Aeriet Air Conditioner Company, 799
American District Steam Company, 897
American Radiator Company, 801, 861-865
Barnes & Jones, Incorporated, 1014
Crane Co., 868-869
Delco Appliance Corporation, 835, 876, 890
C. A. Dunham Company, 1012-1013
Electrol Incorporated, 891
Gar Wood Industries, Inc., 836-837
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William S. Haines & Co., 1020
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New York Air Valve Corporation, 1058
Sarco Company, Inc., 1024-1025
Sterling Engineering Company, 1026
Trane Company, The, 1027
Utica Radiator Corporation, 1000
Warren Webster & Company, 1028-1030

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Airtemp Incorporated, 800
American Blower Corporation, 802-803
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American Radiator Company, 801, 861-865
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Buffalo Forge Company, 901
Burnham Boiler Corporation, 866-867
E. K. Campbell Heating Co., 909
Canadian Sirocco Co., Ltd., 802-803
Carrier Engineering Corporation, 825
Delco Appliance Corporation, 835, 876, 890
General Electric Company, 838-839, 968-969
Grinnell Company, Inc., 919-921, 1015
Modine Manufacturing Co., 912
L. J. Mueller Furnace Co., 844-845
Niagara Blower Company, 814-815
Norge Division, Borg-Warner Corporation, 830, 847, 892
Parks-Cramer Company, 813
Savage Arms Corporation, 816
B. F. Sturtevant Co., 908
Thermal Units Manufacturing Company, 818
Trane Company, The, 1027
Unit Heater & Cooler Co., The, 913
Utica Radiator Corporation, 1000
Westinghouse Electric & Manufacturing Co., 819, 970

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Airtemp Incorporated, 800
American Blower Corporation, 802-803
Bayley Blower Company, 900
Betz Unit Air Cooler Co., 804
Buffalo Forge Company, 901
Canadian Sirocco Co., Ltd., 802-803
Carrier Engineering Corporation, 825
Clarage Fan Company, 805
Delco Appliance Corporation, 835, 876, 890
Electrol Incorporated, 891
Gar Wood Industries, Inc., 836-837
General Electric Company, 838-839, 968-969
Johnson Service Company, 1040-1041
Meyer Furnace Company, The, 846
Niagara Blower Company, 814-815
Norge Division, Borg-Warner Corporation, 830, 847, 892
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Powers Regulator Co., The, 1050-1051
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Aeriet Air Conditioner Company, 799
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American Blower Corporation, 802-803
American Moistening Company, 822
Betz Unit Air Cooler Co., 804
Buffalo Forge Company, 901
Burnham Boiler Corporation, 866-867
Canadian Sirocco Co., Ltd., 802-803
Carrier Engineering Corporation, 825
Clarage Fan Company, 805
General Electric Company, 838-839, 968-969
Grinnell Company, Inc., 919-921, 1015
Modine Manufacturing Co., 912
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Parks-Cramer Company, 813
Savage Arms Corporation, 816
B. F. Sturtevant Co., 908
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Mercoid Corporation, The, 1043
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1936

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(Asterisk indicates authorship of papers)

(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

ABRAMS, Abraham (M 1927; J 1924), Pres., Abbey Htg. Co., Inc., 81 Centre Ave., and (for mail), 100 Clove Rd., New Rochelle, N. Y.
ACHESON, Albert R. (M 1919), Consulting Engr. (for mail), 501 Eckel Theatre Bldg., and 852 Ostrom Ave., Syracuse, N. Y.
ADAMS, Benjamin (M 1919), Dist. Mgr. (for mail), American Blower Corp., 781 Broad St., Station Bldg. 1617 Pennsylvania Blvd., and 3006 W. Coulter St., Queen Lane Manor, Philadelphia, Pa.
ADAMS, Charles W. (M 1920), Salesman, U. S. Radiator Corp., 1405 West 11th St., Kansas City, Mo.
ADAMS, Eugene I. (M 1934), Plant Engr., Michigan State College, and (for mail), 115 S. Pine, Lansing, Mich.
ADAMS, Harold E. (M 1930), Chief Engr. (for mail), Nash Engineering Co., South Norwalk, and Merrill Heights, Norwalk, Conn.
ADAMS, Neil D. (M 1929; A 1925; J 1922), Supt., Franklin Htg. Station (for mail), 220 Second Ave. S.W., and 836 Eighth Ave. S.W., Rochester, Minn.
ADDAMS, Homer (Charter Member; Life Member), (Presidential Member), (Pres., 1924; 1st Vice-Pres., 1923; Treas., 1915-1922; Council, 1915-1925), Pres. Kewanee Boiler Co., Inc., and Fitzgibbons Boiler Co., Inc., 570 Seventh Ave., New York, N. Y.
ADLAM, T. Napier (M 1932), Vice-Pres. and Gen. Mgr., Sarco Mfg. Co., 183 Madison Ave., New York, N. Y., and (for mail), 64 Wellington Ave., West Orange, N. J.
ADLER, Alphonse A.* (M 1921), Consulting Engr., 35 Stewart Ave., Arlington, N. J.
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AHLFF, Albert A. (M 1923; A 1918), Chase Brass & Copper Co., and (for mail), 805 Cook St., Waterbury, Conn.
AIKEN, Jack F. (S 1935), 312 Walnut S.E., Minneapolis, Minn.
AITKEN, James (A 1935), Supt. (for mail), Anthes Foundry, Ltd., and 923 Palmersron Ave., Winnipeg, Man., Canada.
AKERS, George W. (M 1929), Secy-Treas. (for mail), George W. Akers Co., 2847 Grand River Ave., Detroit, and 424 Willits, Birmingham, Mich.
ALEXANDER, Samuel W. (M 1935), Mgr. Htg. Div., James Morrison Brass Co., 276 King St. W., and (for mail), 124 Kingsmount Park Road, Toronto, Ont., Canada.
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ALT, Harold L.* (M 1913), Bldg. Equip. Engr., Gibbs & Hill, Penn Station, New York, N. Y., and (for mail), 18-C Kearny St., Newark, N. J.

AMES, Charles F. (A 1928), Vice-Pres. and Sales Mgr., Ames Pump Co., Inc., 30 Church St., and (for mail), Hotel Walton, 104 West 70th St., New York, N. Y.

AMMERMAN, Charles R. (M 1916), Consulting Engr. (for mail), 772-4 Century Bldg., and 3908 Guilford Ave., Indianapolis, Ind.

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ANDERSON, C. S. (M 1920), Mgr. (for mail), American Blower Corp., 429 Shell Bldg., 1008 W. Sixth St., and 4267 Holly Knoll Drive., Los Angeles, Calif.

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ANDERSON, Sigurd H. (S 1935), 3921 Bloomington Ave., Minneapolis, Minn.

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ANGUS, Harry H.* (M 1918), (Council, 1927-1929), Consulting Engr., 1221 Bay St., and (for mail), 34 Farnham Ave., Toronto, Ont., Canada.

ANTHES, Lawrence L. (A 1935), Pres., Imperial Iron Corp., Ltd., 30 Jefferson Ave., and (for mail), Anthes Foundry, Ltd., 64 Jefferson Ave., and 119 Dowling Ave., Toronto, Ont., Canada.

APT, Sanford R. (M 1935), Mech. Engr., U. S. Government, 1031 Navy Bldg., and (for mail), 4831-36th St. N.W., Washington, D. C.

ARCHER, David M. (M 1934), Sales Repr. (for mail), Sarco Co., Inc., 143 Federal St., Boston, and 10 Thurlow St., W. Roxbury, Mass.

ARENBERG, Milton K. (A 1920), Dist. Mgr. (for mail), Ilg Electric Vtg. Co., 182 N. LaSalle St., Chicago, and 1033 S. Linden Ave., Highland Park, Ill.

ARGUE, Edgar J. (A 1935), Salesman, Lennox Furnace Co. of Canada, Ltd. (for mail), Anthes Foundry, Ltd., and Ste. 23 Estelle Apts., Winnipeg, Man., Canada.

ARMSPACH, Otto W.* (M 1919), Chief Engr., Kroeschell Engrg. Co., 2306 N. Knox Ave., Chicago, and (for mail), 205 S. Summit Ave., Villa Park, Ill.

ARMSTRONG, Robert W. (S 1935), 2809 E. Lake of the Isles Blvd., Minneapolis, Minn.

ARNDT, Heinrich W. (A 1935), Salesman, Barrett Supply Co., 639 Broad St., and (for mail), 214 Third St., Augusta, Ga.

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BAILEY, Edward P. Jr. (M 1925), Consultant Mayfield Road at Lee Blvd., and (for mail), 2475 Lee Blvd., Cleveland, Ohio.

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BAKER, Howard C. (M 1921), The H. C. Baker Co., 128 S. St. Clair St., Toledo, Ohio.

BAKER, Irving C. (M 1921), General Sales Mgr. (for mail), Airtemp, Inc., 8021 Conant Road, Detroit, and 1428 Yorkshire Road, Grosse Pointe, Mich.

BAKER, Roland H. (M 1928; A 1924), Pres. (for mail), R. H. Baker Co., Inc., 145 Broadway, and 420 Memorial Drive, Cambridge, Mass.

BAKER, William H., Jr. (A 1935), Boiler and Radiator Sales (for mail), Crane Co., 836 S. Michigan Ave., and 1211 N. State St., Chicago, Ill.

BALDWIN, William Howard (M 1921), Branch Mgr. (for mail), C. A. Dunham Co., 2988 E. Grand Blvd., and 2246 Calvert, Detroit, Mich.

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BARBERA, Henry A. (S 1932), 1727 Colden Ave., New York, N. Y.

BARBIERI, Patrick J. (S 1933), Engrg. Asst., Dwight D. Kimball, Consulting Engr., Room 1728 Grand Central Terminal Bldg., and (for mail), 2186 Belmont Ave., New York, N. Y.

BARNES, Walter E. (M 1933), Pres., Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain, Boston, and (for mail), 7 Woodlawn Ave., Wellesley Hills, Mass.

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BARNUM, Charles R. (S 1935), 1494 Capitol Ave., St. Paul, Minn.

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BARR, George W. (M 1905), (Board of Governors, 1910), Dist. Mgr., Aerofin Corp., Land Title Bldg., Philadelphia, and (for mail), Woods End, Villanova, Pa.

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BARRY, Patrick I. (M 1920), (Peace Commissioner), MIHVE (for mail), M. Barry, Ltd., 4 Marlboro St., and Budaka, Sidney Park, Cork, Ireland.

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BARTLETT, C. Edwin (M 1922), Pres. (for mail), Bartlett & Co., Inc., 1938 Market St., and 3111 W. Coulter St., Philadelphia, Pa.

BARTON, Delbert H. (S 1935), Box 44 B, Somerville, and (for mail), Box 1088, S.E., College Station, Texas.

BASTEDO, Albert E. (M 1919), Vice-Pres.-Treas.-Mgr. (for mail), Burnham Boiler Corp., Irvington-on-Hudson, and Burnside Drive, Hastings-on-Hudson, N. Y.

BAUER, Albert E. (M 1935), Engr., U. S. Air Cond. Corp., Minneapolis, and (for mail), 59 S. Victoria St., St. Paul, Minn.

BAUGHMAN, L. R. (M 1935), Htg. Engr., Modern Plumbing & Heating Service, and (for mail), 2706 Eschol Ave., Zion, Ill.

BAUM, Albert L. (M 1916), Member of Firm (for mail), Jaros Baum & Bolles, 415 Lexington Ave., and 601 West 113th St., New York, N. Y.

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BAUR, John W. (S 1935), 2517 Leland Ave., Chicago, and (for mail), 1004 S. Second St., Champaign, Ill.

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BEEBE, Frederick E. W. (A 1915), Johnson Service Co., 28 East 29th St., New York, N. Y.

BEGGS, William E. (M 1927), Pres., W. E. Beggs Co., 907 Lloyd Bldg., and (for mail), 3636 Palatine Ave., Seattle, Wash.

BEIGHEL, Howard Atlee (A 1927), Sales Repr. (for mail), The Herman Nelson Corp., 503 Columbia Bank Bldg., Pittsburgh, and 207 Puritan Road, Rosslyn Farms, Carnegie, Pa.

BEITZEL, Albert E. (A 1933; J 1930), Mgr., Air Cond. Div., Combustioneer Stoker Corp., Tenth and D Sts. S.W., and (for mail), 1475 Columbia Road, Washington, D. C.

BELING, Earl H. (A 1930; J 1925), 2428-13th St., Moline, Ill.

BELL, E. Floyd (M 1933), (for mail), 619 Foshay Tower, and 2605 Fremont Ave. S., Minneapolis, Minn.

BEMAN, Myron C. (M 1926), (Council, 1934-1935), Consulting Engr. (for mail), Beman & Candee, 374 Delaware Ave., and 699 Richmond Ave., Buffalo, N. Y.

BENNETT, Edwin A. (J 1929), Sales Engr. (for mail), American Blower Corp., 401 Broadway, New York, N. Y., and 45 Pondfield Road W., Bronxville, N. Y.

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BENSE, William M. (S 1934), Engr., Institute of Thermal Research (for mail), American Radiator Co., 675 Bronx River Road, Yonkers, and 340 Hayward Ave., Mt. Vernon, N. Y.

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- BLACK, F. C.** (M 1919), Pres. (for mail), F. C. Black Co., 622 W. Randolph St., and 4535 N. Ashland Ave., Chicago, Ill.
- BLACK, Harry G.** (M 1917), Prop. (for mail), P. Gormly Co., 155 N. Tenth St., and 927 North 65th St., Philadelphia, Pa.
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- BLACKMORE, F. H.** (M 1923), Mgr., Operating Dept. (for mail), U. S. Radiator Corp., Box 686 Detroit, and 515 Tooting Lane, Birmingham, Mich.
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- BLAIR, Howard A.** (J 1936), Service Engr., Air Cond. Equip., Westinghouse Electric & Mfg. Co., Mansfield, Ohio.
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- BLANKIN, Merrill F.** (M 1927; A 1926; J 1919), Pres. (for mail), Haynes Selling Co., Inc., 1518 Fairmount Ave., and 3328 W. Penn St., Philadelphia, Pa.
- BLESSED, William A.** (A 1935), Mech. Engr. (for mail), Mueller Brass Co., and 515-15th St., Fort Huron, Mich.
- BLISS, George L.** (A 1933), Engr. and Sales (for mail), Allis-Chalmers Mfg. Co., 1410 Waldheim Bldg., 11th and Main, and 7641 Brooklyn Ave., Kansas City, Mo.
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- BODINGER, Jacob H.** (M 1931), Pres. (for mail), Bodinger & Co., Inc., 530 Tenth Ave., New York, and 1429 East 19th St., Brooklyn, N. Y.
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- BOLTON, Reginald Pelham** (Life Member; M 1897), (Presidential Member), (Pres., 1911; 1st Vice-Pres., 1905-1910; 2nd Vice-Pres., 1903; Board of Governors, 1901, 1905, 1910, 1911, 1912, 1913), The R. P. Bolton Co., 116 East 19th St., New York, N. Y.
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- BREDESEN, Bernhard P.** (A 1931), Htg. and Vtg. Engr. (for mail), Herman Nelson Corp., 410 Essex Bldg., and 3119 Knox Ave., N., Minneapolis, Minn.
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GIBBS, Frank C. (M 1921), Gen. Supt. (for mail), National Regulator Co., 2301 Knox Ave., Chicago, and 150 N. Cuyler Ave., Oak Park, Ill.

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GILES, Alfred F. (J 1934), H. H. Robertson Co., 2000 Grant Bldg., and (for mail), 4307 Ludwick St., Pittsburgh, Pa.

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ROSELL, Axel F. (M 1935), Mech. Engr., A. B. Svenska Flaktfabriken Kungsgatan 8, Stockholm, and (for mail), Kv Atlas 3, Lidingsö, Sweden.
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ROSENBURG, William E. (J 1935), Pibg-Htg., John C. Rosenberg, Birch Hill Road., Locust Valley, L. I., N. Y.
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ROTH, Charles F. (A 1930), Mgr., International Htg. & Vtg. Exposition, Grand Central Palace, New York, and (for mail), 141 East 36th St., New York (Nov. 1 to April 30), and Dreamthorp, Bedford Village, N. Y. (May 1 to Oct. 31).
ROTH, Harold R. (M 1935), Mgr. (for mail), Canadian Sirocco Co., Ltd., Room 321, 57 Bloor St. W., and 18 Tichester Road, Toronto, Ont., Canada.
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ROWLEY, Frank B.* (M 1918), (Presidential Member) (Pres., 1932; 1st Vice-Pres., 1931; 2nd Vice-Pres., 1930; Council, 1927-1933), Prof. of Mech. Engrg. and Director of Experimental Engrg. Lab., University of Minnesota, and (for mail), 4801 E. Lake Harriet Blvd., Minneapolis, Minn.
ROYER, Earl B. (M 1928), Designing Engr., Fosdick & Hilmer, 1703 Union Trust Bldg., and (for mail), 6635 Iris Ave., Cincinnati, Ohio.
RUDIO, H. M. (M 1921), Mgr., Air Cond. Dept. (for mail), Gustin Bacon Mfg. Co., 1412 West 12th St., and 6639 Edgerale Road, Kansas City, Mo.

RUEGER, Charles A., Jr. (A 1935), Salesman (for mail), American Radiator Co., P. O. Box 505, Winston-Salem, N. C., and 2805 Floyd Ave., Richmond, Va.

RUFF, Adolph G. (M 1935), Plant Engr., U. S. Playing Card Co., Norwood, and (for mail), Box 355 R. F. D. 5, Lockland, Ohio.

RUFF, DeWitt C. (M 1922), Healy-Ruff Co., 765 Hampden Ave., St. Paul, Minn.

RUGART, Karl (A 1924), Dist. Mgr. (for mail), Warren Webster & Co., 26 South 20th St., and 5830 Willows Ave., Philadelphia, Pa.

RUNKEL, Charles (M 1935), Pres. (for mail), Acme Htg. & Vtg. Co., Inc., 4224 S. Lowe Ave., and 7921 S. Hermitage Ave., Chicago, Ill.

RUPPERT, Edward H. (A 1923), 85 Eastern Pkwy., Brooklyn, N. Y.

RUSSELL, J. Nelson (M 1899), Managing Dir. (for mail), Rosser & Russell, Ltd., Romney House Marsham St., Westminster, and Fernacres Fulmer near Slough, Buckinghamshire, England.

RUSSELL, W. A. (M 1921), (Council, 1934-1935), Mgr., K. C. Branch (for mail), U. S. Radiator Corp., 1405 West 11th St., and 239 Ward Pkwy., Kansas City, Mo.

RUSSELL, William B. (M 1928), Colorado Ave., R. F. D. 1, Joliet, Ill.

RYAN, Harry J. (M 1922), 47 Harris Ave., Albany, N. Y.

RYAN, James D. (M 1935), Bldg. Mgr. and Engr., Whitney National Bank, St. Charles and Gravier St., and (for mail), 215 N. Rendon St., New Orleans, La.

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RYDELL, Carl A. (M 1931; A 1931; J 1928), Owner, C. A. Rydell Associates (for mail), 168 Dartmouth St., Boston, and 286 Quinobequin Road, Waban, Mass.

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SADLER, C. Boone (M 1928), Assoc. Civil Engr., (for mail), Public Works Office, 11th Naval District, and 4826 Voltaire St., San Diego, Calif.

SAITO, Shozo (M 1923), Marunouchi Bldg., Opposite Tokyo Sta., Tokyo, Japan.

SAKOUTA, Mathieu L. (M 1924), Consulting Engr. and Expert, Gavan, Simanskaia 4-A, Leningrad, U. S. S. R.

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SAWHILL, R. V. (A 1929), Executive Vice-Pres. (for mail), Domestic Engineering, 110 East 42nd St., New York, and 115 Townsend Ave., Pelham Manor, N. Y.

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SCHICK, Karl W. (A 1934), Dist. Mgr., Minneapolis-Honeywell Regulator Co., 561 Reading Road, and (for mail), 2300 Auburn Ave., Cincinnati, Ohio.

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SCHMIDT, Richard H. (S 1934), 2139 Abington Road, Cleveland, Ohio.

SCHMUTZ, Jean (M 1933), Administrateur-Delegue Societe PRSM 8, Passage de l'Atlas, and (for mail), 18 Rue Dufrenoy, Paris 16, France.

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SHANKLIN, John A. (M 1928), Secy-Treas. (for mail), West Virginia Htg. & Pbg. Co., 233 Hale St., and 1507 Quarrier St., Charleston, W. Va.

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SHARP, Henry C. (M 1935), Mgr., Oil Htg. Div. (for mail), Smith Oil & Refining Co., and 1928 Rockton Ave., Rockford, Ill.

SHAVER, Herbert H. (A 1929), Asst. Gen. Sales Agt. (for mail), Hudson Coal Co., 424 Wyoming Ave., and 1507 Wyoming Ave., Scranton, Pa.

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SHAW, Edgar (M 1923), Pres. (for mail), Lynch & Woodward, Inc., 320 Dover St., Boston, and 51 Royal St., Wollaston, Mass.

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SHEPPARD, William G. F. (M 1922), Partner (for mail), Sheppard & Abbott, 119 Harbord St., and 1 Clarendon Ave., Toronto, Ont., Canada.

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SHROCK, John H. (M 1924), Mgr. (for mail), New York Blower Co., Factory St., and 1524 Michigan Ave., La Porte, Ind.

SHULTZ, Earle (A 1919), Vice-Pres. (for mail), Illinois Maintenance Co., 1136-72 W. Adams St., and Edgewater Beach Apts., Chicago, Ill.

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SMITH, Gard W. (M 1927), Salesman, Premier Warm Air Heater Co., Dowagiac, Mich., and (for mail), 248 Roche St., Huntington, Ind.

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SPIELMANN, Gordon P. (A 1931; J 1923), Vice-Pres. (for mail), Harrison-Speilmann Co., 480 Milwaukee Ave., Chicago, and 730 N. Prospect Ave., Park Ridge, Ill.

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SPROULL, Howard E. (M 1920), Div. Sales Mgr. (for mail), American Blower Corp., 1005-6 American Bldg., and 3588 Raymar Drive, Cincinnati, Ohio.

SPURGEON, Joseph H. (M 1924), Salesman (for mail), Spurgeon Co., 5-203 General Motors Bldg., and 17215 Pennington Drive., Detroit, Mich.

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WALTHER, Vernon H. (M 1928; J 1925), Mech. Engr., 6321 Osceola Ave., Edison Park, Chicago, Ill.

WALTERTHUM, John J. (A 1922), Htg. and Vtg. Contractor, 212 East 58th St., New York, N. Y., and (for mail), 42-a Van Reipen Ave., Jersey City, N. J.

WALTON, Charles W., Jr. (M 1934), Mech. Engr. (for mail), Rockefeller Center, Inc., 30 Rockefeller Plaza, New York, N. Y., and 120 Monte Viste Ave., Ridgewood, N. J.

WANDLESS, Franklin W. (M 1925), Registered Engr. (for mail), 1518 Fairmount Ave., Philadelphia, and Berwyn, Pa.

WARD, Frank J. (M 1935), Owner, Frank J. Ward Co., Cold Spring, Ky.

WARD, Oscar G. (M 1919), Dist. Repr. (for mail), Johnson Service Co., 1230 California St., and 1607 Jasmine St., Denver, Colo.

WARDELL, Arthur (M 1935), Partner (for mail), Thomas & Wardell, 229 College St., and 124 Melrose Ave., Toronto, Ont., Canada.

WARING, J. M. S. (M 1932), Consulting Engr. (for mail), Chase & Waring, 17 East 42nd St., and 277 Park Ave., New York, N. Y.

WARREN, Charles W. (S 1935), Mech. Engr. (for mail), A. J. Warren, 2313 Ave. E., and 3317-R4½, Galveston, Texas.

WARREN, Clarence N. (M 1919), Vice-Pres., Hayes Bros., Inc., 236 W. Vermont St., and (for mail), 419 East 48th St., Indianapolis, Ind.

WARREN, Francis C. (M 1934), Salesman (for mail), American Blower Corp., 228 N. LaSalle St., Chicago, and 127 East Ave., Park Ridge, Ill.

WARREN, Harry L. (M 1930), Htg. Engr., Southern California Gas Co., 950 S. Broadway, Los Angeles, and (for mail), 1303 Huntington Drive, South Pasadena, Calif.

WASHBURN, Marcus J. (A 1934), Insulation Engr. (for mail), Eagle-Picher Lead Co., Temple Bar Bldg., and 2211 Park Ave., Cincinnati, Ohio.

WASHINGTON, George (M 1934), Sales Engr. (for mail), Hoffman Specialty Co., 130 N. Wells St., Chicago, and 4327 Johnson Ave., Western Springs, Ill.

WASHINGTON, Laurence W. (M 1929), Vice-Pres. (for mail), National Regulator Co., 2301 Knox Ave., Chicago, and 778 Laurel Ave., Des Plaines, Ill.

WATERMAN, John H. (M 1931), Engr. (for mail), Chas. T. Main, Inc., 201 Devonshire St., Boston, and 7 Centre St., Cambridge, Mass.

WATERS, George G. (M 1931; A 1926), Dist. Mgr. (for mail), American Blower Corp., 1433 Oliver Bldg., and 52 Vernon Drive, Pittsburgh (16), Pa.

WATSON, H. Dalton (A 1935), Br. Mgr. (for mail), Linde Canadian Refrigeration Co., 124 King St., and 61 Furby St., Winnipeg, Man., Canada.

WATSON, M. Barry (M 1928), Consulting Engr., 121 Welland Ave., Toronto, Ont., Canada.

WAUNG, Tsing-fu (M 1935; J 1933), Htg. Engr. (for mail), Andersen, Meyer & Co., Ltd., Yuen Ming Yuen Road, and 16 Lane, 152 Edinburgh Road, Shanghai, China.

WEATHERBY, Edward P., Jr. (S 1935), 813 N. Zangs Blvd., Dallas, Texas.

WEBB, Ernest C. (M 1935), Engrg. Service Mgr., Iron Fireman Mfg. Co., 3170 West 106th St., Cleveland, and (for mail), 1202 Woodside Ave., Rocky River, Ohio.

WEBB, John S. (M 1920), Sales Mgr., Webster Tallmadge Co., Inc., 300 Madison Ave., New York, N. Y., and (for mail), 345 Brookline St., Needham, Mass.

WEBB, John W. (M 1926), Managing Dir. (for mail), Webb Dust Removing & Drying Co., Ltd., Princess and Park St. Works, and "Ebor," Brinnington, Stockport, England.

WEBSTER, E. Kessler (M 1915), Warren Webster & Co., 17th and Federal Sts., Camden, N. J.

WEBSTER, Warren (Life Member 1933; M 1906; A 1899), Pres., Warren Webster & Co., 17th and Federal Sts., Camden, N. J.

WEBSTER, Warren, Jr. (M 1932; A 1932; J 1927), Vice-Pres. (for mail), Warren Webster & Co., 17th and Federal Sts., Camden, and Washington Ave. and Colonial Ridge, Haddonfield, N. J.

WEBSTER, William H., Jr. (A 1935), Vice-Pres. (for mail), Hurst Heating Engrs., Inc., 400 York St., and 625 Maryland Ave., Norfolk, Va.

WECHSBERG, Otto (M 1932), Pres. and Gen. Mgr., Coppus Engrg. Corp., 344 Park Ave., and (for mail), 1006 Main St., Worcester, Mass.

WEGMANN, Albert (M 1918), 6206 North 17th St., Philadelphia, Pa.

WEIL, Martin (A 1925), Vice-Pres. (for mail), Weil-McLain Co., 641 W. Lake St., and 4259 Hazel Ave., Chicago, Ill.

WEIL, Maurice I. (A 1928), Pres. (for mail), Chicago Pump Co., 2336 Wolfram St., and 1409 Elmdale Ave., Chicago, Ill.

WEIMER, Fred G. (A 1919), Salesman, Kewanee Boiler Corp., 1741 W. St. Paul Ave., and (for mail), 3958 N. Stowell Ave., Milwaukee, Wis.

WEINSHANK, Theodore* (Life Member 1933; M 1906), (Board of Governors, 1913), Consulting Engr., 3307 Belden Ave., Chicago, Ill.

WEISS, Arthur P. (M 1928), Burnham Boiler Corp., Irvington, and (for mail), 134 Farrington Ave., North Tarrytown, N. Y.

WEISS, Carl A. (A 1924), Supt. (for mail), Kornbrodt Kornice Ko., 1811 Troost Ave., and 29 East 68th St., Kansas City, Mo.

WEITZEL, Paul H. (S 1934), Cameron B. Weitzel, 122 E. High St., Manheim, Pa.

WELCH, Louis A., Jr. (A 1929), 443 Second St., Schenectady, N. Y.

WELDY, Lloyd O. (M 1930), Sales Engr. (for mail), The Powers Regulator Co., 2720 Greenview Ave., and 2846 North 77th Ave., Chicago, Ill.

WELSH, Harry S. (M 1906), Sales Engr., Weil-McLain Co., and (for mail), 53 Kemphurst Road, Rochester, N. Y.

WELTER, M. A. (A 1925), (for mail), Twin City Furnace Co., 410-12 W. Lake St., and 4306 Garfield St., Minneapolis, Minn.

WENDT, Edgar F. (M 1918), Pres. (for mail), Buffalo Forge Co., 490 Broadway, and 120 Lincoln Pkwy., Buffalo, N. Y.

WEST, Perry* (M 1911), (Council, 1920-1925; Treas., 1924-1925), Consulting Engr. (for mail), 13 Central Ave., and 445 Ridge St., Newark, N. J.

WETZEL, Horace E. (M 1934), Chief Engr. (for mail), The Smith & Oby Co., 6107 Carnegie Ave., and 8796 Elsmere Drive, Cleveland, Ohio.

WHALLON, Fletcher (S 1935), 3852 Lyndale Ave. S., Minneapolis, Minn.

WHEELER, Otto J. (M 1923), Pres.-Treas. (for mail), The Samuel A. Esswein Htg. & Pbg. Co., 548-558 W. Broad St., and 2044 Collingswood Road, Columbus, Ohio.

WHEELER, Harry S. (M 1916), Vice-Pres., L. J. Wing Mfg. Co., 154 West 14th St., New York, and (for mail), 725 Union Ave., Elizabeth, N. J.

WHITE, Eugene B. (M 1934), Arch. and Engr. (for mail), Y. M. C. A., 19 S. LaSalle St., Chicago, and 309 N. Taylor Ave., Oak Park, Ill.

WHITE, Elwood S. (M 1921), Pres. (for mail), Taco Heaters, Inc., Room 1224, 342 Madison Ave., New York, N. Y., and Meadowbank Road, Old Greenwich, Conn.

WHITE, Everett A. (M 1921), Engrg. Dept., Crane Co., 30 South 16th St., and (for mail), 5244 Nottingham St., St. Louis, Mo.

WHITE, John C. (M 1932), State Power Plant Engr. (for mail), 624 E. Main St., and 622 E. Main St., Madison, Wis.

WHITELAW, H. Leigh (M 1916), Vice-Pres. (for mail), American Gas Products Corp., 40 West 40th St., New York, N. Y., and Overbrook Lane, Darien, Conn.

WHITELEY, Stockett M. (M 1933), Consulting Engr. (for mail), Baltimore Life Bldg., and 3931 Canterbury Road, Baltimore, Md.

WHITMER, Robert P. (M 1935), Secy. (for mail), American Foundry & Furnace Co., 915 E. Washington St., and 1402 E. Washington St., Bloomington, Ill.

WHITNEY, C. W. (M 1935), Mgr., Trilling & Montague, 24th and Walnut Sts., Philadelphia, and (for mail), Apt. F-1, Sevilla Court, Bala-Cynwyd, Pa.

WHITTAKER, Wayne K. (A 1935), Htg.-Vtg. Maint., Irving Trust Co. Bldg., 1 Wall St., New York, and (for mail), 198-40-32nd Ave., Flushing, L. I., N. Y.

WHITTALL, Ernest T. (A 1933), Vice-Pres., May Oil Burner of Canada, Ltd., 17 Elm St., and (for mail), 11 Cottingham Road, Toronto, Ont., Canada.

WIEGNER, Henry B. (M 1919), Mgr., Boston Office, Johnson Service Co., 20 Winchester St., Boston, and (for mail), 143 Standish Road, Watertown, Mass.

WIERENGA, Peter O. (A 1931), Vice-Pres. (for mail), C. C. James Co., 49 Coldbrook St. N.E., and 231 Brown St. S.E., Grand Rapids, Mich.

WIGGINS, Oswald J. (J 1935; S 1933), Engr., Moran & Co., 602 Globe Bldg., St. Paul, and (for mail), 1629 Sixth St. S.E., Apt. 8, Minneapolis, Minn.

WIGGS, G. Lorne (A 1932; J 1924), Consulting Engr. (for mail), University Tower, and 4797 Grosvenor Ave., Montreal, P. Q., Canada.

WIGLE, Bruce M. (A 1926), Pres. (for mail), Bruce Wigle Pbg. & Htg. Co., 9117 Hamilton Ave., and 18114 Oak Drive, Detroit, Mich.

WILDER, Edward L. (M 1915), Mgr., Gas Sales (for mail), Utility Management Corp., 150 Broadway, New York, and 12 Mereland Road, New Rochelle, N. Y.

WILEY, Edgar C. (M 1909), Wiley & Wilson, Lynchburg, Va.

WILHELM, Joseph E. (S 1934), 1355 West 87th, Cleveland, Ohio.

WILKINSON, Farley J. (M 1933), Mgr., Engrg. Service, Montgomery Ward & Co., and (for mail), 18257 Martin Ave., Homewood, Ill.

WILLARD, Arthur C.* (M 1914), (*Presidential Member*), Pres., 1928; 1st Vice-Pres., 1927; 2nd Vice-Pres., 1926; Council, 1925-1929, (for mail), President, University of Illinois, and 711 Florida Ave., Urbana, Ill.

WILLEY, Earl C. (M 1934), Mech. Engrg. Instr., Oregon State College, and (for mail), 1652 "A" St., Corvallis, Ore.

WILLIAMS, Allen W. (A 1915), Managing Director (for mail), National Warm Air Htg. & Air Cond. Association, 50 W. Broad St., Columbus, and 51 Meadow Park Ave., Bexley, Ohio.

WILLIAMS, Clinton R. (M 1935), Draftsman, E. I. DuPont de Nemours Co., Wilmington, Del., and (for mail), 155 Primos Ave., Folcroft, Pa.

WILLIAMS, Frank H. (J 1934), Air Cond. Tester, Frigidaire Div., General Motors, and (for mail), 118 Maplewood Ave., Dayton, Ohio.

WILLIAMS, J. McFarland, Jr. (A 1928; J 1927), Sales Engr., 1407-35th St. N.W., Washington, D. C.

WILLIAMS, J. Walter (M 1915), Pres. (for mail), Forest City Plbg. Co., 332-36 East State St., and 923 E. State St., Ithaca, N. Y.

WILLIAMS, Leo E. (A 1933; J 1930), Viscose Co., and (for mail), 827 Liberty St., Meadville, Pa.

WILLIS, Leonard L. (S 1935), Engr., Conrad Refrigeration Co., 17 E. Hennepin, and (for mail), 1212 Oliver N., Minneapolis, Minn.

WILMOT, Charles S. (M 1919), (for mail), Building Insulation Co., 106 South 16th St., Philadelphia, and 406 Essex Ave., Narberth, Pa.

WILSON, George T. (M 1925), Sales Engr., Gurney Foundry Co., Ltd., 4 Junction Road, Toronto, and (for mail), Tyre Ave., Islington, Ont., Canada.

WILSON, Harold A., Jr. (J 1933), Gen. Sales Dept., American Radiator Co., 40 West 40th St., and (for mail), 1133 Park Ave., New York, N. Y.

WILSON, James W. (S 1935), 3603 Cole Ave., Dallas, Texas.

WILSON, Raymond W. (M 1934), Member of Firm (for mail), Wilson-Brinker Co., 412 Pythian Bldg., and 429 Creston Ave., Kalamazoo, Mich.

WILSON, W. H. (A 1932), Steamfitter Foreman, Pullman-Standard Car Mfg. Co., 11001 Cottage Grove Ave., and (for mail), 22 West 110th Place, Chicago, Ill.

WILSON, William H. (A 1923), Branch Mgr. (for mail), Johnson Service Co., 507 E. Michigan St., and 2023 E. Olive St., Milwaukee, Wis.

WILTBERGER, Constant F. (M 1935), Mech. Engr., Stewart A. Jellett Co., 1200 Locust St., and (for mail), 2650 N. Ninth St., Philadelphia, Pa.

WINANS, Glen D. (M 1929), Engr. of Steam Distribution (for mail), The Detroit Edison Co., 2000 Second Ave., and 16183 Wisconsin, Detroit, Mich.

WINQUIST, Walter J. (A 1930), Htg. and Vtg. Engr., 294 Nostrand Ave., Brooklyn, N. Y.

WINSLOW, C.-E. A.* (M 1932), Prof. of Public Health (for mail), Yale University, 310 Cedar St., and 314 Prospect St., New Haven, Conn.

WINTERBOTTOM, Ralph F. (M 1923), Htg. Engr., Winterbottom Supply Co., Commercial and Miles, and (for mail), 1002 Riehl St., Waterloo, Iowa.

WINTERER, Frank C. (M 1920), Sales Mgr. (for mail), Cochran Sargent Co., Broadway and Kellogg Blvd., and 836 Juno St., St. Paul, Minn.

WINTHER, Anker (J 1932), Air Cond. Engr., York Ice Machinery Corp., 2116 Gilbert Ave., Cincinnati, Ohio.

WISE, Daniel E. (S 1934), 400 Temple St., New Haven, Conn.

WITMER, Charles N. (J 1930), Dist. Dealer Supervisor (for mail), Carrier Engrg. Corp., 2022 Bryan St., and 4154½ Prescott St., Dallas, Texas.

WOESE, Carl F. (M 1934), Consulting Engr. (for mail), Robson & Woese, Inc., 1001 Burnet Ave., and 256 Robineau Road, Syracuse, N. Y.

WOHL, Maurice W. (M 1934), 100 Linden Blvd., Brooklyn, N. Y.

WOLF, John C. (M 1923), Draftsman and Engr. (for mail), B. F. Sturtevant Co., and 76 Beacon St., Hyde Park, Mass.

WOLF, Philip (M 1935), Prop., City Contracting Co., 2025 Fifth Ave., New York, N. Y.

WOLFERT, Edward R. (M 1935), Section Engr. (for mail), Westinghouse Electric & Mfg. Co., East Springfield, and 119 Windmere St., Springfield, Mass.

WOLFF, Peter P. (M 1935), Engr., Bell & Gossett Co., 3000 Wallace St., and (for mail), 7058 Rhodes Ave., Chicago, Ill.

WOOD, Frederick C. (J 1931), Sales Engr., Air Cond. (for mail), Airtemp, Inc., Div. Chrysler Motors, 8021 Conant Road, and 16220 Normandy, Detroit, Mich.

WOOD, J. Sydney (M 1926), Estimator (for mail), Bennett & Wright, Ltd., 72 Queen St. E., and 163 Briar Hill Ave., Toronto, Ont., Canada.

WOODMAN, Lawrence E. (M 1934), Pres. (for mail), Woodman Appliance & Engrg. Corp., 203 E. Capitol, and 1014 Fairmount, Jefferson City, Mo.

WOODS, Edward H. (M 1934), Prop. (for mail), F. H. Higgins, 311 E. State St. and Hook Place, Ithaca, N. Y.

WOOLLARD, Mason S. (M 1934), Draftsman, H. H. Angus, Consulting Engr., 1221 Bay St., and (for mail), 31 Hillcrest Park Ave., Toronto, Ont., Canada.

WOOLSTON, A. H. (M 1919), Chief Engr. (for mail), Bowers Bros. & Co., 2015 Sansom St., and 4815 North 12th St., Philadelphia, Pa.

WORSHAM, Herman (M 1925; J 1918), (for mail), Frigidaire Corp., 224 West 57th St., New York, N. Y., and 103 N. Walnut St., East Orange, N. J.

WRIGHT, Clarence E. (J 1935; S 1933), Htg. and Vtg. Engr., Fairmont Wall Plaster Co., and (for mail), Locust Ave. Ext., Fairmont, W. Va.

WRIGHT, Kenneth A. (M 1921), Mgr. (for mail), Johnson Service Co., 1113 Race St., Cincinnati, Ohio, and 113 Orchard St., Ft. Mitchell, Ky.

WRIGHT, M. Birney (A 1932; J 1929), Mech. Engr., E. I. duPont de Nemour & Co. (for mail), Route 1 Box 78-C, Charleston, W. Va.

WRIGHT, William J. (S 1935), 1111 W. Illinois St., Urbana, Ill.

WUNDERLICH, Milton S.* (M 1925), Mech. Engr., The Insulite Co., 1100 Builders Exchange, Minneapolis, and (for mail), 545 Mount Curre Blvd., St. Paul, Minn.

WYLIE, Howard M. (M 1925; J 1917), Vice-Pres. in charge of Sales (for mail), Nash Engrg. Co., and 51 Elmwood Ave., South Norwalk, Conn.

Y

YAGER, John J. (M 1921), 425 Woodbridge Ave., Buffalo, N. Y.

YAGLOU, Constantin P.* (M 1923), Asst. Prof. of Industrial Hygiene (for mail), Harvard School of Public Health, 55 Shattuck St., Boston, and 10 Vernon Road, Belmont, Mass.

YATES, George L. (S 1934), Instructor, Dept. of Oil and Gas Prod. (for mail), University of Pittsburgh, Pittsburgh, Pa., and 1220 Johnstone, Bartlesville, Okla.

YATES, James E. (M 1934), Mgr. (for mail), Yates Neale & Co., 231 Tenth St., and 431-16th St., Brandon, Man., Canada.

YATES, Walter (*Life Member* 1934; M 1902), Governing Dir. (for mail), Matthews & Yates, Ltd., Cyclone Works, and Parksend Swinton, Man., England.

YOUNG, Emil O. (A 1935), Pres.-Treas. (for mail), The Young Ventilating Co., 2703-7 Woodland Ave., and 2040 East 83rd St., Cleveland, Ohio.

Z

ZACK, Hans J. (M 1928), Mgr., The Zack Co., 2311 Van Buren St., Chicago, Ill.

ZANGRILLI, Albert J. (S 1935), 537 Turrett St., Pittsburgh, Pa.

ZIBOLD, Carl E. (M 1929), Mech. Engr., Htg. and Vtg., 13 Chadwick Road, Westminster Ridge, White Plains, N. Y.

ZIESSE, Karl L. (A 1931), Secy.-Treas. (for mail), Phoenix Sprinkler & Htg. Co., 115 Campau Ave. N.W., and 315 Hampton Ave. S.E., Grand Rapids, Mich.

ZIEBER, William E. (M 1935), Asst. Chief Engr. (for mail), York Ice Machinery Corp., Roosevelt Ave., and 112 S. Penn St., York, Pa.

ZIMMERMAN, Alexander H. (A 1930), Ventilation Engr., Chicago Board of Health, 707 City Hall, and (for mail), 5147 N. St. Louis Ave., Chicago, Ill.

ZINK, David D. (M 1931), Consulting Engr., Owner (for mail), 225 Plaza Theatre Bldg., Kansas City, and Hickman Mills, Mo.

ZOKELT, C. G. (M 1921), Consulting Engr., 3810-24th Ave. S., Seattle, Wash.

ZUHLKE, William R. (M 1928), 530 McLean Ave., Yonkers, N. Y.

Summary of Membership

(Corrected to January 1, 1936)

UNITED STATES AND ISLAND TERRITORIES

Alabama.....	2	Nebraska.....	2
Arizona.....	2	New Hampshire.....	1
Arkansas.....	3	New Jersey.....	105
California.....	40	New York.....	362
Colorado.....	5	North Carolina.....	9
Connecticut.....	33	North Dakota.....	1
Delaware.....	6	Ohio.....	105
District of Columbia.....	26	Oklahoma.....	28
Florida.....	7	Oregon.....	1
Georgia.....	11	Pennsylvania.....	225
Hawaii.....	1	Phillipine Islands.....	2
Illinois.....	230	Rhode Island.....	5
Indiana.....	18	South Carolina.....	2
Iowa.....	8	South Dakota.....	1
Kansas.....	5	Tennessee.....	6
Kentucky.....	8	Texas.....	38
Louisiana.....	8	Utah.....	1
Maine.....	4	Vermont.....	3
Maryland.....	20	Virginia.....	15
Massachusetts.....	107	Washington.....	23
Michigan.....	111	West Virginia.....	7
Minnesota.....	113	Wisconsin.....	60
Missouri.....	86		
Montana.....	4		1860

Canada..... 115

FOREIGN COUNTRIES

Australia.....	4	Japan.....	5
Belgium.....	1	Mexico.....	3
China.....	9	New Zealand.....	3
Czechoslovakia.....	1	Norway.....	2
Denmark.....	1	Scotland.....	1
England.....	16	South Africa.....	2
France.....	8	Spain.....	2
Germany.....	2	Sweden.....	3
India.....	1	U. S. S. R.....	1
Ireland.....	1		69
Italy.....	3		

Total Membership..... 2044

SUMMARY OF MEMBERSHIP BY GRADES

Honorary Members.....	1
Presidential Members.....	24
Members.....	1248
Associate Members.....	407
Junior Members.....	216
Student Members.....	148
	2044

LIST OF MEMBERS Geographically Arranged

UNITED STATES and ISLAND TERRITORIES

ALABAMA	Pasadena—	New London—	Danielson, W. A.
	Gifford, R. L.	Chapin, C. G.	Downes, H. H.
Birmingham—		Forsberg, W.	Febrey, E. J.
Fried, H. V.	San Diego	Hopson, W. T.	Feltwell, R. H.
Lichty, C. P.	Sadler, C. B.		Fife, G. D.
ARIZONA	San Francisco—	Norwalk—	Frankel, G. S.
	Bouey, A. J.	Mead, E. A.	Gardner, S. F.
Tucson—	Cochran, L. H.	Riverside—	Grimes, F. M.
Moreau, D.	Corrao, J.	Murphy, J. R.	Hewett, J. B.
Phoenix—	Haley, H. S.		Hood, O. P.
Keys, L. F.	Hudson, R. A.	South Norwalk—	Kiczeles, M. D.
ARKANSAS	Krueger, J. I.	Adams, H. E.	Kingswell, W. E.
	Leland, W. E.	Harvey, A. D.	Mayette, C. E.
Fort Smith—	South Gate—	Jennings, I. C.	Mullen, T. J., Jr.
Herrick, L.	Barnum, W. E., Jr.	Lyons, C. J.	Oursuff, L. S.
Pine Bluff—	South Pasadena—	Wylie, H. M.	Smallman, E. W.
Greer, W. R.	Warren, H. L.		Thompson, N. S.
Siloam Springs—		Stamford—	Urdahl, T. H.
Jones, C. R.		Hoyt, L. W.	Vaughan, J. G., Jr.
CALIFORNIA		Torrington—	Vinson, N. L.
		Doster, A.	Williams, J. M., Jr.
Beverly Hills—	Colorado Springs—	Kaczinski, C.	
Nelson, H. A.	Davis, A. F.	Wallingford—	FLORIDA
Glendale—	Jardine, D. C.	Burns, J. R.	Ft. Lauderdale—
Moon, F. L.	Denver—		Charlton, J. F.
Los Angeles—	McQuaid, D. J.	Waterbury—	Jacksonville—
Anderson, C. S.	O'Rear, L. R.	Ahlff, A. A.	Varner, J. L.
Binford, W. M.	Ward, O. G.	Davis, O. E.	Miami—
Bullock, H. H.		Lincoln, R. L.	Burke, W. J.
Cawby, E. L.	CONNECTICUT	Simpson, W. K.	Lingo, C. K.
Cooper, A. W.		Stewart, C. W.	Munro, E. A.
Cranston, W. E., Jr.	Bridgeport—		Pizie, S. G.
Edwards, H. B.	Monroe, R. R.	Westport—	West Palm Beach—
Ellingwood, E. L.	Smak, J. R.	Goldschmidt, O. E.	Hodeaux, W. L.
English, H.	Fairfield—		GEORGIA
Hendrickson, H. M.	Osborn, W. J.	DELAWARE	Atlanta—
Hill, F. M.	Greenwich—		Baker, C. T.
Hogue, W. M.	Jones, A. L.	Wilmington—	Clare, F. W.
Hungerford, L.	Opperman, E. F.	Gawthrop, F. H.	Gouedy, K. E.
Kendall, E. H.		Hayman, A. E., Jr.	Kagey, I. B., Jr.
Kennedy, M.	Hartford—	Kershaw, M. G.	Kent, L. F.
Koolstra, J. F.	Millard, J. W.	Lownsbury, B. F.	Klein, E. W.
Lauer, H. B.		Ponsell, F. I.	Kuempel, L. L.
Ness, W. H. C.	Manchester—	Schoenijahn, R. P.	McKinney, W. J.
Orar, A. G.	Buck, L.		Templin, C. L.
Ott, O. W.	New Haven—	DISTRICT OF COLUMBIA	Thornton, W. B.
Park, J. F.	Blakeley, H. J.	Washington—	Augusta—
Pierce, E. D.	Hughes, C. E.	Apt. S. R.	Arndt, H. W.
Polderman, L. H.	Seeley, L. E.	Beitzell, A. E.	
Scofield, P. C.	Teasdale, L. A.	Brown, W. A.	HAWAII
Simonds, A. H.	Winslow, C. E. A.	Coward, H.	Honolulu—
Oakland—	Wise, D. E.	DallaValle, J. M.	Petersen, S. E.
Cummings, G. J.			
Mears, L. A.			

ILLINOIS

Bloomington—

MaGill, W. J.
Soper, H. A.
Whitmer, R. P.

Champaign—

Baur, J. W.
Ficnerski, P.
Hintz, H. P.
Logan, T. M.
Ransom, C. F.
Strakosh, W. C.
Thetford, J. E.

Chicago—

Aeberly, J. J.
Anderson, L. B.
Arenberg, M. K.
Baker, W. H., Jr.
Baumgardner, C. M.
Becker, W. A.
Bishop, M. W.
Black, F. C.
Boehmer, A. P.
Boite, E. E.
Borling, J. R.
Bracken, J. H.
Braun, L. T.
Brigham, C. M.
Broom, B. A.
Brown, A. P.
Brown, T.
Carman, G. G.
Casey, B. L.
Christman, W. F.
Christophersen, A. E.
Clegg, R. R.
Close, P. D.
Crone, C. E., Jr.
Cunningham, T. M.
Cutler, J. A.
DeLand, C. W.
Doherty, R.
Dunham, C. A.
Emmert, L. D.
Ericsson, E. B.
Evleth, E. B.
Fatz, J. L.
Finan, J. J.
Fleming, J. P.
Fleming, T. F.
Foster, T. R.
Frank, J. M.
Freas, R. B.
Gardner, W., Jr.
Gaylord, F. H.
Getschow, R. M.
Gibbs, F. C.
Goetz, A. H.
Gossett, E. J.
Gotschall, H. C.
Graham, E. W.
Graves, W. B.
Haas, S. L.
Haines, J. J.
Hale, J. F.
Hanley, T. F., Jr.
Hart, H. M.
Hattis, R. E.
Hayden, C. F.
Hayes, J. J.
Hayward, R. B.
Hobley, H. F. J.
Herlihy, J. J.
Hill, E. V.
Hinckley, H. B.
Hornung, J. C.
Horton, H. F.
Howatt, J.
Howell, L.
Hubbard, G. W.
Hustool, A. M.
Jennings, W. G.

Jenson, J. S.
Johns, H. B.
Johnson, C. W.
Kaufman, C. W.
Keeney, F. P.
Kehm, H. S.
Kreissl, H. G.
Krez, L.
Kyle, W. J.
Lagodzinski, H. J.
Lautenschlager, F.
Lawler, M. M.
Lees, H. K.
Lewis, S. R.
Lockhart, H. A.
Machen, J. T.
Malone, D. G.
Malvin, R. C.
Marshall, P. J.
Martin, A. B.
Matchett, J. C.
Mathis, E.
Mathis, H.
Mathis, J. W.
May, M. F.
McCauley, J. H.
McClellan, J. E.
McDonnell, E. N.
McFarland, W. P.
McIlvaine, J. H.
Mehl, O. H.
Mertz, W. A.
Miller, F. A.
Miller, R. T.
Mittendorf, E. M.
Muller, H. C.
Murphy, E. T.
Narowetz, L. L., Jr.
Needler, J. H.
Neiler, S. G.
Nelson, C. L.
Newport, C. F.
O'Brien, J. H.
Olson, C. F.
Paul, L. O.
Peller, L.
Pitcher, L. J.
Piwinski, E. J.
Pope, S. A.
Powers, F. W.
Prentice, O. J.
Presdee, C. W.
Price, C. E.
Rasmussen, R. P.
Raymond, F. I.
Rager, H. P.
Reid, H. P.
Ries, L. S.
Rietz, E. W.
Rottmayer, S. I.
Runkel, C.
Schneider, D. B.
Schwein, H. J.
Seelig, L.
Shultz, E.
Snider, L. A.
Spielman, G. P.
Stannard, J. M.
Sutcliffe, A. G.
Thinn, C. A.
Thomas, L. G. L.
Thomas, R. H.
Thommen, A. A.
Tobin, J. F.
Torquist, E. L.
Truit, J. E.
Trumbo, S. M.
Turner, G. G.
Vernon, J. R.
Wallace, K. S.
Walters, W. T.
Walther, V. H.
Warren, F. C.
Washington, G.

Washington, L. W.
Well, M.
Well, M. I.
Weinshank, T.
Weldy, L. O.
White, E. B.
Wilson, W. H.
Wolff, P. P.
Zack, H. J.
Zimmerman, A. H.

Cicero—
Brown, N. A.
Keppner, H. W.

Decatur—
Shorb, W. A.

Elmhurst—
Potvin, L. J.

Evansville—
Hayes, J. J.
Maccubbin, H. A.
Miller, J. E.
Moore, R. E.

Glen Ellyn—
Sherman, V. L.

Homewood—
Wilkinson, F. J.

Joliet—
Russell, W. B.

Kewanee—
Bronson, C. E.
Dickson, R. B.
Hartman, J. M.
Pursell, J. E.

LaGrange—
Eaton, B. K.
Linn, H. R.

Lake Forest—
Moore, C. H.
Scudder, B.

Moline—
Beling, E. H.
Nelson, H. W.
Nelson, R. H.
Otis, G. E.

Mt. Vernon—
Benoist, L. L.

Norwood Park—
Olson, B.

Oak Park—
Blanding, G. H.
Fitzgerald, M. J.
May, E. M.
Nightingale, G. F.
Uhlhorn, W. J.

Park Ridge—
Heckel, E. P.
Spielmann, H. J.

Peoria—
Baird, S. A.
Farnsworth, J. G.
Meyer, F. L.

Rockford—
Braatz, C. J.
Dewey, R. P.
Merwin, G. E.
Sharp, H. C.
Stewart, D. J.

Washington, L. W.

Well, M.

Well, M. I.

Weinshank, T.

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Wilson, W. H.

Wolff, P. P.

Zack, H. J.

Zimmerman, A. H.

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Evansville—

Glen Ellyn—

Homewood—

Joliet—

Kewanee—

LaGrange—

Lake Forest—

Moline—

Mt. Vernon—

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Oak Park—

Park Ridge—

Peoria—

Rockford—

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White, E. B.

Wilson, W. H.

Wolff, P. P.

Zack, H. J.

Zimmerman, A. H.

Urbana—

Bowditch, R. P.
Broderick, E. L.
Compton, W. E.
Cunningham, J. S.
Fahnestock, M. K.
Faulkner, G.
Gallagher, P.
Konzo, S.
Kratz, A. P.
Severns, W. H.
Willard, A. C.
Wright, W. J.

Villa Park—
Armspach, O. W.

Willmette—
Norris, W. D.

Winnetka—
Ellis, E. E.

Zion—
Baughman, L. R.

INDIANA

Evansville—
Bulleit, C. R.

Huntington—
Smith, G. W.

Indianapolis—
Ammerman, C. R.
Fenstermaker, S. E.
Hagedorn, C. H.
Hayes, J. G.
Krusse, R. W.
Ott, R. C.
Poehner, R. E.
Supple, G. B.
Warren, C. N.

Lafayette—
Hoffman, J. D.

LaPorte—
Shrock, J. H.

Michigan City—
Stockwell, W. R.

Peru—
Pyle, J. W.
Thrush, H. A.

St. Mary-of-the-
Woods
Bisch, B. J.

Wabash—
Shivers, P. F.

IOWA

Ackley—
Nelson, G. O.

Adel—
Shaw, B. E.

Cedar Falls—
Clay, C. H.

Cedar Rapids—
Pappenfus, W. G.

Council Bluffs—
Huffacker, H. B.

LeMars—
Mathey, N. J.

Washington, L. W.

Well, M.

Well, M. I.

Weinshank, T.

Weldy, L. O.

White, E. B.

Wilson, W. H.

Wolff, P. P.

Zack, H. J.

Zimmerman, A. H.

Cicero—

Decatur—

Elmhurst—

Evansville—

Glen Ellyn—

Homewood—

Joliet—

Kewanee—

LaGrange—

Lake Forest—

Moline—

Mt. Vernon—

Norwood Park—

Oak Park—

Park Ridge—

Peoria—

Rockford—

Washington, L. W.

Well, M.

Well, M. I.

Weinshank, T.

Weldy, L. O.

White, E. B.

Wilson, W. H.

Wolff, P. P.

Zack, H. J.

Zimmerman, A. H.

Sioux City—

Hagan, W. V.

Waterloo—

Winterbottom, R. F.

KANSAS

Hutchinson—

Mann, A. R.
Stevens, H. L.

Lawrence—

Machin, D. W.
Sluss, A. H.

Salina—

Ryan, W. F.

KENTUCKY

Cold Spring—

Ward, F. J.

Fr. Thomas—

Reik, R. C.
Stevens, W. R.

Lexington—

May, J. W.
O'Bannon, L. S.

Louisville—

Hellstrom, J.
Murphy, H. C.
Reed, W. M.

LOUISIANA

Choudrant—

Rinehart, W. R.

New Orleans—

Gamble, C. B.
Gammill, O. E., Jr.
May, G. E.
McLaren, F. S.
Moore, H. W.
Ryan, J. D.

Shreveport—

Fitzgerald, W. E.

MAINE

Auburn—

Fowles, H. H.

Portland—

Fels, A. B.
Merrill, C. J.
Swanson, H.

MARYLAND

Baltimore—

Axeman, J. E.
Collier, W. I.
Hall, M. S.
Lednum, J. M.
Leitch, R. L.
McCormack, D.
Posey, J.
Seiter, J. E.
Shepard, J. deB.
Smoot, T. H.
Steele, M. G.
Vance, L. G.
Vincent, P. J.
Whiteley, S. M.

Brooklyn Park—

Rodgers, J. S.

Ellicott City—

Tibbets, J. C.

Rockville—

Brunett, A. L.

Roland Park—

Dorsey, F. C.

Silver Spring—

Fineran, E. V.
Stack, A. E.

MASSACHUSETTS

Arlington—

Shaw, N. J. H.

Arlington Heights—

Tarr, H. M.

Boston—

Archer, D. M.
Berchtold, E. W.
Boyden, D. S.
Brinton, J. W.
Brissette, L. A.
Bryant, A. G.
Bullock, T. A.
Cummings, C. H.
Donohoe, J. B.
Drinker, F.
Dusossoit, E. A.
Edwards, D. J.
Foulds, P. A. L.
Franklin, R. S.
Gerrish, G. B.
Gleason, G. H.
Hajek, W. J.
Hilliard, C. E.
Hoyt, C. W.
Keefe, E. T.
Kelley, J. J.
Kellogg, A.
Kimball, C. W.
McCoy, T. F.
Merrill, F. A.
Miller, J. F. G.
Moulton, D.
Osborne, M. M.
Palmer, R. T.
Rydell, C. A.
Shaw, E.
Stetson, L. R.
Swaney, C. R.
Turner, J.
Tuttle, J. F.
Waterman, J. H.
Yaglou, C. P.

Lawrence—

Bride, W. T.

Leominster—

Kern, R. T.

Lexington—

Brigham, F. H.

Lynn—

Feehan, J. B.
Oates, W. A.

Malden—

Ahlberg, H. B.

Medford—

Cushman, L. D.

Melrose—

Cole, E. O.
Dodge, H. G.

Milton—

Mitchell, C. H.

Needham—

Park, C. D.
Webb, J. S.

Newton Center—

Murray, J. J.

Newtonville—

Emerson, R. R.
McMurrer, L. J.

Quincy—

Stone, E. R.

Reading—

Ingalls, F. D. B.

Revere—

Foulds, S. T. N.

Springfield—

Cross, R. E.
Holmes, R. E.
Leland, W. B.
McDowell, B. W.
Murphy, W. W.
O'Neil, J. M.

Swampscott—

Knowles, M. G.

Waban—

Jones, W. T.

Watertown—

Wiegner, H. B.

Wellesley Hills—

Barnes, W. E.
Gilling, W. F., Jr.

West Roxbury—

Christie, A. Y.
McPherson, W. A.
Walsh, E. R., Jr.

Weymouth—

Clough, L.

Winchester—

Jackson, A. B.

Woburn—

Parker, P.

Wollaston—

Hodgdon, H. A.

Worcester—

Robinson, H. C.
Turner, P. K.
Wechsberg, O.

Watertown—

Wiegner, H. B.

Wellesley Hills—

Barnes, W. E.
Gilling, W. F., Jr.

West Roxbury—

Christie, A. Y.
McPherson, W. A.
Walsh, E. R., Jr.

Weymouth—

- McConachie, L. L.
McIntire, J. F.
McLean, D.
Miller, R. E.
Milward, R. K.
Morse, C. T.
O'Gorman, J. S., Jr.
Paetz, H. E.
Parrott, L. G.
Partlan, J. W.
Pike, W. H.
Purcell, F. C.
Purcell, R. E.
Randall, W. C.
Rickner, C. A.
Rowe, W. A.
Sanford, S. S.
Sauer, R. L.
Schechter, J. P.
Shea, M. B.
Snell, E.
Snyder, J. W.
Spitzley, R. L.
Spurgeon, J. H.
Taylor, W. E.
Toonder, C. L.
Turner, J. W.
Tuttle, G. H.
Walker, J. H.
Wallace, J. B.
Wallich, A. C.
Wigle, B. M.
Winans, G. D.
Wood, F. C.
- Port Huron—**
Blessed, W. A.
- St. Joseph—**
Milliken, V. D.
- MINNESOTA**
- Bayport—**
Swanson, E. C.
- Cloquet—**
Spaford, A.
- Duluth—**
Foster, C.
- International Falls**
Nelson, R. A.
- Minneapolis—**
Aiken, J. F.
Algren, A. B.
Anderson, S. H.
Armstrong, R. W.
Bean, G. S.
Bell, E. F.
Bensen, C. L.
Betts, H. M.
Bjorken, M. H.
Bredesen, B. P.
Buot, A. V.
Burns, E. J.
Burritt, C. G.
Butts, R. L.
Cash, T. T.
Copperud, E. R.
Dahlstrom, G. A.
Dovolis, N. J.
Edelman, B. P.
Fergestad, M. L.
Forfar, D. M.
Gelb, A.
Gerrish, H. E.
Gordon, E. B., Jr.
Gordon, W. J., Jr.
Gross, L. C.
Haavetvedt, S. R.
Hall, J. R.
Hammerski, F. D.
Hanson, L. P.
Harris, J. B.
Hildebrandt, H. A.
Hitchcock, P. C.
Howard, E.
Huch, A. J.
Johnson, L. H.
Jordan, L. E.
Jordan, R. C.
King, R. L.
Kingland, G. D.
Knudtson, C. M.
Kuehn, W. C.
Kuns, J. W.
Lange, F. F.
Legler, F. W.
Leslie, D. E.
Lund, C. E.
Magney, G. R.
Martens, J. V.
Maynard, H. R.
Miller, H. A.
Mills, H. C.
Mjolsnes, L. O.
Morgan, G. C.
Morton, H. S.
Myers, C. R.
Ostrin, A.
Pfeifer, O. J., Jr.
Powell, K. A.
Pung, D. W.
- Roberts, J. R.**
Rossiter, P. A.
Rowley, F. B.
Sanford, A. L.
Scherneck, F. H.
Seelert, E. H.
Seepe, P. E.
Spencer, J. B.
Stiller, F. W.
Sundell, S. S.
Sutherland, D. L.
Swanson, R. G.
Sweiven, C. E.
Swenson, J. E.
Tabor, C. B.
Uhl, E. J.
Uhl, W. F.
Van Horn, H. T.
Welter, M. A.
Whallon, F.
Wiggins, O. J.
Willis, L. L.
- Owatonna—**
Clarkson, W. B.
- Robbinsdale—**
Sweatt, C. H.
- Rochester—**
Adams, N. D.
- Shevlin—**
Nesdahl, E.
- St. Paul—**
Anderson, D. B.
Ayers, E. H.
Backstrom, R. E.
Barnum, C. R.
Bauer, A. E.
Buenger, A.
Collins, M. D.
Fitts, C. D.
Gausman, C. E.
Gill, J. W.
Haslett, H. M.
Hickey, D. W.
Hyde, L. L.
Jones, E. F.
McNamara, W.
Ober, H. C.
Pennel, R.
Poucher, R. C.
Ruff, D. C.
Swanstrom, A. E.
Winterer, F. C.
Wunderlich, M. S.
- Wayzata—**
Heberling, C. W.
- MISSOURI**
- Clayton—**
DuBois, L. J.
- Ferguson—**
Szombathy, L. R.
- Jefferson City—**
Woodman, L. E.
- Kansas City—**
Adams, C. W.
Allan, N. J.
Arthur, J. M.
Belz, H. D.
Bliss, G. L.
Buckley, M. B.
Caleb, D.
Campbell, E. K.
Campbell, E. K., Jr.
- Cassell, W. L.**
Chase, L. R.
Clegg, C.
Cook, B. F.
Dawson, T. L.
Dean, F. J., Jr.
Disney, M. A.
Dodds, F. F.
Downes, N. W.
Eppright, J. O.
Fehlig, J. B.
Filkins, H. L.
Flarsheim, C. A.
Gillham, W. E.
Haas, E., Jr.
Kell, W. R.
Kitchen, J. H.
Lewis, J. G.
Maillard, A. L.
Matthews, J. E.
Millis, L. W.
Natkun, B.
Nottberg, G.
Nottberg, H.
Olchoff, M.
Olson, G. E.
Pines, S.
Rivard, M. M.
Rudio, H. M.
Russell, W. A.
Sawyer, J. N.
Sheppard, F. A.
Stephenson, L. A.
Weiss, C. A.
Zink, D. D.
- Maplewood—**
Walters, A. L.
- St. Joseph—**
Harton, A. J.
- St. Louis—**
Barry, J. G., Jr.
Bayse, H. V.
Bradley, E. P.
Carlson, E. E.
Cooper, J. W.
Corrigan, J. A.
Davis, C. R.
Edwards, D. F.
Fagin, D. J.
Fillo, F. B.
Foster, J. M.
Gilmore, L. A.
Grossmann, H. A.
Hamig, L. L.
Hamilton, J. E.
Hartwein, C. E.
Hester, T. J.
Hugoniot, V. E.
Kent, J. K.
Langenberg, E. B.
Manahan, J. E.
McLarney, H. W.
McMahon, T. W.
Moon, L. W.
O'Brien, W. N.
Reed, P. L.
Rodenheiser, G. B.
Rosebrough, R. M.
Sodemann, P. W.
Sodemann, W. C. B.
Stammer, E. L.
Tenkonohy, R. J.
White, E. A.
- Sedalia—**
Middleton, H. A.
- University City—**
Falvey, J. D.
- Webster Groves—**
Myers, G. W. F.
Ranck, G. L.

- MONTANA**
- Big Timber—**
Strickland, A. W.
- Billings—**
Cohagen, C. C.
- Bozeman—**
Powers, F. I.
- Great Falls—**
Ginn, T. M.
- NEBRASKA**
- Clarks—**
Manning, W. M.
- Omaha—**
Lindberg, A. F.
- NEW HAMPSHIRE**
- Holderness—**
Meakin, J. B.
- NEW JERSEY**
- Arlington—**
Adler, A. A.
Bock, B. A.
- Asbury Park—**
Strevell, R. P.
- Atlantic City—**
Strouse, S. B.
- Bayonne—**
Schwartz, J.
- Belmar—**
Merkel, F. P.
- Bloomfield—**
Clericuzio, G. P.
Faust, F. H.
Harrington, E.
Hochuli, H. W.
Lyon, P. S.
McLennan, D. W.
Tenney, D.
- Camden—**
Brown, W. M.
Coward, C. W.
Kappel, G. W. A.
Lanning, E. K.
Rohlin, K. W.
Webster, E. K.
Webster, W.
Webster, W., Jr.
- Collingswood—**
Mohrfield, H. H.
Plum, L. H.
- Cranford—**
Roos, E. B. J.
- East Orange—**
Ferguson, R. R.
Gombers, H. B.
Grah, V. F.
Rack, E. C.
Reilly, J. H.
Tallmadge, W.
Turno, W. G. W.
- Elizabeth—**
Atkinson, K. B.
Cornwall, G. T.
Grant, W. A.
Merle, A.
Wheller, H. S.
- Essex Fells—**
Stacey, A. E., Jr.
- Grantwood—**
Butler, P. D.
- Haddonfield—**
Dobbs, C. E.
Jones, R. E.
- Hasbrouck Heights—**
Goodwin, S. L.
- Hawthorne—**
Lawton, F. C.
Spoolstra, W. J.
- Irrington—**
Reinke, A. G.
Stengel, F. J.
- Jersey City—**
Hashagen, J. B.
Jones, H. L.
Kelly, C. J.
Ritchie, W.
Walterthum, J. J.
- Lyndhurst—**
Ehrlich, M. W.
- Maplewood—**
Kepeler, D. A.
Kylberg, V. C.
Smith, M. S.
- Merchantville—**
Binder, C. G.
- Montclair—**
Benz, H.
- Newark—**
Alt, H. L.
Ashley, C. M.
Bryant, P. J.
Carey, P. C.
Carrier, W. H.
Connell, H.
Day, V. S.
French, D.
Holbrook, F. M.
Holton, J. H.
Ingels, M.
Leinroth, J. P.
Lewis, L. L.
Lewis, T.
Lyle, J. I.
Lyman, S. E.
Machouse, H. P.
Rachal, J. M.
Ray, L. B.
Raymer, W. F., Jr.
Rice, R. B.
Soule, L. C.
Steinmetz, C. W. A.
Tavanlar, E. J.
West, P.
- North Arlington—**
Bermel, A. H.
Vogelbach, O.
- Orange—**
Crawford, J. H., Jr.
- Paterson—**
Bannon, L. E.
Cox, H. F.
Pryor, F. L.
- Perth Amboy—**
Simkin, M.
- Plainfield—**
Hedges, H. B.
Tobin, G. J.
- Ridgefield Park—**
Davis, A. C.
- Ridgewood—**
Fitts, J. C.
- River Edge—**
Mawby, P.
- Rochelle Park**
Emery, G. W.
- Roselle Park—**
Kampish, N. S.
- South Orange—**
Browne, A. L.
Hansen, C. C.
- Summit—**
Oaks, O. O.
- Teaneck—**
Heebner, W. M.
- Union City—**
Taverna, F. F.
- Verona—**
Shotwell, R. W.
Stone, G. F.
- Westfield—**
Scribner, E. D.
- West Orange—**
Adlam, T. N.
- West New York—**
Stinard, R. L.
- NEW YORK**
- Albany—**
Bond, H. A.
Dick, A. V.
Johnson, H. S.
Murray, T. F.
Ryan, H. J.
Taggart, R. C.
- Binghamton—**
Brown, R. F.
Marum, O.
- Bronxville—**
Dornheim, G. A.
Livar, A. P.
- Buffalo—**
Beman, M. C.
Booth, C. A.
Cherry, L. A.
Cheyney, C. C.
Cressy, R. E.
Crique, A. H.
Currier, C. A.
Danforth, N. L.
Davis, J.
Day, H. C.
Dyer, O. K.
Erdle, G. F.
Eschenbach, S. P.
Evans, C. A.
Farnham, R.
Farrar, C. W.
Gifford, C. A.
Hamlin, C. J., Jr.
- Harding, L. A.**
Heath, W. R.
Hedley, P. S.
Hexamer, H. D.
Hirschman, W. F.
Jackson, M. S.
Johnson, E. E.
Kaiser, F.
Kamman, A. R.
Landers, J. J.
Lighthart, C. H.
Love, C. H.
Madison, R. D.
Mahoney, D. J.
McCaughy, J. E., Jr.
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Mosher, C. H.
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Strouse, S. W.
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- Larchmont—**
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Young, E. O.

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Hull, H. B.
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Dolan, R. G.
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Howlett, I. G.
Leverance, H. J.
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Miner, M. H.
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Tulsa—
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OREGON

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Blairsville—
Stitt, E. W.
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Brookline—
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Reed, V. A.
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Pottsville—
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Primos, Del. Co.—
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Prospect Park—
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Reading—
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Ridley Park—
Stalib, J. G.

Rutledge—
Jones, N. R.
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Springfield—
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Bonthron, R. C.
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Carey, J. A.

Wallingford—
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Wilkes-Barre—
Santee, H. C.

Wilkinsburg—
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Wormleysburg—
Miller, T. G.

York—
Chapin, H. G.
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Macrae, R. B.

RHODE ISLAND

Providence—

Coleman, J. B.
Gibbs, E. W.
Hartwell, J. C.
McLaughlin, J. D.
Moulder, A. W.

SOUTH CAROLINA

Clemson College—

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Columbia—

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SOUTH DAKOTA

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Pullen, R. R.

TENNESSEE

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2nd Vice-President.....F. Paul Anderson
Treasurer.....Perry West
Secretary.....F. C. Houghten

Council
Chairman, S. E. Dibble
Wm. H. Driscoll, Vice-Chm.
Homer Addams
F. Paul Anderson
W. H. Carrier
J. A. Cutler
W. E. Gillham
W. T. Jones
Thornton Lewis
J. H. Walker
Perry West
A. C. Willard
F. C. Houghten, *Secy.*

1926
President.....W. H. Driscoll
1st Vice-President.....F. Paul Anderson
2nd Vice-President.....A. C. Willard
Treasurer.....W. E. Gillham
Secretary.....A. V. Hutchinson

Council
Chairman, W. H. Driscoll
F. Paul Anderson, Vice-Chm.
W. H. Carrier
J. A. Cutler
S. E. Dibble
W. E. Gillham
C. V. Haynes
W. T. Jones
E. B. Langenberg
Thornton Lewis
J. F. McIntire
A. C. Willard

1927
President.....F. Paul Anderson
1st Vice-President.....A. C. Willard
2nd Vice-President.....Thornton Lewis
Treasurer.....W. E. Gillham
Secretary.....A. V. Hutchinson

Council
Chairman, F. Paul Anderson
A. C. Willard, Vice-Chm.
John Howatt
H. H. Angus
W. H. Carrier
W. H. Driscoll
Roswell Farnham
H. H. Fielding
W. E. Gillham
C. V. Haynes
W. T. Jones
J. J. Kissick
E. B. Langenberg
Thornton Lewis
J. F. McIntire
H. Lee Moore
F. B. Rowley

ROLL OF MEMBERSHIP

1928
President.....A. C. Willard
1st Vice-President.....Thornton Lewis
2nd Vice-President.....L. A. Harding
Treasurer.....W. E. Gillham
Secretary.....A. V. Hutchinson

Council
Chairman, A. C. Willard
Thornton Lewis, Vice-Chm.
F. Paul Anderson
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
J. F. McIntire
H. Lee Moore
F. B. Rowley

1929
President.....Thornton Lewis
1st Vice-President.....L. A. Harding
2nd Vice-President.....W. H. Carrier
Treasurer.....W. E. Gillham
Secretary.....A. V. Hutchinson
Technical Secretary.....P. D. Close

Council
Chairman, Thornton Lewis
L. A. Harding, Vice-Chm.
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
W. A. Rowe
F. B. Rowley
A. C. Willard

1930
President.....L. A. Harding
1st Vice-President.....W. H. Carrier
2nd Vice-President.....F. B. Rowley
Treasurer.....C. W. Farrar
Secretary.....A. V. Hutchinson
Technical Secretary.....P. D. Close

Council
Chairman, L. A. Harding
John Howatt, Vice-Chm.
H. H. Angus
D. S. Boyden
R. H. Carpenter
J. D. Cassell
N. W. Downes
Roswell Farnham
C. W. Farrar
W. T. Jones
E. B. Langenberg
G. L. Larson
Thornton Lewis
F. C. McIntosh
W. A. Rowe
F. B. Rowley

1931
President.....W. H. Carrier
1st Vice-President.....F. B. Rowley
2nd Vice-President.....W. T. Jones
Treasurer.....F. D. Mensing
Secretary.....A. V. Hutchinson
Technical Secretary.....P. D. Close

Council
Chairman, W. H. Carrier
F. B. Rowley, Vice-Chm.
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
Roswell Farnham
E. H. Gurney
L. A. Harding
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
F. D. Mensing
W. A. Rowe

1932
President.....F. B. Rowley
1st Vice-President.....W. T. Jones
2nd Vice-President.....C. V. Haynes
Treasurer.....F. D. Mensing
Secretary.....A. V. Hutchinson
Technical Secretary.....P. D. Close

Council
Chairman, F. B. Rowley
W. T. Jones, Vice-Chm.
D. S. Boyden
E. K. Campbell
R. H. Carpenter
W. H. Carrier
John D. Cassell
E. O. Eastwood
Roswell Farnham
F. E. Giesecke
E. H. Gurney
John Howatt
G. L. Larson
J. F. McIntire
F. D. Mensing
W. E. Stark

1933
President.....W. T. Jones
1st Vice-President.....C. V. Haynes
2nd Vice-President.....John Howatt
Treasurer.....D. S. Boyden
Secretary.....A. V. Hutchinson

Council
Chairman, W. T. Jones
C. V. Haynes, Vice-Chm.
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
R. Farnham
F. E. Giesecke
E. H. Gurney
E. H. Gurney
John Howatt
G. L. Larson
J. F. McIntire
F. C. McIntosh
L. W. Moon
F. B. Rowley
W. E. Stark

1934
President.....C. V. Haynes
1st Vice-President.....John Howatt
2nd Vice-President.....G. L. Larson
Treasurer.....D. S. Boyden
Secretary.....A. V. Hutchinson

Council
Chairman, C. V. Haynes
John Howatt, Vice-Chm.
M. C. Beman
D. S. Boyden
Albert Buenger
R. H. Carpenter
J. D. Cassell
F. E. Giesecke
E. H. Gurney
W. T. Jones
G. L. Larson
J. F. McIntire
F. C. McIntosh
L. Walter Moon
O. W. Ott
W. A. Russell
W. E. Stark

Council
Chairman, John Howatt
G. L. Larson, Vice-Chm.
M. C. Beman
D. S. Boyden
Albert Buenger
R. H. Carpenter
J. D. Cassell
F. E. Giesecke
E. H. Gurney
C. V. Haynes
J. F. McIntire
F. C. McIntosh
L. Walter Moon
Offner, A. J.
O. W. Ott
W. A. Russell
W. E. Stark

