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AIR CONDITIONING

By

HERBERT HERKIMER
Consulting and Industrial Engineer
Past Director, Herkimer Institute of
Mechanical Trades

and

HAROLD HERKIMER
Design Engineer and
Project Manager,
Raisler Corporation

First published in Great Britain

1949

CHAPMAN & HALL LTD.

37 ESSEX STREET, W.C.2

657.9

Acc. 1153A

32768

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1947

CHEMICAL PUBLISHING CO., INC.
BROOKLYN NEW YORK

Printed in U.S.A.

PREFACE

Briefly defined, air conditioning means the production of controlled artificial weather for a special purpose.

One purpose is to provide, in manufacturing, atmospheric conditions conducive to a standard quality and quantity of product.

Another purpose is to provide air at ideal temperature and humidity wherever people gather indoors.

A humid indoor atmosphere is required for certain industrial processes, e.g., in textile mills, bakeries, cigar factories, etc.

A dry indoor climate is desirable in the candy, match, photographic chemical, drug, explosive industry, etc.

Air conditioning can be applied to any industry or business if there exists a need of rectifying a condition of the normal atmosphere.

Industrial plants that would otherwise be compelled to shut down due to unfavorable weather conditions, if equipped with scientifically controlled air-conditioning equipment, may operate on a year-round schedule.

The successful selection of satisfactory equipment demands an exceedingly high degree of practical engineering judgment, combined with a clear conception of the fundamental laws of chemistry and physics.

Chemists and engineers, specializing in a particular field, when confronted suddenly with an air-conditioning problem, will find it absolutely necessary to recall their working knowledge of chemistry and physics, perhaps hazy from lack of use.

More than thirty years ago, I began to follow the advice given to student engineers at a Cornell University lecture. "Every engineer should make his own engineers' pocket manual, as he proceeds in study and practice, to suit his particular business."

I have been engaged in the air-conditioning industry since 1912 and have collected a mass of notes and personal empirical formulas.

At the request of the publishers, this accumulated data has been arranged for publication.

Furthermore, I have made careful study of the latest published transactions of the two leading technical societies of the industry, The American Society of Heating and Ventilating Engineers and The American Society of Refrigerating Engineers.

In order to insure that no important developments had been overlooked, selections were made from representative manufacturers' technical literature.

The first part of this book reviews the laws of chemistry and physics associated with the science of air conditioning.

The second part treats the more practical aspects of the subject, e.g., equipment, material and costs.

I am indebted to my son, Harold Herkimer for assistance in preparation of special sections, and general arrangement.

HERBERT HERKIMER

CONTENTS

INTRODUCTION

Air conditioning means simultaneous control of any or all of the physical and/or chemical properties of atmospheric air:

1. Temperature
2. Humidity
3. Purity
4. Distribution

Since the physical characteristics of air vary from hour to hour, day to day and month to month, the process of air conditioning must also change.

The functions of an air-conditioning system may also be classified as follows:

1. Year-round circulating and cleaning
2. Heating cycle—heating and humidifying
3. Cooling cycle—cooling and dehumidifying

In the pioneering period, regulation of these functions was attempted by manual control, but it was soon proved that even only partially automatic control provided more reliable and economical means of maintaining specified conditions.

Today automatic controls are considered an inherent part of the system itself, rather than an accessory, which may be credited to developments brought about by the National Defense Program.

Presuming that the prospective reader of this book is not entirely conversant with the purely industrial uses of air conditioning, a partial list is offered of over 200 different industries. In these industries, it has been demonstrated that vagaries of weather can be overcome in controlling:

1. Rates and limits of chemical reactions in materials.
2. Biological or bacteriological processes.

3. Physical conditions or properties of hygroscopic substances.

4. Physical conditions or properties of non-hygroscopic products.

5. Physiological reactions for human health, efficiency and production.

Industries and Materials Utilizing Air Conditioning

Automobiles
- Inner tubes

Breweries
- Fermentation room

Chemicals
- Baking powder
- Powder or other explosives

Clay Products
- Not including the simpler and coarser products

Comfort
- All places of assembly
- Banks
- Churches
- Department stores
- Dance halls
- Hotels in warm climate
- Legitimate theaters
- Laundries
- Moving picture theaters
- Offices
- Public dining and grille rooms
- Pullman cars
  - (Sleepers)
  - (Diners)
- Residences
- Restaurants
- Safe deposit vaults

Confectionery
- Bon bons
- Chocolate
- Caramels
- Chewing gum
- Gum drops
- Hard candy
- Marshmallows
- Starch and various sugars

Drugs and Pharmaceuticals
- Capsules
- Hygroscopic colloidals
- Many crystals
- Penicillin
- Serum

Electrical Goods
- Toll cable departments
- Telephone exchanges

Films and Film Laboratories
- Drying cabinets
- Moving picture studios
- Perforating rooms
- Projection assembly rooms
- Printing rooms

Foods
- Bread and cake
- Fish
- Fruits (particularly bananas)
- Macaroni
- Meats
- Mushroom growing
- Some cereals
- Yeast
- Other enzymatic products

Furs
- Felt hat manufacturing plants
- Fur storage

Incubators
- Baby
- Pneumonia enclosures

Laboratories
- All kinds

Laundries

Linoleum

Matches
- Drying rooms
- Machine rooms

Metal Enamelling
- Including mottled wear

Minerals

Gold beaters and gold and silver leaf

Nylon

Paper Products

Particularly wax paper

Pearls

Artificial

Printing, Lithography and Rotogravure

Playing cards

Other process work

Rayon

Acetate method

Celanese method

Cuprammonium method

Chemical house

Spinning

Twisting

Reeling

Winding

Storage

Inspection

Viscose method

Rubber—(synthetic)

Silk

Throwing

Soap

Crystallizing under the cold process

Steel

Blast furnaces and pig iron

Textiles

Tin and lead weighting

Regain rooms (where exact regains are required)

Tobacco

All machine processes for cigars and cigarettes

Chapter I

GAS LAWS

The word gas itself comes from the Greek word, chaos; this in turn comes from a verb signifying to yawn, to gape or to open widely. The conception of chaos in the Greek mind was an infinite formlessness, a confused, unorganized state of matter which was conceived to exist before the creation of order and distinct objects. The etymology of the word expresses the confusion of thought with regard to gases and the fact that their true nature was a puzzle to scientific thinkers for centuries. The exact relation between the pressure of a gas and its volume was first explained in 1660 by Robert Boyle in his book *Experiments on the Spring of Air*. He stated that, at constant temperature, the volume of a gas varies inversely with the pressure. Charles and Gay-Lussac made the discovery that the volume of a gas, at constant pressure, varies directly as the absolute temperature. Dalton formulated the law of partial pressures: the total pressure of a mixture of gases is the sum of the pressures which each gas would exert independently. This law has appeared in many air-conditioning calculations, since, in the case of moist air at ordinary atmospheric pressures, it gives a close approximation to the truth. Avogadro, in 1811, concluded from his studies that equal volumes of gases, at the same temperature and pressure, contain the same number of molecules. These laws are used in engineering computations.

1. Air-Conditioning Applications

In air-conditioning processes, certain laws of physics and chemistry are applied. As the name implies, the air-conditioning engineer works primarily with a mixture of gases called air. The composition of dry air is subject to such slight variation that it may be considered constant. Another gas with which the air-conditioning engineer deals, is water vapor, a variable constituent of the atmosphere.

2. Specific Volume and Density

Specific volume, v_s, is the volume of unit mass. In the English system, mass is expressed in pounds, and volume in cubic feet, specific volume being expressed in cubic feet per pound. Density, d, is mass per unit volume and equals the reciprocal of specific volume. The unit of density is pounds per cubic foot.

3. Boyle's Law

The volume of a gas at constant temperature is inversely proportional to its pressure.

If v_1 denotes the volume at pressure p_1, and the temperature remains constant the volume v_2 at pressure p_2 is expressed approximately as

$$v_2 = v_1 \left(\frac{p_1}{p_2}\right) \quad (1)$$

for gases at not too high pressures.

The following example illustrates the practical application of this law.

Example 1. A gas occupies a volume of 100 cu ft at 20 lb gage pressure. What volume will it occupy if pressure is reduced to 10 lb gage?

Solution. In equation (1)

$$\begin{aligned} v_2 &= \text{unknown} \\ v_1 &= 100 \text{ cu ft} \\ p_1 &= 14.7 + 20 = 34.7 \text{ lb absolute} \\ p_2 &= 14.7 + 10 = 24.7 \text{ lb absolute} \\ v_2 &= 100\left(\frac{34.7}{24.7}\right) = 140 \text{ cu ft} \end{aligned}$$

Absolute pressure is the pressure referred to a perfect vacuum and is the sum of gage pressure and barometric pressure. Zero gage pressure is equivalent to 14.7 lb absolute.

4. Charles' Law and Absolute Zero

If dry air or any gas is heated through 1° C, the expansion is $\frac{1}{273.1}$ or 0.003665 of its original volume; if heated through 1° F, the expansion is $\frac{1}{491.6}$ of its original volume. If a gas is cooled it will contract to the same degree. If air could be cooled to −273.1° C or −459.6° F, all molecular motion would cease, i.e., it would have zero volume. According to Charles' law, a gas at absolute zero temperature has zero volume, but all gases become liquids before reaching that temperature, and this law does not apply to liquids. Charles' law is usually stated as follows:

The volume of a gas at constant pressure varies directly as the absolute temperature.

If v denotes the volume at absolute temperature T, and the pressure remains constant, the specific volume v_2 at the absolute temperature T_2 is expressed approximately as

$$v_2 = v_1 \left(\frac{T_2}{T_1}\right) \tag{2}$$

Absolute temperature is expressed as degrees Kelvin (0° C = 273.1°K) or degrees Rankine (0° F = 459.6° R); round numbers (273,460) are generally used in effecting the conversion.

Example 2. If 100 cc of a gas are heated from 10 to 50° C what volume will the gas occupy?

Solution. In equation (2)

$$\begin{aligned} V_2 &= \text{Unknown} \\ V_1 &= 100 \text{ cc} \\ T_1 &= 273° + 10° = 283° \text{ K} \\ T_2 &= 273° + 50° = 323° \text{ K} \end{aligned}$$

$$V_2 = 100 \times \frac{323}{283} = 114.1 \text{ cc}$$

5. The General Gas Law

This law refers to changes in both pressure and temperature. The laws of Charles and Boyle may be combined to calculate simultaneous change in any two of the three quantities, volume, pressure and temperature. The usual form of expressing the combined law is

$$\frac{PV}{T} = \frac{P'V'}{T'} \tag{3}$$

In this equation P, V, and T represent pressure, volume and absolute temperature of the gas in its original condition, and P', V', and T' represent the pressure, volume and absolute temperature of the gas in its final state.

Example 3. A quantity of gas at 50° F occupies 100 cu ft at 20 lb gage pressure. What gage pressure is required to make it occupy 150 cu ft at 30° F?

Solution. In equation (3)

Original conditions	Final conditions
$P = 20 \text{ lb} + 14.7 \text{ lb} = 34.7 \text{ lb}$ absolute	$P' =$ unknown
$V = 100$ cu ft	$V' = 150$ cu ft
$T = 50° + 460° = 510°$ R	$T' = 460° + 30° = 490°$ R

$$\frac{34.8 \times 100}{510} = \frac{P' \times 150}{490}$$

$$P' = 22.3 \text{ lb absolute}$$

To reduce to gage pressure subtract 14.7 lb

$$22.3 \text{ lb} - 14.7 \text{ lb} = 7.6 \text{ lb gage}$$

Example 4. A quantity of gas occupies 1 cu ft at 9.92 in. vacuum gage pressure and 68° F. What is the gage pressure required to expand the gas to 2 cu ft at −132° F?

Solution. In equation (3)

Original conditions	Final conditions
$P = 29.92 - 9.92 = 20$ in. Hg absolute	$P' =$ unknown
$V = 1$ cu ft	$V' = 2$ cu ft
$T = 460° + 68° = 528°$ R	$T' = 460° - 132° = 328°$ R

$$P' = 6.2 \text{ in. Hg absolute}$$
$$P' = 29.92 \text{ in.} - 6.2 \text{ in.} = 23.72 \text{ in. Hg vacuum gage}$$

In this case 29.92 in. Hg pressure = 0 in. gage pressure; vacuum gage is a negative quantity.

6. Terminology

A brief glossary of physical terms referring to pressure and temperature is presented here as a convenience in the solution of examples.

Gage pressure. Pressure measured with reference to 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. The manometer is a U-tube partially filled with a liquid; with water for measurement of small air pressures, or with mercury (Hg) for indicating higher pressures.

Inch of water. The pressure exerted at 60° F by a column of water 1 in. high.

Vacuum. Any pressure below atmospheric; commonly expressed in inches of mercury.

Absolute pressure. The sum of gage and barometric pressures.

Specific volume. The volume, expressed in cubic feet, of one pound of a substance.

$$v_s = \frac{1}{d} = \frac{V}{W}$$

Vapor pressure. The pressure exerted by a vapor, either by itself or in a mixture of gases. Often synonymous with saturated vapor pressure, which is the vapor pressure of a vapor in contact with its liquid form. The saturated vapor pressure increases with rise of temperature. A graph of these pressures plotted against corresponding temperatures is known as a vapor-pressure curve.

Temperature conversion factors. To convert Fahrenheit temperatures to Centigrade, subtract 32° and multiply by $\frac{5}{9}$ or 0.555. To convert Centigrade temperatures to Fahrenheit, multiply by $\frac{9}{5}$ or 1.8 and add 32°.

TABLE 1

PRESSURE CONVERSION FACTORS

One pound per square inch is equivalent to

144	lb per square foot
2.0355	in. Hg at 32° F
2.0416	in. Hg at 62° F
2.309	ft water at 62° F
27.71	in. water at 62° F

7. The Universal Gas Law

At sufficiently high temperatures, low pressures and in the absence of any chemical reaction, all gases approach such a condition that their p-v-t relationships may be expressed by the equation of Charles and Gay-Lussac

$$pv = RT \tag{4}$$

where R is a constant, characteristic of the given gas. Modern refined pressure measurements have shown that this law is approximately correct and the error is extremely small. Therefore, equation (4) is satisfactory for practical calculations. If v is expressed in specific volume, or volume per unit weight, the value of the gas constant R in equation (4) is different for every gas. If v is expressed as the volume of one molecular weight as explained in topic 14 then R is the same for all gases in any selected system of units. For a given quantity of gas the universal gas equation may be written

$$pv = nMRT \tag{5}$$

where v is the total gas volume, n is the number of molecular weights of gas in volume v, M is the molecular weight and MR the universal gas constant.

If the quantity of gas is given as w pounds the equation is expressed

$$pv = wRT \tag{6}$$

where P = absolute pressure of gas in pounds per square foot
v = volume of weight W in cubic feet
w = weight of gas in pounds
T = absolute temperature, ° R
R = 53.351 for dry air
R = 85.782 for water vapor
Where p = absolute pressure of dry air in inches Hg
v = volume in cubic feet per pound
T = absolute temperature, ° R
the gas constant R for dry air is 0.7543.

One pound mol of any gas occupies a volume of 358.7 cu ft at 32° F and standard pressure.* For many air-conditioning engineering computations use of the gas laws is permissible up to a pressure of 200 lb per square inch, if the absolute temperature is twice the minimum critical temperature. Below critical temperatures, errors may be neglected up to a pressure of 15 lb per square inch, although errors of 5% are often encountered in calculations involving saturated vapors.

Example 5. Prove R = 53.3 in equation (6) when the quantity of air is given in pounds, temperature in degrees Fahrenheit

* See topic 8.

and absolute pressure in pounds per square foot; assume that one cubic foot of dry air at 32° F and standard pressure weighs 0.08077 pounds.

Solution. Air expands $\frac{1}{492}$ of its volume per degree rise in temperature. The volume of 1 lb of dry air is $\frac{1}{0.08077}$ or 12.38 cu ft. Substituting in equation (6)

$$R = \frac{pv}{wT} = \frac{14.7 \times 144 \times 12.38}{1 \times 492} = 53.3$$

8. Standard Conditions

The air-conditioning engineer is generally interested in the density of air. Density varies with pressure and temperature. Chemists and physicists have agreed, for the sake of uniformity, to measure gases under standard conditions, or to correct their measurements, for comparative purposes, to these conditions. A temperature of 32° F or 0° C has been adopted as the standard temperature, and one atmosphere, the average pressure of air at sea level, as the standard pressure. One standard atmosphere has the following equivalents:

TABLE 2

STANDARD ATMOSPHERE EQUIVALENTS

14.695	lb per sq in.
29.921	in. of mercury (Hg)
760	mm of mercury (Hg)
1033.243	g (grams) per sq cm
1,013,191	dynes

Air-conditioning engineers have adopted as a standard an air density of 0.075 lb per cubic foot. The density of air at a barometric pressure of 29.92 in. of mercury, 68° F dry-bulb and 50% relative humidity is 0.07497 lb per cubic foot and that of dry air at 70° F dry-bulb is 0.07496 lb per cubic foot.

TABLE 3

WEIGHT OF GASES AT STANDARD CONDITIONS (14.693 LB/SQ IN. AT 32° F OR 29.92 IN. HG AT 0° C)

Name of Gas	Col. 1 Atomic Weight	Col. 2 Molecular Weight	Col. 3 Chemical Symbols	Col. 4 Density Pounds Per Cu Ft	Col. 5 Density Grams Per Liter
Xenon	131.30	131.30	Xe	0.365	5.85
Phosgene		98.90	$COCl_2$	0.274	4.4172
Chlorine	35.45	70.914	Cl_2	0.199	3.214
Nitrous oxide		44.02	N_2O	0.124	1.978
Argon	39.94	39.94	A	0.111	1.783
Fluorine	19.00	38.00	F_2	0.106	1.700
Hydrogen chloride		36.46	HCl	0.101	1.6392
Hydrogen sulfide		34.06	H_2S	0.095	1.539
Oxygen	16.00	32.00	O_2	0.08921	1.429
Nitric oxide		30.00	NO	0.0836	1.340
Air (Dry)		28.967	Mixture	0.08077	1.2929
Air (Saturated)		28.834	"	0.08026	1.2856
Nitrogen	14.00	28.00	N_2	0.07803	1.2506
Carbon monoxide		28.00	CO	0.078	1.2504
Acetylene		26.00	C_2H_2	0.0727	1.173
Neon	20.183	20.183	Ne	0.0562	0.9
Helium	4.002	4.002	He	0.011237	0.1785
Hydrogen	1.0078	2.0156	H_2	0.005606	0.0898

Note: Column 5 is obtained by dividing column 2 by 22.4.
Column 4 is obtained by multiplying column 5 by 0.06243.

TABLE 4

WEIGHT OF REFRIGERANT GASES AT 32° F AND ONE ATMOSPHERE PRESSURE

Name of Gas	Col. 1 Molecular Weight	Col. 2 Chemical Symbols	Col. 3 Pounds Per Cu Ft	Col. 4 Grams Per Liter
Air (Dry)	28.967	Mixture	0.08077	1.2929
Ammonia	17.032	NH_3	0.048	0.77
Butane (n)	58.121	C_4H_{10}	0.167	2.67
Carbon dioxide	44.01	CO_2	0.124	1.98
{Dichloro-methane / Methylene chloride}	84.94	CH_2Cl_2	0.238	3.79
Dichlorodifluoromethane (Freon-12)	120.924	CCl_2F_2	0.337	5.40

TABLE 4 (*continued*)

Name of Gas	Col. 1 Molecular Weight	Col. 2 Chemical Symbols	Col. 3 Pounds Per Cu Ft	Col. 4 Grams Per Liter
Dichloromonofluoro-methane (Freon-21)	102.930	$CHCl_2F$	0.286	4.59
Dichlorotetrafluoro-ethane (Freon-114)	170.930	$C_2Cl_2F_4$	0.476	7.63
Dieline	96.93	$C_2H_2Cl_2$	0.269	4.30
Ethyl chloride	64.518	C_2H_5Cl	0.180	2.88
Ethyl ether	74.077	$(C_2H_5)_2O$	0.225	3.30
Isobutane	58.121	C_4H_{10}	0.162	2.59
Methyl chloride	50.491	CH_3Cl	0.144	2.25
Methyl formate	60.03	$C_2H_4O_2$	0.168	2.66
Methane	16.04	CH_4	0.045	0.72
Propane	44.06	C_3H_8	0.126	2.19
Sulfur dioxide	64.065	SO_2	0.183	2.86
Trichloromonofluoro-methane (Freon-11)	137.38	CCl_3F	0.382	6.13
Trichlorotrifluoro-ethane (Freon-113)	187.39	$C_2Cl_3F_3$	0.521	8.35
Trieline	96.934	C_2HCl_3	0.269	4.30
Water	18.016	H_2O	0.0505	0.804

Note: Column 4 obtained by dividing column 1 by 22.4.
Column 3 obtained by multiplying column 4 by 0.06243

9. Avogadro's Law and Law of Multiple Proportions

This law states that equal volumes of gases, at the same temperature and pressure, contain equal numbers of molecules.

10. Atomic Weights

The law of definite proportions is the basis of the system of atomic weights. The ratio of combining weights can only be explained by the theory that chemical compounds are composed of units of matter which have definite weights. These small units are called atoms. Chemists have analyzed many compounds and have recorded the ratios of the weights of the elements making up each compound. The atomic weight of each element has been determined from the collected data. Hydrogen has the lowest atomic weight. If the value 16 is adopted as

Table 5

CONSTANTS OF VAPORS AND GASES

Gas	Chemical Formula	Molecular Weight (1939)	Specific Gravity at 29.92 in Hg and 60°F Air = 1	Density Lb/Cu Ft at 29.92 in Hg and 60°F	Volume Cu Ft/Lb at 29.92 in Hg and 60°F	Specific Heat at Constant Pressure at 29.92 in Hg and 60°F/Cp	Specific Heat at Constant Volume at 29.92 in Hg and 60°F/Cv	** Cp/Cv at 29.92 in Hg and 60°F	Boiling Point at Atmospheric Pressure °F	Critical Temperature °F	Critical Pressure Lb/Sq In. absolute	Critical Volume Cu Ft/Lb
Acetylene	C_2H_2	26.0362	0.91	0.069	14.5	0.383	0.304	1.26–1.29	—118	97	911	0.0690
Air (Dry)	—	—	1.000	0.0764	13.100	0.2375	0.1689	1.406	—318	221	547	0.0458
Ammonia	NH_3	17.0323	0.596	0.045	22	0.52	0.39	1.31–1.32	—28	—270	1639	0.0682
Argon	A	39.944	1.379	0.105	9.49	0.123	0.074	1.66–1.67	—302.3	—187.6	705.4	0.0302
Benzene	C_6H_6	78.1086	—	—	—	—	—	1.1–1.4*	176	551	701	0.0527
Butane (n)	C_4H_{10}	58.121	2.067	0.158	6.32	—	—	1.108–1.112	31	307	551	0.0691
Butene—1	C_4H_8	56.1048	1.93	0.148	6.76	—	—	—	23	—	—	—
Butene—2	C_4H_8	56.1048	2.00	0.153	6.56	—	—	—	34	—	—	—
Carbon dioxide	CO_2	44.010	1.53	0.117	8.55	0.203	0.156	1.30	—109	88	1073	0.0348
Carbon disulfide	CS_2	76.13	—	—	—	—	—	1.2*?	115	523.4	1117	0.0363
Carbon monoxide	CO	28.010	0.967	0.074	13.5	0.24	0.17	1.40–1.404	—314	—218	515	0.0515
Carbon tetrachloride	CCl_4	153.838	—	—	—	—	—	1.13–1.18*	170.2	541.7	661	0.0287
Chlorine	Cl_2	70.914	2.486	0.190	5.27	0.116	0.086	1.32–1.35	—30.3	—291.2	1118	0.0280
Dichlorodifluoromethane (F-12)	CCl_2F_2	120.924	4.25	0.325	3.07	—	—	1.138?	—21.7	232.7	582.0	0.0289
Dichloromonofluoromethane (F-21)	$CHCl_2F$	102.9321	—	0.284	3.52	—	—	1.12?	48.0	353.3	744.0	—
Dichlorotetrafluoroethane (F-114)	$C_2Cl_2F_4$	170.934	—	0.465	2.20	—	—	1.09?	38.4	294.3	474.0	—
Dichloromethane	CH_2Cl_2	84.9402	—	—	—	—	—	1.18–1.224*	104	421	1490	—
Ethane	C_2H_6	30.0686	1.05	0.08	12.5	—	—	1.22	—127	90	717	0.0756
Ethyl chloride	C_2H_5Cl	64.518	2.26	0.173	5.78	0.275	0.231	1.13–1.19	53.9	369.0	764.2	0.0485
Ethylene	C_2H_4	28.0524	0.975	0.0746	13.4	—	—	1.18–1.25	—155	50	748	0.0729
Helium	He	4.003	0.138	0.0105	94.9	1.25	0.75	1.66	—452.1	—450.2	33.2	0.2311
Hexane (n)	C_6H_{14}	86.1734	—	0.227*	4.4*	—	—	1.08–1.09*	156	455	434	0.0685
Heptane (n)	C_7H_{16}	100.1996	—	—	—	—	—	—	209	512.2	393.9	0.0685

CONSTANTS OF VAPORS AND GASES (*continued*)

Gas	Chemical Formula	Molecular Weight (1939)	Specific Gravity at 29.92 in. Hg and 60°F Air = 1	Density Lb/Cu Ft at 29.92 in. Hg and 60°F	Volume Cu Ft/Lb at 29.92 in. Hg and 60°F	Specific Heat at Constant Pressure at 29.92 in. Hg and 60°F/Cp	Specific Heat at Constant Volume at 29.92 in. Hg and 60°F/Cv	** Cp/Cv at 29.92 in. Hg and 60°F	Boiling Point at Atmospheric Pressure °F	Critical Temperature °F	Critical Pressure Lb/Sq In. absolute	Critical Volume Cu Ft/Lb
Hydrogen	H_2	2.0162	0.0696	0.00532	187.85	3.41	2.42	1.41	—422.7	—400	188	0.5167
Hydrogen chloride	HCl	36.4651	1.268	0.097	10.33	0.193	0.137	1.39–1.41	—121.2	124.5	1199	0.0381
Hydrogen sulfide	H_2S	34.0762	1.19	0.091	11.0	0.25	0.19	1.32	—75.2	212.7	1306	—
Isobutane	C_4H_{10}	58.121	2.067	0.158	6.32	—	—	1.11?	14	273	544	0.0685
Isobutene	C_4H_8	56.1048	1.99	0.152	6.58	—	—	—	20	290	—	—
Isopentane	C_5H_{12}	72.1472	—	0.19*	5.27*	—	—	1.097*?	82	370	484	0.0686
Krypton	Kr	83.7	2.862	0.219	4.58	—	—	1.67–1.68	—241.2	—81.4?	793.6?	0.0205?
Methane	CH_4	16.0424	0.554	0.0424	23.57	—	—	1.290–1.316	—259	—117	673	0.0989
Methyl chloride	CH_3Cl	50.4913	1.785	0.136	7.34	0.24	0.20	1.20–1.28	—11.3	289.6	967	0.0433
Neon	Ne	20.183	0.696	0.0532	18.81	—	—	1.64	—410.7	—379.7	380.6	0.0331
Nitric oxide	NO	30.008	1.037	0.0792	12.63	0.232	0.166	1.394–1.41	—239.9	—137.2?	955.2?	0.0308?
Nitrogen (atm)	N_2	28.016	0.9718	0.0744	13.44	0.247	0.176	1.404–1.41	—320.4	—232.8	492.3	0.0515
Nitrous oxide	N_2O	44.016	1.529	0.1141	8.76	0.213	0.163	1.303–1.31	—129.1	97.7	1053.7	0.0355
Oxygen	O_2	32.000	1.1053	0.0846	11.82	0.217	0.155	1.398–1.401	—297.4	—182	731	0.0373
Pentane	C_5H_{12}	72.1472	—	0.19*	5.3*	—	—	1.086–1.097*	97	387	485	0.0688
Propane	C_3H_8	44.0948	1.56	0.119	8.36	—	—	1.10–1.136	—44	206	617	0.0709
Propylene	C_3H_6	42.0786	1.45	0.11	9.0	—	—	1.146?	—54	198	662	—
Sulfur dioxide	SO_2	64.06	2.264	0.1733	5.77	0.154	0.123	1.25–1.29	14.0	315.0	1142	0.0308
Trichloromonofluoromethane (F-11)	CCl_3F	137.381	—	0.388*	2.58*	—	—	1.135*?	74.67	388.4	636.0	—
Trichlorotrifluoroethane (F-113)	$C_2Cl_3F_3$	187.391	—	0.518*	1.93*	—	—	1.09*?	117.9	417.4	499.0	—
Water vapor	H_2O	18.0162	—	0.0374*	26.80*	0.48*	0.36*	1.27–1.33*	212.0	705.4	3206.2	0.0503
Xenon	Xe	131.3	4.525	0.3455	2.89	—	—	1.66–1.67	—164.4	61.9	855.3	0.0139

* At boiling point.

** A range of values is given for Cp Cv where values of various authorities differ. A question mark indicates that insufficient independent sources of information are available.

Based on data from M. W. Kellogg Co., Kinetic Chemicals, Inc., International Critical Tables, U. S. Bur. of Stds. Scientific Paper, No. 541 and Keenan & Keys.

the atomic weight of oxygen the atomic weight of hydrogen is exactly 1.008. This is so very close to 1. that for practical purposes the atomic weight of hydrogen may be used as 1 with oxygen as 16. The following table gives the international atomic weights of the elements.

TABLE 6

THE CHEMICAL ELEMENTS

(From National Bureau of Standards Mis. Pub. M 126)

Atomic symbol	Name	Melting point, °C.	Atomic weight
Ac	Actinium	1600	—
Al	Aluminum	660.0	26.97
Sb	Antimony	630.5	121.76
A	Argon	—189.3	39.944
As	Arsenic	814	74.91
Ba	Barium	704	137.36
Be	Beryllium	1280	9.02
Bi	Bismuth	271.3	209.00
B	Boron	2300	10.82
Br	Bromine	—7.2	79.916
Cd	Cadmium	320.9	112.41
Ca	Calcium	850	40.08
C	Carbon	3700	12.00
Ce	Cerium	600	140.13
Cs	Cesium	28	132.91
Cl	Chlorine	—101	35.457
Cr	Chromium	1800	52.01
Co	Cobalt	1490	58.94
Cb	Columbium	2000	92.91
Cu	Copper	1083.0	63.57
Dy	Dysprosium	—	162.46
Er	Erbium	—	167.64
Eu	Europium	—	152.0
F	Fluorine	—223	19.00
Gd	Gadolinium	—	157.3
Ga	Gallium	29.78	69.72
Ge	Germanium	958	72.60

THE CHEMICAL ELEMENTS (*continued*)

Atomic symbol	Name	Melting point, ° C.	Atomic weight
Au	Gold	1063.0	107.2
Hf	Hafnium	1700	178.6
He	Helium	—271.4	4.002
Ho	Holmium	—	163.5
H	Hydrogen	—259.2	1.0078
	H_2 (normal)	—259.2	—
	HD	—256.5	—
	D_2 (normal)	—254.5	—
Il	Illinium	—	—
In	Indium	156.4	114.76
I	Iodine	114	126.92
Ir	Iridium	2454	193.1
Fe	Iron	1535	55.84
Kr	Krypton	—157	83.7
La	Lanthanum	826	138.92
Pb	Lead	327.4	207.22
Li	Lithium	186	6.940
Lu	Lutecium	—	175.0
Mg	Magnesium	650	24.32
Mn	Manganese	1260	54.93
Ma	Masurium	2700	—
Hg	Mercury	—38.87	200.61
Mo	Molybdenum	2625	96.0
Nd	Neodymium	840	144.27
Ne	Neon	—248.6	20.183
Ni	Nickel	1455	58.69
N	Nitrogen	210.0	14.008
Os	Osmium	2700	191.5
O	Oxygen	—218.8	16.0000
Pd	Palladium	1554	106.7
P	Phosphorus, Y	44.1	31.02
	Phosphorus, R	590	—
Pt	Platinum	1773.5	195.23
Po	Polonium	600	—
K	Potassium	63	39.096
Pr	Praseodymium	940	140.92
Pa	Protoactinium	3000	231
Ra	Radium	700	226.05

THE CHEMICAL ELEMENTS (*continued*)

Atomic symbol	Name	Melting point, ° C.	Atomic weight
Rn	Radon	—71	222
Re	Rhenium	3000	186.31
Rh	Rhodium	1966	102.91
Rb	Rubidium	39	85.44
Ru	Ruthenium	2500	101.7
Sm	Samarium	>1300	150.43
Sc	Scandium	1200	45.10
Se	Selenium	220	78.96
Si	Silicon	1430	28.06
Ag	Silver	960.5	107.880
Na	Sodium	97.7	22.997
Sr	Strontium	770	87.63
S	Sulfur:		32.06
	Monoclinic	119.2	—
	Rhombic	112.8	—
Ta	Tantalum	3000	180.88
Te	Tellurium	450	127.61
Tb	Terbium	327	159.2
Tl	Thallium	300	204.39
Th	Thorium	1800	232.12
Tm	Thulium	—	169.4
Sn	Tin	231.9	118.70
Ti	Titanium	1820	47.90
W	Tungsten	3410	184.0
U	Uranium	3600	238.14
V	Vanadium	1735	50.95
Xe	Xenon	—112	131.3
Yb	Ytterbium	—	173.04
Y	Yttrium	1490	88.92
Zn	Zinc	419.5	65.38
Zr	Zirconium	1750	91.22
—	Element 85	250	—
—	Element 87	23	—

11. Molecular Weights

Molecular weight is determined by adding the atomic weights of the atoms which make up the molecule.

Example 6. What is the molecular weight of water? The formula for water is H_2O.

Solution. Molecular weight $= 2 \times 1 + 16 = 18$.

Example 7. What is the molecular weight of the refrigerant dichlorodifluoro-methane (commonly called Freon-12 or F-12)? The formula for Freon-12 is CCL_2F_2.

Solution. Molecular weight $= 12.010 + 2 \times 35.45$
$$+ 2 \times 19.00 = 120.91.$$

12. Weights of Gases

Equal volumes of gases do not have equal weights under equal conditions of pressure and temperature, even though equal volumes contain the same number of molecules One cubic foot of hydrogen weighs less than one cubic foot of oxygen because each molecule of hydrogen weights less than a molecule of oxygen. The oxygen molecule is approximately 16 times heavier than the hydrogen molecule. Since one cubic foot of each of these gases contains the same number of molecules, the weight of one cubic foot of oxygen is approximately 16 times that of one cubic foot of hydrogen. A cubic foot of oxygen weighs 0.0898 pounds and a cubic foot of hydrogen weighs 0.005606 pounds.

$$0.0898 \div 0.005606 = 16.01$$

The weights of equal volumes of gases are in the same ratio as their molecular weights.

13. Laboratory Method of Finding Weights of Gases

In air conditioning the English system of units is generally used. Most of the problems so far have been given in this system of units to acquaint the reader with this custom. However, in the discussion of intrinsically chemical problems the metric system will be used.

At standard conditions the molecular weight of a gas occupies 22.4 l. In the laboratory the molecular weight of a gas can be derived by weighing one liter and calculating its volume under standard conditions. The weight of one liter of hydrogen at standard conditions is known to be 0.0898 grams; when the weight of the unknown gas is divided by 0.0898, the result will show how much heavier than hydrogen is the unknown gas. This ratio is termed the vapor density, which may be defined as the weight of any volume of vapor (or gas) divided by the weight of an equal volume of hydrogen measured and weighed under the same conditions. Since the molecular weight of hydrogen is approximately 2, the molecular weight of the unknown gas is roughly twice the vapor density.

Example 8. At standard conditions one liter of an unknown gas weighs 0.77g. If one liter of hydrogen at standard conditions weighs 0.0898g, what is the molecular weight of the unknown gas?

Solution. Weight unknown gas under standard conditions $= 0.77$g.

Weight of 1 liter of hydrogen $= 0.0898$ g

Vapor density of unknown gas $= \dfrac{0.77}{0.0898} = 8.56$

Molecular weight of unknown gas $= 2 \times 8.56 = 17.12$
Molecular weight of ammonia gas $= 17.12$

Example 9. One liter of an unknown gas at standard conditions weighs 0.0898g. One liter of hydrogen at standard conditions weighs 0.0898g. What is the molecular weight of the unknown gas?

Solution. Weight unknown gas under standard conditions $= 0.0898$g

Weight of 1 liter of hydrogen $= 0.0898$ g

Vapor density of unknown gas $= \dfrac{0.0898}{0.0898} = 1$ (hydrogen)

Molecular weight of unknown gas $= 2 \times 1 = 2$
Molecular weight of hydrogen $= 2$
(Molecular weight of hydrogen more accurately $= 2.016$)

14. Gram-molecular Volume

Properties of gases are usually most readily correlated on the mol basis. A gram-mol is the weight in grams equal to the molecular weight. A pound-mol is the weight in pounds equal to the molecular weight. Thus one pound-mol of oxygen weighs 32 pounds, since the atomic weight of oxygen is 16 and the molecular weight of oxygen is $2 \times 16 = 32$.

Example 10. Solve for the volume in liters occupied by one gram-mol of oxygen. One liter of oxygen under standard conditions weighs 1.429 g. The atomic weight of oxygen is 16 and the molecular weight is $2 \times 16 = 32$. One gram-mol of oxygen weighs 32 g.

Solution. The volume occupied by 32 g (one gram-mol) of oxygen is $32 \div 1.429 = 22.4$ l.

Example 11. Solve for the volume in liters, occupied by one gram-mol of hydrogen. One liter of hydrogen under standard conditions weighs 0.0898 g. The atomic weight of hydrogen is 1.008 and the molecular weight is $2 \times 1.008 = 2.016$. One gram-mol of hydrogen weighs 2.016 grams.

Solution. The volume occupied by 2.016 g (one gram-mol) of hydrogen is $2.016 \div 0.0898 = 22.4$ l.

This volume, 22.4 l, is called the gram-molecular volume or the molal volume of a gas. The following rule applies in general: One gram-mol (molecular weight in grams) of any gas occupies 22.4 l under standard conditions. The number of molecules in 22.4 l of gas has been determined to be approximately 6×10^{23}, i.e., 6 followed by 23 ciphers. This is known as Avogadro's number.

Example 12. Using the English system of units, solve for the volume in cubic feet occupied by one pound-mol of oxygen. One cubit foot of oxygen under standard conditions weighs 0.08921 lb. The atomic weight of oxygen is 16 and the molecular weight is $2 \times 16 = 32$. One pound-mol of oxygen weighs 32 lb.

Solution. The volume occupied by 32 lb or one pound-mol of oxygen is $32 \div 0.08921 = 358.7$ cu ft.

The following rule applies in general when using the English system of units. One pound-mol of any gas occupies 358.7 cu ft under standard conditions of temperature and pressure.

15. Specific Gravity of Gases

The ratio of the weight of a given volume of a gas to the weight of the same volume of air is called the specific gravity of the gas.

The weight of a liter of air under standard conditions is approximately 1.293 g. A gram-molecular volume of air weighs 22.4×1.29 or 28.89 g. One cubic foot of dry air weighs 0.08077 lb, therefore a pound-molecular volume or pound-mol weighs 0.8077×358.7 or 28.97 lb. If the gram-molecular weight of a gas is less than that of air, the gas is lighter than air; if it is greater, the gas is heavier than air.

16. Molecular Weight of Atmospheric Dry Air

Dry air is itself a mixture of several gases of the composition by weight given in Table 7 (taken from the International Critical Tables).

TABLE 7

COMPOSITION OF DRY AIR

Gas	Percentage By Weight
Nitrogen	78.03
Oxygen	20.99
Carbon dioxide	0.03
Hydrogen	0.01
Noble gases	0.94
Mixture	100.00

TABLE 8

NOBLE GASES

Gas	Percentage of Air By Weight
Argon	0.933
Neon	0.0018
Helium	0.0005
Krypton	0.0001
Xenon	—
	0.944

TABLE 9

COMPOSITION OF DRY AIR

Gas	Mol Per Mol Dry Air		Grams Per Mol		Grams Per Mol Dry Air	Grams Per Gram Dry Air
Nitrogen	0.7803	×	28.016	=	21.861	0.7547
Oxygen	0.2099	×	32.000	=	6.717	0.2319
Carbon Dioxide	0.0003	×	44.003	≐	0.013	0.0004
Hydrogen	0.0001	×	2.016	=	0.000	0.0000
Argon	0.0094	×	39.444	=	0.376	0.0130
	1.0000				28.967	1.0000

Col 2—Gives the percentage by volume in that portion of the atmosphere which is at sea level.
Col 3—Gives molecular weights.
Col 4—The sum of figures in column 4 gives the apparent molecular weight of the mixture.
Col 5—Gives the percentage composition by weight.

17. Dalton's Law of Partial Pressures

Atmospheric air is a mixture of various gases and water vapor. Table 10 gives the percentage by volume of the constituents of the atmosphere at sea level.

TABLE 10

ATMOSPHERIC AIR AND WATER VAPOR MIXTURE

Constituent	Percentage By Volume
Nitrogen	77.56
Oxygen	20.86
Carbon dioxide	0.03
Hydrogen	0.01
Argon	0.94
Water vapor	0.6

The amount of water vapor in air varies under different conditions of climate and with the seasons and is an important constituent since it affects bodily comfort and greatly affects various hygroscopic materials. A mixture of dry air and water vapor, such as atmospheric air, obeys (for all practical purposes) Dalton's law of partial pressures, which states: *When two or more gases are mixed in a vessel, the total pressure is equal to the sum of the pressures which each component gas would exert if it occupied the vessel alone.*

18. Mixtures of Perfect Gases

Let V denote the total volume of the mixture; $w_1, w_2, w_3 \ldots$ the weights of the constituent gases; $R_1, R_2, R_3 \ldots$ the corresponding gas constants, and R_m the constant for the mixture. The partial pressures of the constituents, i.e., the pressures which the constituents would have if occupying the total volume V, are

$$p_1 = \frac{w_1 R_1 T}{V} \tag{7}$$

$$p_2 = \frac{w_2 R_2 T}{V}$$

$$p_3 = \frac{w_3 R_3 T}{V}$$

According to Dalton's law, the total pressure p of the mixture is the sum of the partial pressures; i.e., $p = p_1 + p_2 + p_3 \dots$ Let $w = w_1 + w_2 + w_3 \dots$ denote the total weight of the mixture; then

$$pV = wR_mT$$

$$R_m = \frac{w_1R_1}{w} + \frac{w_2R_2}{w} + \frac{w_3R_3}{w} \dots \tag{8}$$

$$\frac{p_1}{p} = \frac{w_1R_1}{wR_m}$$

$$\frac{p_2}{p} = \frac{w_2R_2}{wR_m} \quad \text{etc.} \tag{9}$$

Let V_1, V_2, V_3 denote the volumes which would be occupied by the constituents at pressure p and temperature T. Then $V = V_1 + V_2 + V_3 + \dots$ and the apparent molecular weight M_m of the mixture is

$$M_m = \frac{w_1V_1}{V} + \frac{w_2V_2}{V} + \frac{w_3V_3}{V} \tag{10}$$

$$\text{then } R_m = \frac{1545.4}{M_m} \tag{11}$$

under standard conditions. (See topic 8).

$$\text{Volume of 1 lb of gas} = \frac{358.7}{M} \tag{12}$$

$$\text{Weight of 1 cu ft of gas} = 0.002788\,M_m \tag{13}$$

$$M_m \text{ for dry air} = 28.901 \tag{14}$$

19. The Atmosphere in General

Water, as a colorless, invisible vapor, forms an important constituent of the air. Humidity is the term applied to the water vapor (moisture) content of air. Absolute humidity is the water vapor content per unit volume of moist air, expressed in pounds per cubic foot or grains per cubic foot. Relative humidity is the ratio of the partial pressure of the water vapor in the air to the saturation pressure of water vapor at the same temperature. The daily change of relative humidity is very pronounced. In general, it decreases with the diurnal increase of

TABLE 11

PARTIAL PRESSURES AND COMPOSITION OF SATURATED AIR

(32° F and 14.695 lb Absolute Pressure)

Gas	Mol Per Mol Air Mixture		Pounds Per Mol		Pounds Per Mol Mixture	Pounds Per Pound Mixture	Absolute Pressure Pounds Per Square Inch
Nitrogen	0.7756	×	28.016	=	21.729	0.7519	11.3974
Oxygen	0.2086	×	32.000	=	6.675	0.2310	3.0653
Carbon dioxide	0.0003	×	44.003	=	0.013	0.0004	0.0044
Hydrogen	0.0001	×	2.016	=	0.000	0.0000	0.0014
Argon	0.0094	×	39.944	=	0.375	0.0129	0.1381
Water vapor	0.0060	×	18.015	=	0.109	0.0037	0.0886
Air	1.0000				28.901	1.0000	14.6952

Column 2: Composition of dry air, mol percentage.
Column 3: Pound-molecular weight (358.7 cu ft).
Column 4: Apparent molecular weight.
Column 5: Percentage by weight.
Column 6: Partial pressures = col 2 × 14.695.

temperature. There is an early morning maximum and an afternoon minimum. The daily change in absolute humidity varies greatly for different places. There are two pronounced types of humidity the *maritime* and the *continental*. The *maritime* has the maximum at the warmest part of the day with an early morning minimum. The *continental* type has two maxima, one just before noon and the other late in the afternoon; the minimum occurs in the early morning. The annual variation of relative humidity is somewhat irregular. The maximum is in midwinter and minimum in early summer. The annual amplitude varies from 5% on the coasts to 30% in the interior of the continent.

The annual variation in absolute humidity is very similar to that in temperature. The minimum is in midwinter and the maximum in summer. The amplitude for the year is least on the coasts and greatest in the interior.

Table 12

ATMOSPHERIC PRESSURE AT DIFFERENT LEVELS

At sea level	14.7 lb per square inch
At 0.25 mile above sea level	14.02 lb per square inch
At 1 mile above sea level	12.02 lb per square inch
At 2 miles above sea level	9.8 lb per square inch

At 32° F and at sea level, one cubic foot of air weighs 0.08076 lb at a pressure of 14.7×144 or 2116.8 lb per square foot.

$$\frac{2116.8}{0.08076} = 26{,}213 \text{ ft}$$

The quantity 26,213 is called the height of the *homogeneous atmosphere*, i.e., the theoretical height which the atmosphere would need to possess, if incompressible and non-expansible in order to exert a pressure of 2116.8 lb per square foot or 14.7 lb per square inch.

Table 13

ATMOSPHERIC PRESSURE AND BAROMETER READINGS AT DIFFERENT ALTITUDES

Altitude above Sea Level Feet	Atmospheric Pressure Pounds Per Square Inch	Barometer Reading Inches of Mercury
0	14.69	29.92
500	14.42	29.38
1000	14.16	28.86
1500	13.91	28.33
2000	13.66	27.82
2500	13.41	27.31
3000	13.16	26.81
3500	12.92	26.32
4000	12.68	25.84
4500	12.45	25.36
5000	12.22	24.89
5500	11.99	24.43

ATMOSPHERIC PRESSURE AND BAROMETER READINGS AT DIFFERENT ALTITUDES (*continued*)

Altitude above Sea Level Feet	Atmospheric Pressure Pounds Per Square Inch	Barometer Reading Inches of Mercury
6000	11.77	23.98
6500	11.55	23.53
7000	11.33	23.09
7500	11.12	22.65
8000	10.91	22.22
8500	10.70	21.80
9000	10.50	21.38
9500	10.30	20.98
10000	10.10	20.58
10500	9.90	20.18
11000	9.71	19.75
11500	9.52	19.40
12000	9.34	19.03
12500	9.15	18.65
13000	8.97	18.29
13500	8.80	17.93
14000	8.62	17.57
14500	8.45	17.22
15000	8.28	16.88

Reprinted from National Advisory Committee for Aeronautics—report No. 246.

Chapter II

PHYSICAL AND CHEMICAL PROPERTIES OF MATTER

Matter occurs in three states, gaseous, liquid and solid. Physical properties of matter include melting and boiling points, specific gravity, crystal structure, solubility, etc. Chemical properties are those relating to chemical reactivity of matter, e.g., the ease with which a substance burns.

These properties concern the air-conditioning engineer when he deals with building materials, brines, and the many products made or handled in air-conditioned plants. The physical properties of substances are of greater importance to the air-conditioning engineer than the chemical properties.

20. Solutions

Some solids when placed in a liquid, e.g., sugar in water, are disaggregated. The molecules of the solid become free and wander through the liquid. The solid is said to dissolve in the liquid and the mixture, which soon becomes homogeneous, is called a solution. Gases are in many cases absorbed by a liquid and one liquid often mixes with another. When a liquid at a given temperature has dissolved the greatest possible amount of the substances to be dissolved, the resulting solution is said to be saturated. The dissolved substance is called the *solute* and the dissolving medium is called *the solvent.*

21. Suspensions

A substance is said to be in suspension when its particles are visible, when they settle out on standing, and when they are uniformly distributed throughout the liquid in which they are suspended. Muddy waters containing clay and fine crayon dust mixed with water are typical illustrations. The particles of substance which are in suspension are not in the form of single molecules, but are small clumps or aggregates of many molecules.

22. Solubility

The mass of a substance per unit mass of liquid in a saturated solution at a given temperature is a constant called the solubility of the substance at that temperature. In nearly every case, the solubility of solids increases rapidly with rise in temperature and the solubility of gases decreases with rise in temperature. Gases become more soluble as the pressure increases. The solubility of a solid is not changed appreciably by application of pressure to the liquid in contact with the solid, but the solubility of a gas in a liquid depends upon the surrounding pressure. If the pressure on a gas is doubled, twice as much of the gas will dissolve. This principle is known as Henry's law. Soft drinks are carbonated by forcing carbon dioxide gas under pressure into the bottle. The maximum quantity of water vapor which can be dissolved in the atmosphere at a particular temperature is also a constant for any particular barometric pressure. This constant is called the vapor pressure at saturation. Vapor pressure tables are the basic factors in preparing air-conditioning psychrometric data.

23. Eutectic Solutions and Freezing of Brines

A solution of any salt in water has a certain concentration at which the freezing point is lowest. A solution of this concentration is called a *eutectic mixture* and the temperature at which it freezes is the eutectic temperature.

Containers of suitable size and shape holding such frozen solutions are substituted for salt and ice in refrigerated trucks. Refrigeration is produced by the melting of the brine. The containers are frozen in rooms held at suitable low temperatures. This system eliminates the necessity of salting ice, thus overcoming damage to truck bodies by the corrosive action of salt. These cartridges are made of lead-coated iron. The container is usually a triangular prism 4.5 in. by 4.5 in. by 5.75 in. Each cartridge contains three quarts of solution and weighs 10 pounds. It is capable of producing 1000 Btu of refrigeration at 6° F. Calcium chloride solution is used in brine circulating systems principally as a secondary refrigerating medium. Circulation recommended is at the rate of 5 gpm per ton of refrigeration. Velocity through pipe lines is kept low, allowing a maximum friction loss of 3 lb per 100 ft of run.

TABLE 14

EUTECTIC POINTS OF AQUEOUS SOLUTIONS

Substance	Eutectic Concentration $\frac{\text{Weight of Solute}}{\text{Weight of Solution}}$	Eutectic Temperature, ° F
Ammonia (NH_3)	0.318†	—145†
	0.338†	—168†
Ammonium nitrate (NH_4NO_3)	0.428	+2.0
Ammonium chloride (NH_4Cl)	0.191	+4.3
Calcium chloride ($CaCl_2$)	0.324	—59.8
Sodium chloride ($NaCl$)	0.233	—6.0
Sodium sulfate (Na_2SO_4)	0.049	+30.0
Sodium nitrate ($NaNO_3$)	0.397	+0.6
Sodium carbonate (Na_2CO_3)	0.080	+28.2
Potassium chloride (KCl)	0.197	+12.8
Potassium sulfate (K_2SO_4)	0.065	+28.3
Potassium nitrate (KNO_3)	0.112	+26.6
Potassium carbonate (K_2CO_3)	0.355	—34.7
Methyl alcohol (CH_4O)	0.695	—164
Ethyl alcohol (C_2H_6O)	0.763	—94
Glycerin ($C_3H_8O_3$)	0.669	—60.4

A solution of 30 lb of calcium chloride per gallon is generally allowed for in estimating for 5° F brine. In small refrigerating units other non-freezing solutions are used, e.g., alcohol, glycerin or Prestone mixed with water. The principal problem arising from the use of calcium chloride is corrosion.

TABLE 14A

FREEZING POINTS OF AQUEOUS SOLUTIONS

Alcohol		Glycerin		Ethyl Glycol	
%	° F	%	° F	%	° F
2.5	30.2	10	29.1	12.5	25
6.8	26.6	20	23.4	17.0	20
13.8	21.0	30	14.9	25.0	10
17.5	16.3	40	4.3	32.5	0
20.3	12.9	50	—9.4	38.5	—10
24.2	6.8	60	—30.5	44.0	—20
29.9	—2.0	70	--38	49.0	—30
39.0	—19.7	80	—5.5	52.5	—40
56.1	—41.8	90	29.1		
		100	62.6		

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24. Diffusion

When two liquids which are miscible are brought carefully together so as not to mix by convection, it is found that in the course of time they become uniformly mixed. This phenomenon is called diffusion; it shows that the molecules of liquids wander. Gases and vapors mix in the same way but with greater rapidity.

25. Diffusion Through Membranes; Osmotic Pressure

Some membranes allow water to diffuse through them freely, while salts in solution pass with difficulty. These are called semi-permeable membranes. If a solution of salt is separated from pure water by a membrane, water will pass through until the vapor pressure of the solution exceeds that of the vapor pressure of the water by a certain amount called the osmotic pressure of the solution. When air is forced through a porous plate of plaster or unglazed porcelain, the nitrogen penetrates the plate more rapidly than does the oxygen, and the air that filters through is richer in nitrogen than oxygen. The air infiltrating through plaster walls is therefore slightly low in oxygen content.

Rubber allows oxygen to escape more freely than nitrogen, so that gas in an automobile tire that has not been pumped up for a long time is chiefly nitrogen. These factors must be recognized in maintaining a room at a very low humidity; because if walls and roof are not protected by a waterproof membrane, moisture in form of vapor passes rapidly through plaster, concrete and insulation. Moisture carries through partition walls faster than the other constituents of atmospheric air because of the high velocity of water vapor molecules.

An air-conditioned space may be adjacent to other spaces which have in them a higher water vapor pressure than that in the conditioned space. Water vapor will flow through ordinary building materials in proportion to the vapor pressure difference. The total amount of water vapor transmitted is dependent on the permeability which is usually expressed in grains of moisture per square foot per hour per inch of mercury vapor pressure difference. The values in the following table are quoted from a publication of the National Bureau of Standards. The water vapor entering the conditioned space must be added to the latent cooling load. Waterproofing materials, acting as vapor barriers, must completely seal the walls, ceilings, and floors which are exposed to space having excessive vapor pressure, and all doors must have weatherstripping gaskets applied to them to complete the barrier effectively.

TABLE 15

PERMEABILITY OF VARIOUS BUILDING MATERIALS TO WATER VAPOR

Building Material	Permeability Grains per square foot per hour per inch Hg
Plaster base and plaster, ¾ in.	14.7
Fir sheathing, ¾ in.	2.9
Waterproof paper	49.1
Pine lap siding	4.9
Paint film	3.4
Sugar cane fiberboard ¾ in.	12.5
Brick masonry 4 in.	1.1
Foil-surfaced reflective insulation	0.08 to 0.13
Roll roofing, smooth; 40 to 65 lb roll 108 sq ft	0.13 to 0.17
Duplex or laminated papers 30–30–30	1.37 to 2.58
Plaster, wood lath	11
Plaster, 3 coats lead and oil	3.68 to 3.84
Plaster, 2 coats of Al paint	1.15
Plaster, fiberboard or gypsum lath	19.73 to 20.57
Plywood, ½ in. 5-ply Douglas Fir	2.67 to 2.74
Insulating lath and sheathing, board type	25.68 to 34.27
Insulating sheathing, surface coated	3.03 to 4.36
Insulating cork blocks, 1 in.	6.19
Mineral wool unprotected, 4 in.	29.07

26. Graham's Law

Gases when mixed, become uniformly distributed in a container and each gas fills the entire space and exerts its own pressure as if the other gases were not present, provided the gases do not react with each other. The rate at which a gas mixes with other gases or diffuses through a porous wall is determined largely by its density. At the same temperature and pressure, light gases diffuse faster than do heavy gases. If a cylinder containing hydrogen and oxygen is opened at one end of a room, hydrogen molecules will reach the opposite end of room in ¼ the time required for the oxygen molecules to travel the same distance. If sulfur dioxide gas and methane are forced under pressure into unglazed porcelain vessels, the methane gas will diffuse through the walls of its container twice as fast as the sulfur dioxide.

Graham's law states that *the rates of diffusion of gases are inversely proportional to the square roots of their densities.* The densities of gases (weights per liter or per cubic foot) are proportional to their molecular weights, because equal volumes of gases contain the same number of molecules. For example the weight of a liter of oxygen is 1.429 g and of hydrogen is 0.0898 g. The ratio is

$$\frac{0.0898}{1.429} = \frac{1}{16} \text{ (approximately)} \quad \text{and} \quad \frac{\sqrt{1}}{\sqrt{16}} = \frac{1}{4}$$

Hydrogen therefore diffuses four times as fast as oxygen. The molecular weight of hydrogen is 2 and of oxygen 32. The ratio is $\frac{2}{32} = \frac{1}{16}$ and the square roots of these relative weights are 1 and 4, the same result as above.

27. Dehydrating Agents

Dehydrating agents may be divided into two general classifications:

1. *Adsorbent.* A material which has the ability to condense water vapor on its surface without itself being changed physically or chemically. Certain solids such as silica gel, activated alumina and activated carbon possess this property.

2. *Absorbent.* A material which has the ability to take up water vapor but which changes physically, chemically or both, during the cycle. Calcium chloride is an example of a solid material while liquid materials include ethylene glycol and lithium chloride, calcium chloride and lithium bromide brines.

28. Adsorbents

These substances are characterized by a very porous physical structure which still retains enough mechanical strength to resist the wear and handling to which they are subjected. Adsorbed gases form a layer one molecule thick on the surface. The lower the temperature the more slowly the gas molecules move and the more easily they are held on the surface of the adsorbent. The gases which are most easily liquefied are most easily adsorbed. Colloids are highly adsorbent because their small particle size offers a large surface. Charcoal, which is porous, has a vast surface per unit weight and is used to adsorb gases. To be suitable for dehydration purposes such substances must fulfill the following requirements:

1. Possess suitable vapor pressure characteristics.
2. Be available at an economical cost.
3. Adsorb sufficient moisture per pound of adsorbent to avoid excessive bed dimensions.
4. Be chemically stable and resist contamination.
5. Be physically rugged to resist breakdown from handling.
6. Withstand breakdown from repeated reactivation cycles.
7. Possess practical and efficient reactivation temperatures.

Aluminum oxide in a porous amorphous form is a solid adsorbent commonly called activated alumina. A good grade contains 92% Al_2O_3. This substance has the property of adsorbing gases and vapors other than water vapor, a property which is useful in certain air-conditioning installations. It is available commercially in sizes ranging from fine powder to pieces 1.5 in. in diameter. It may be repeatedly re-activated after use without practical loss of its adsorptive qualities by heating at temperatures below 350° F. The pores occupy approximately 58% of the volume of each particle.

Silicon dioxide, in a special form obtained by mixing sulfuric acid with sodium silicate, is commonly called silica gel; this is another solid adsorbent. The gel is available in granules from 4 to 300 mesh. It has high adsorptive capacity and may be repeatedly reactivated at temperatures up to 600° F. The pores occupy 50 to 70% of the volume of each particle. Other substances used as adsorbents are charcoal and Lamisilate.

29. The Adsorption Process

The adsorbent does not go into solution but water vapor is extracted from the air-vapor stream passing through the bed of adsorbent material and is caught and retained in the capillary pores. The exact nature of the process which goes on during adsorption is not known, but it is stated that the action is brought about by surface condensation, and also by a difference

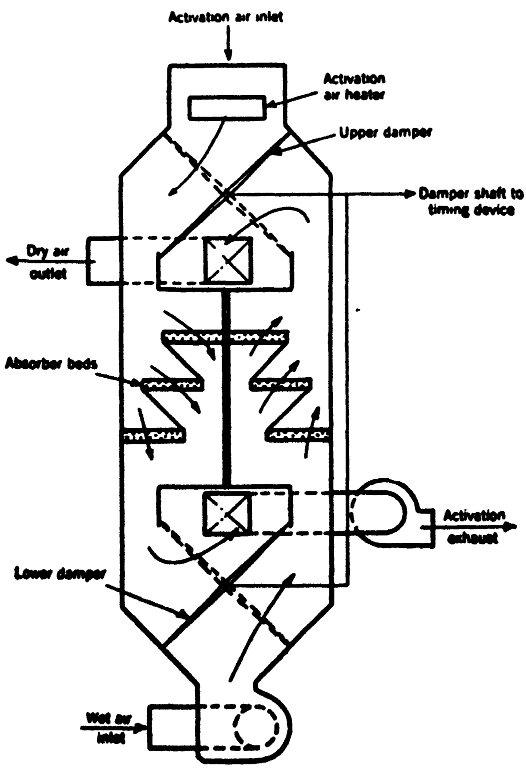

FIGURE 1

Solid Absorbent Dehydrator—Stationary Bed Type
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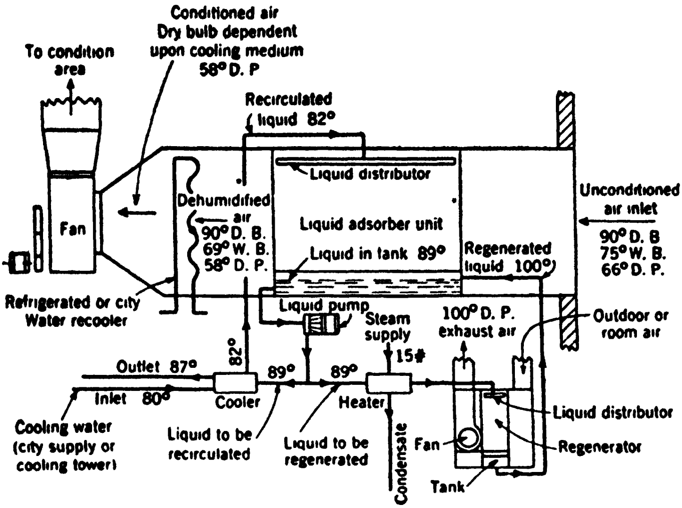

FIGURE 2

Diagram of Lithium Chloride Adsorption System
Reprinted from A.S.H.V.E. Guide.

between the vapor pressure of the water condensing inside the pores and the partial pressure of the water vapor in the air-vapor mixture. The adsorbing process in the bed can continue until the vapor pressures come into equilibrium. The amount of vapor adsorbed will depend on the adsorbent substances being used, but for any single substance the amount depends on the temperature of the bed as well as on the partial vapor pressure of the air-vapor mixture being passed over it.

As the bed of material adsorbs moisture, its vapor pressure approaches that of the contacting air and the rate of adsorption gradually slows down so that equilibrium may not be reached for 24 to 48 hours. Because of this diminishing rate of adsorption, commercially designed systems do not permit the state of equilibrium to be reached but generally operate on a 10 to 30 minute contact time—the period of most rapid adsorption.

As the process of adsorption goes on heat is liberated in the bed. The heat so liberated is the latent heat of the water vapor condensed together with the so-called heat of wetting. For a pound of water vapor at 60° F the latent heat released by condensation is approximately 1057 Btu. The heat of wetting for silica gel, for example, is about 200 Btu, making a total heat of adsorption of approximately 1257 Btu per pound of water adsorbed from the air-vapor mixture passing through the silica gel bed. The heat of wetting varies with the substance being used as the adsorbent while the latent heat of condensation depends only on the temperature and pressure of the water vapor.

30. Absorbents

Any absorbent substance may be used as an air-drying agent if it has a vapor pressure lower than the vapor pressure in the air-vapor mixture from which the moisture is to be removed.

Solid Absorbents. The substances used are in general the solid forms of the liquid absorbents, more commonly calcium chloride, due to its low cost. At present they are used principally in small desiccating chambers, and in small dryers of the cartridge type, through which air is forced under pressure.

Liquid Absorbents. These are characteristically water solutions of materials in which the vapor pressure is reduced to a suitable level by governing the concentration of the solution. In addition to having suitable vapor pressure characteristics, a practical absorbent must also be widely available at economical cost, be non-corrosive, odorless, non-toxic, non-inflammable, chemically inert to any impurities in the air stream, stable over the range of use and especially it must not precipitate out at the lowest temperature to which the apparatus is exposed. It must have low viscosity and be capable of being economically regenerated or concentrated after having been diluted by absorbing moisture.

Water solutions, or brines, of the chlorides or bromides of various inorganic elements such as lithium and calcium are the absorbents most frequently used in connection with air-conditioning applications and detailed attention is confined to these two in this chapter.

31. Nature of Absorption Process

The application consists of bringing the air-vapor stream into intimate contact with the absorbent, permissibly by passing the air stream through a finely divided spray of the brine but more generally by passing the air over a contacting pack where the liquid absorbent presents a large surface to the air stream. The difference in vapor pressure causes some of the vapor in the air-vapor mixture to migrate into the brine. Here it condenses into liquid water and decreases the concentration of the absorbent.

As the water vapor is added to the absorbent and condenses, it gives up its latent heat of condensation which tends to raise the temperature of both the absorbent and the moist air stream. For every pound of water absorbed and condensed the heat added to the air stream and the brine combined is obtainable from steam tables. For instance, at 60° F the amount of this heat is about 1057 Btu. In addition to this heat there is involved also the so-called heat of mixing which is frequently considerable.

A more complicated cycle involves heat removal from the contacting medium, either within or external to the interchanger. Thus the temperature of the medium may be higher than, equal to, or lower than that of the air, depending on the agent used and the function to be performed. In such a cycle, the dehydration process may be accompanied by cooling or heating, or neither, and such effect, if present, may be either a necessary by-product of the process, or for the specific purpose of obtaining both latent and sensible heat removal simultaneously. The heat thus produced in the bed is to a large extent transferred to the air being dried, and in the average air-conditioning installation must be removed by passing the air through an aftercooler.

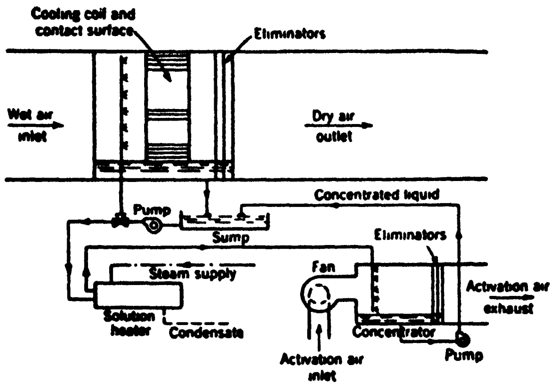

FIGURE 3

Liquid Absorbent Dehydrator in which Solution Cooler and Contactor are Combined

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32. Moisture in Refrigeration Systems

Moisture may be present in a system as the result of

1. Faulty drying of equipment at factory.
2. Introduction during assembly or installation.
3. Leaks in low side resulting in entrance of moist atmospheric air.
4. Leakage of water condenser.
5. Oxidation of certain hydrocarbons in oil, to produce moisture.
6. Moisture in refrigerant due to faulty dehydration in manufacture or handling.
7. Moisture in the lubricating oil.

It is necessary that moisture be eliminated or the following troubles may occur:

1. *Freezing up* at expansion valves, capillary tubes, or other orifices.
2. Corrosion of metals to form sludge.
3. Copper plating.

There are two methods used for drying refrigerating equipment at the shop:

1. Heat to evaporate moisture by placing parts in an oven which requires a certain period of time.
2. Dry air produced by adsorption processes, a blast being directed through the apparatus.

After a refrigerating system has been installed *driers* are incorporated if moisture trouble develops. A *drier* is a suitable container filled with adsorbing material. The *drier* is inserted in the liquid refrigerant piping system so that the refrigerant passes through it.

Chapter III

CHANGE OF STATE

Substances exist under ordinary conditions as solids, liquids and gases. However, by changing temperature, pressure, or both, most substances can be made to exist in any of the three states desired.

Water is a liquid under ordinary conditions. When the temperature is reduced sufficiently it freezes into ice, a solid; when the temperature is raised sufficiently it turns into steam, a gas.

This change of state is physical. A chemical change is a change in composition.

33. Liquefaction and Solidification

Substances in their natural state appear in either the solid, liquid or gaseous condition. Heat is an agent which causes change of state; it always acts in such a manner that a substance passes from the solid to the liquid, and from the liquid to the gaseous state by the addition of heat, and back in the reverse direction by the withdrawal of heat. This order is universal and is never reversed. If a substance is capable of assuming the "viscous state" it does so before it melts and it passes gradually from a solid state, through a semi-solid viscous state to that of a liquid of evident mobility. Sealing wax is an example of this nature. In like manner, iron before it melts loses its hardness and becomes so soft that pieces may be welded together. In many cases, however, the change from solid to liquid is very abrupt. As there is a marked difference between substances as to the abruptness with which they change state, there is also a class of substances which change their compositions in the act of changing state. Saline solutions are of this group. In many of these a greater quantity of salt is retained in solution at high temperatures than at low ones, so that when they are left to cool, crystals of salt are deposited.

TABLE 16

BOILING AND MELTING POINTS, ° F

Substance	*Boiling Point*	*Melting Point*	*Substance*	*Boiling Point*	*Melting Point*
Alcohol	172	—148	Manganese	3452	2237
Aluminum	3272	1217.6	Mercury	674.6	—37.96
Antimony	3272	1166	Molybdenum	6548	4596
Brass or Bronze	——	1650	Nickel	——	2646.1
Calcium	——	1481	Platinum	7070	3191
Cast Iron (gray)	——	2200	Porcelain	——	2820
Cast Iron (white)	——	2070	Potassium	1314	144
Chromium	——	2939	Silver	3551	1760.9
Copper	4190	1981.4	Sodium	1382	207
Glycerin	554	—4	Steel	——	2370–2550
Gold	——	1915.4	Sulfur	832	230
India Rubber	——	257	Tin	4118	449.6
Iron	4442	2786	Tungsten	10520	6152
Lead	2777	621	Water	212(ice)	32
			Zinc	1706	786.9

34. Latent Heat of Fusion

When heat is applied to ice, it is not instantly but gradually converted into water. The reason for this is that a large quantity of heat must first enter into the ice at 32° F before it becomes water at 32° F. This heat is called latent because it is absorbed by the ice without producing any rise in temperature, a rough formula of the process may represent it as:

$$\text{Water at } 32° \text{ F} \rightleftharpoons \text{ice at } 32° \text{ F} + \text{latent heat}$$

All substances in passing from the solid to the liquid state absorb heat which may be accurately measured. The latent heat of fusion is the same as the latent heat of liquefaction.

TABLE 17

LATENT HEAT OF FUSION

	Temperature ° F	Btu per pound
Ammonia	–107.8	194.5
Benzene	40	55.1
Bromine	45.1	29.2
Candy (hard or chocolate)	234–310	25
Cast iron (gray)	2460–2550	41.4
Cast iron (white)	1920–2010	59.4
Ice	32	144
Ice cream	27	90
Lead	608–620	9.66
Mercury	–38.2	5.06
Phosphorus	111.5	9.0
Paraffin	125	63.2
Phenol	77	44.9
Silver	1749–1800	37.93
Tin	451	50.63
Wax (bees)	143	76.1
Zinc	680	50.63

35. Freezing Mixtures

In solution, just as in fusion, a certain quantity of heat becomes latent; this is utilized to advantage in producing intense cold. If two solids or one solid and one liquid produce on mixing a substance which is not solid but liquid, this substance has a low fusion temperature. Freezing mixtures may be used for obtaining constant temperatures for test purposes as listed in Table 18.

TABLE 18

ENDOTHERMAL MIXTURES

Mixtures		Temperature drop °F
Ammonium nitrate Water	1 1	From +40° to +4°
Ammonium chloride Potassium nitrate Water	5 5 16	From +50° to +10°
Ammonium chloride Potassium nitrate Sodium sulfate Water	5 5 8 16	From +50° to +4°
Sodium nitrate Nitric acid, diluted	3 2	From +50° to −3°
Ammonium nitrate Sodium carbonate Water	1 1 1	From +50° to −7°
Sodium phosphate Nitric acid, diluted	9 4	From +50° to −12°
Sodium sulfate Sulfuric acid, diluted	5 4	From +50° to +3°
Sodium sulfate Ammonium chloride Potassium nitrate Nitric acid, diluted	6 4 2 4	From +50° to −10°
Sodium sulfate Ammonium nitrate Nitric acid, diluted	6 5 4	From +50° to −40°
Snow or pounded ice Sodium chloride	2 1	From +50° to −5°
Snow or pounded ice Sodium chloride Ammonium chloride	5 2 1	From +50° to −12°
Snow or pounded ice Sodium chloride Ammonium chloride Potassium nitrate	24 10 5 5	From +50° to −18°
Snow or pounded ice Sodium chloride Ammonium nitrate	12 5 5	From +50° to −25°

ENDOTHERMAL MIXTURES (*continued*)

Mixtures		Temperature drop °F
Snow Sulfuric acid, diluted	2 3	From +32° to −23°
Snow Hydrochloric acid	8 5	From +32° to −27°
Snow Nitric acid, diluted	7 4	From +32° to −30°
Snow Calcium chloride	4 5	From +32° to −40°
Snow Calcium chloride, crystallized	2 3	From +32° to −50°
Snow Potash	3 4	From +32° to −51°

36. A Natural Thermal Unit

The heat of fusion of ice, 79.75 calories per gram or 144 British thermal units (Btu) per pound, forms the most natural thermal unit because it is quite independent of the idea of temperature. A quantity of heat has no more connection with temperature than a quantity of water with a height above sea level. Indeed, the only reason this natural unit has not been adopted instead of the calorie (which does involve temperature) is the experimental difficulty of using it in the laboratory. However refrigeration engineers have adopted the melting effect of a ton of ice as a unit which is equivalent to one ton of ice melting in 24 hours or 200 Btu per minute.

37. Latent Heat of Vaporization

A large quantity of heat is required by substances in order to enable them to pass from the solid to the liquid state or from the liquid to the gaseous state. This heat which is absorbed does not as a rule manifest itself by producing an increase in temperature, and it has on this account been called latent heat. These processes may be represented roughly:

Water at 32° F = Ice at 32° F + latent heat of liquefaction
Steam at 212° F = water at 212° F + latent heat of vaporization

Latent heat was discovered by Black in 1762.

TABLE 19

APPROXIMATE HEAT ABSORBED IN LOWERING THE TEMPERATURE OF THE WATER CONTENT OF FOOD PRODUCTS FROM +60° F TO −5° F

Product	Percentage water, average	Heat absorbed, per pound of product, Btu
Fish Fillets		
Cod	83	157
Haddock	82	155
Halibut	75	142
Mackerel *	73	138
Salmon	65	123
Shellfish, Edible Portion		
Clams	86	163
Lobsters	79	149
Oysters	87	165
Meat, Edible Portion *	75	142
	70	132
	60	113
	50	95
	40	76
Poultry (Dressed)		
Chickens (Broilers)	44	83
Fowl	47	89
Geese	39	74
Turkeys	42	79
Vegetables, Edible Portion		
Asparagus (Blanched)	92	174

* Water content varies over wide range, principally in relation to fat content.

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TABLE 19 (*continued*)

Product	Percentage water, average	Heat absorbed, per pound of product, Btu
Beans, Lima	69	130
Beans, String	89	168
Carrots	88	166
Corn	75	142
Mushrooms	88	166
Peas	74	140
Rhubarb	94	178
Spinach (Blanched)	90	170
Squash	88	166
Berries and Fruit, Edible Portion		
Apples	85	161
Cherries	81	153
Cranberries	89	168
Peaches	89	168
Pineapples	89	168
Raspberries (Red)	86	163
Strawberries	90	170
Distilled Water	100	189

TABLE 20

LATENT HEAT OF VAPORIZATION AT ATMOSPHERIC PRESSURE (29.92 in. MERCURY)

Substance	*Heat of Vaporization Btu per pound*	*Substance*	*Heat of Vaporization Btu per pound*
Alcohol (ethyl)	369	Acetone	233
Alcohol (methyl)	480.6	Hydrogen	222
Benzol	169	Nitrogen	81.5
Ether	159.1	Oxygen	92
Mercury	122	Water	970.4

38. Sublimation

Ordinarily, when sufficient heat is applied to a substance it assumes the gaseous state unless it is of such a nature that it will decompose before assuming this state. This change of state is usually preceded by an intermediate change into a liquid, but sometimes the gaseous state is assumed without intermediate liquid formation. This phenomenon is called sublimation and the passage of a liquid to the gaseous state goes under the general name of vaporization. No matter how the gaseous condition is achieved, a considerable quantity of latent heat is always involved. Familiar examples of sublimation are the vaporization of ice below 32° F or that of iodine on the application of gentle heat. The drying of clothes hung on the line in freezing weather is perhaps the most familiar example.

39. Production of Vapor and its Condensation

The following rough formula applies to conversion of a liquid into a gas

Steam at 212° F = water at 212° F + latent heat of vaporization

The latent heat of vaporization is greater than that of fusion. The latent heat must be disposed of in a sensible form when the gas is condensed into a liquid, and thus the latent heat of vaporization is of great service in retarding the change from the liquid to the gaseous state or vice versa.

Gaseous substances have been divided into gases and vapors but the distinction is merely conventional. Vapor denotes the gaseous form of a substance which at ordinary temperatures exists as a liquid or solid, gas denotes a substance which exists under ordinary conditions in the gaseous form, and which can be reduced to the solid or liquid form only by intense pressure or intense cold or both.

40. Vaporization

Vaporization is the general name for a process which occurs in two forms, namely;

(a) Evaporation, the conversion of a liquid into a gas quietly and without the formation of bubbles

(b) Ebullition, where bubbles of vapor are formed in the mass of the liquid itself

41. Vapors in a Vacuum

Vapors are formed more readily in a vacuum than at atmospheric pressure. The presence of air or of any foreign gas retards the formation of vapors, but in a vacuum a liquid is converted very quickly into vapor. Dalton discovered the following law: In a space destitute of air the vaporization of a liquid continues until the vapor has attained a determinate pressure dependent on the temperature, so that in every space void of air which is saturated with vapor, determinate vapor pressure corresponds to determinate temperature.

Mixtures of gas and vapor in a confined space have already been described, but a restatement of Dalton's Law in his own words follows: "In a space filled with air the same amount of water evaporates as in a space destitute of air; and precisely the same relation subsists between the temperature and the pressure of the vapor whether the space contains air or not."

42. Low Temperatures Caused by Evaporation

Whenever a vapor is produced, a quantity of heat is rendered latent. This heat is necessary to the formation of vapor and must be supplied from some foreign source; or, if this is not available, from the very liquid which is being evaporated. In this case the temperature of the liquid falls in order to supply the heat necessary to form the vapor. Low temperatures may be produced by a rapid absorption of heat by certain substances during either a chemical or physical change of state. Chemical changes require, and are accompanied by, a heat transfer. Artificial refrigeration, however, is produced commercially by apparatus causing a physical change of state by mechanical process.

The following systems are in use at present:

1. Vapor compression machines using a volatile liquid such as ammonia, dichlorodifluoromethane, methyl chloride, etc.
2. Steam jet vacuum machines using water vapor as the medium.
3. Absorption machines using ammonia as the refrigerant.

The refrigerants generally used exist only as a gas at atmospheric pressure and ordinary temperatures but they are liquefied when cooled and compressed by a sufficiently high pressure. The heat absorbed in re-evaporating the liquid at a reduced pressure causes the refrigerating effect. To return the refrigerating medium periodically to its original liquid state, the system must comprise the following parts:

1. *Evaporating coils.* The liquid evaporates in the evaporating coils, absorbing the latent heat it requires from its surroundings and producing the refrigerating effect.
2. *The compressor.* Vapor from the evaporating coils is removed by suction from the compressor and discharged into the condenser at a terminal pressure corresponding to the temperature of saturated vapor obtainable with the cooling water available.
3. *The condenser.* The latent heat and heat of compression are removed in the condenser, and the vapor reliquefied by cooling. This is effected by circulating cold water or air through or over the condenser tubes or pipes.

43. Evaporation

Evaporation denotes the quiet production of vapor at the surface of a liquid and is governed by the following conditions:

1. It varies with the temperature.
2. It varies with the extent of surface exposed.

3. It is rapid in a vacuum; slow in a space filled with air.
4. It is more rapid in dry air than in air containing vapor.
5. It is assisted by any agitation tending to renew the particles of air over the evaporating surface.

44. Ebullition; the Boiling Point

It has been shown that liquids undergo evaporation at all temperatures. The region over the free surface of a liquid may contain so much vapor from the liquid that condensation will occur as rapidly as new vapor is formed. The phenomenon of boiling occurs when the temperature has reached a point where vapor is formed in the interior of the liquid and rises to the top. But if the pressure of this hot vapor were not equal to that at the free surface of the liquid, the bubble of vapor would collapse in the liquid.

The temperature of boiling (what is ordinarily called the boiling point) is simply that temperature at which the vapor of the liquid exerts a pressure equal to the atmospheric pressure at the free surface of the liquid.

45. Vaporization of Liquids in a Limited Space; Critical Temperature and Pressure

When a liquid is enclosed within a limited space, and heat is applied, each increment of heat produces a certain quantity of vapor which accumulates and, by its pressure, prevents the liquid from boiling, the temperature of the liquid meanwhile rising considerably above its ordinary boiling point.

Eventually, however, the critical point is reached. This is defined as follows:

Critical temperature is that temperature above which a gas cannot be liquefied by pressure alone. The pressure at which a substance exists as a gas in equilibrium with the liquid at the critical temperature is the critical pressure.

TABLE 21

PROPERTIES OF REFRIGERANTS

Name	Symbol	Nickname	Boiling point, °F	Freezing point, °F	Critical Point Pressure lb/in.²	Critical Point Temperature °F
Ammonia	NH_3	——	−28.0	−107.86	1651	271.4
Butane	C_4H_{10}	——	30.9	−211.0	529.0	308.0
Carbon dioxide	CO_2	——	−109.3	−69.9	1066.2	87.8
Carbon tetrachloride	CCl_4	——	170.2	−9.4	658.5	541.4
Dichlorodifluoromethane	CCl_2F_2	Freon–12	−21.7	−247	582.0	232.6
Dichloroethylene	$C_2H_2Cl_2$	Dieline	122.4	−70.0	795	470.0
Dimethyl ether	$(CH_3)_2O$	——	−12.8	−216.0	216	772.0
Ethane	C_2H_6	——	−127.5	−278.0	718.0	89.8
Ethylamine	$C_2H_5NH_2$	——	61.6	−115.0	——	——
Ethyl bromide	C_2H_5Br	——	100.4	−102.0	——	447.0
Ethyl chloride	C_2H_5Cl	——	54.4	−217.7	764.0	369.0
Ethylene	C_2H_4	——	−154.7	−272.0	749.0	49.5
Ethyl ether	$(C_2H_5)_2O$	——	94.2	−177.34	521.0	382.0
Hexane	C_6H_{16}	——	155.9	−138.0	434.0	460.5
Isobutane	C_4H_{10}	——	10.3	−229.0	522.0	272.7
Methane	CH_4	——	−258.9	−297.0	537.1	115.7
Methyl chloride	CH_3Cl	——	−11.0	−153.7	969.2	289.6
Methyl formate	$C_2H_4O_2$	——	89.3	−147.5	870.5	417.0

PROPERTIES OF REFRIGERANTS (*continued*)

Name	Symbol	Nickname	Boiling point, ° F	Freezing point, ° F	Critical Point Pressure lb/in.^{2}	Critical Point Temperature ° F
Methylamine	CH_3NH_2	——	18.4	—134.5	1082.0	314.0
Methylene chloride	CH_2Cl_2	Carrene	104.6	—142.0	670.0	421.0
Pentane	C_5H_{12}	——	97.1	—203	470.0	387.0
Propane	C_3H_8	——	—44.0	—309.8	661.5	204.1
Propylene	C_3H_6	——	—52.5	—301.0	645.0	196.0
Sulfur dioxide	SO_2	——	13.6	— 98.9	1141.5	314.8
Trichloroethylene	C_2HCl_3	Trieline	190.2	—124.3	728.0	520.0
Trichloromethane	$CHCl_3$	Chloroform	142.2	—82.3	807.0	500.0
Water	H_2O	——	212.0	32.0	——	——

TABLE 22

LOW TEMPERATURE GASES

	Boiling Points, ° C
Argon	—186.1
Fluorine	—187
Helium	—267
Hydrogen	—252.6
Krypton	—152.9
Neon	—239
Nitrogen	—195
Oxygen	—182.7
Ozone	—119
Xenon	—109.1
Air	—200

Oxygen

t °C		p mm
—210.4		——
—204.5		36.11
—195.5		162.15
—186.9		493.3
—182.9	bp	1 atm
—154.5*		atm
—149.3		12.51
—129.9		——
—125.3		38.57
—118.8	cp	49.71

Nitrogen

t °C		p mm
—209.6	tp	96.4
—198.3		561.3
—195.8	bp	1 atm
—173.6		7.37
—152.1		25.84
—147.1	cp	33.49

* Liquid, 9758 g/cm^3, vapor 0.0385 g/cm^3

These tables reprinted by courtesy of A.S.R E.

Chapter IV

HEAT AND HEAT TRANSFER

What is heat? Is it matter or is it energy? Among the ancients it was thought to be one of the elements. According to the philosophy of Aristotle everything in the world was composed of four fundamental or elementary materials; earth, air, fire and water. The Greeks conceived of heat as a substance and until one hundred years ago scientists held this view, calling it *caloric*. This was conceived to be a substance just as real as air or water. Heat is a form of energy which, like other forms, is invisible and imponderable. It must be studied from the effects which it produces on various materials.

46. Theory of Heat

The accepted explanation of the phenomenon of heat is that it is a form of energy which can be produced by the conversion of any other form of energy. It is transferred wherever a temperature difference exists.

47. Temperature and Temperature Measurement

Heat is measured in terms of temperature. Temperature may be defined as the condition of a body which determines the transfer of heat to or from other bodies. Two bodies are at the same temperature when there is no transfer of heat between them. If there is a transfer of heat between them, the one from which the heat flows is said to have the higher temperature. In mechanical engineering work, temperatures are usually measured on the Fahrenheit scale. The ordinary form of mercury thermometer is used for temperatures ranging from —40° F to 500° F.

For measuring temperatures below —40° F, thermometers filled with alcohol are used. For measuring temperatures from 500° F to 1500° F some form of pyrometer is generally used. Electrical pyrometers or thermocouples may be used up to 2500° F. For still higher temperatures optical pyrometers give satisfactory results.

48. Absolute Zero

The theory of heat requires some absolute standard of comparison for the scale of temperature, so that the absolute scale has been adopted. A perfect gas contracts $\frac{1}{491.6}$ of its volume at 32° F for each degree Fahrenheit it is reduced in temperature. The point of zero volume, −459.6° F, is called absolute zero and is manifestly an imaginary one, about 460° below the freezing point 32° on the Fahrenheit scale. See topic 4.

49. Unit of Heat

Because heat is not a substance it can only be measured by the effect which it produces. The British thermal unit (Btu) is generally used in engineering work. It is the heat required to raise the temperature of one pound of water at its maximum density through 1° F.

The kilogram calorie or large calorie (kg-cal, Cal) is the quantity of heat required to raise the temperature of one kilogram of water from 14.5° C to 15.5° C, or $\frac{1}{100}$ of the heat required to raise one kilogram of water from 0° C to 100° C.

The gram calorie, or small calorie (g-cal, cal) generally used in scientific work, is the quantity of heat required to raise one gram of water from 3.5° to 4.5° C.

The mechanical equivalent of heat as determined by U. S. Bureau of Standards is the number of foot pounds equivalent to one Btu, i.e., *778.57* (*778.6*) ft lb.

50. Specific Heat

The quantity of heat which a substance gives up depends, in general, upon three factors: mass, difference of temperature, and a specific capacity for absorbing heat. This is a constant called *specific heat.* The ratio between the amount of heat required to raise the temperature of one pound of a substance 1° F and that required to raise the temperature of one pound of water 1° F is known as the specific heat of the substance.

In solid and liquid substances it is necessary to consider but one specific heat, because the change in volume in a heated solid or liquid is so small that its effect may be neglected. In gases the change in volume on heating is great, and if heat is applied under a constant pressure this change is directly proportional to the change in the absolute temperature. If there is a change in volume there must be external work done. On the other hand, when a gas is confined and is heated, it cannot expand. If it does not expand, there is no external work done. Therefore, two conditions arise when considering the specific heat of a gas, viz.,

(a) *Pressure constant and volume increasing when gas is heated.*

(b) *Volume constant and pressure increasing when gas is heated.*

Therefore, in the case of a gas, there are two specific heats, the *specific heat of constant pressure* and the *specific heat at constant volume.* The specific heat of constant volume will be denoted by C_v and the specific heat of constant pressure by C_p, both being expressed in Btu. When expressed in foot-pounds they will be denoted by K_v and K_p respectively.

TABLE 23

SPECIFIC HEAT OF GASES AND VAPORS

Substance	Temperature ° F	Specific Heat at Constant Pressure	Ratio of Specific Heats C_p/C_v	Specific Heat at Constant Volume (Computed)
Air	32–392	0.2375	1.405	0.169
Ammonia	80–392	0.5356	1.277	0.419
Carbon Dioxide	52–417	0.2169	1.3003	0.1668

TABLE 23 (*continued*)

Substance	Temperature °F	Specific Heat at Constant Pressure	Ratio of Specific Heats C_p/C_v	Specific Heat at Constant Volume (Computed)
Carbon Monoxide	79–388	0.2426	1.395	0.1736
Coal Gas	68–1900	0.3145	——	——
Flue Gas	——	0.24 (Approx.)	——	——
Hydrogen	70–212	3.41	1.419	2.402
Nitrogen	32–392	0.2438	1.41	0.1729
Oxygen	55–404	0.2175	1.3977	0.155
Water Vapor	212	0.421	1.305	0.322
Water Vapor	356	0.51	——	——

Courtesy of A.S.H.V.E.

TABLE 24

SPECIFIC HEAT OF VARIOUS SOLIDS

	Weight, Pounds Per Cubic Foot	Specific Heat
Asbestos	—	0.20
Ashes	43.0	0.20
Bakelite, laminated	86.0	0.35
Bakelite, wood filler	85.0	0.33
Bakelite, asbestos filler	118.0	0.38
Benzol	55.0	0.42
Borax	—	0.24
Bronze, phosphor	554	0.09
Calcium carbonate	177	0.18
Calcium sulfate	185	0.27
Carborundum	195	0.16
Cellulose	94	0.37
Celluloid	90	0.36
Chalk	142	0.21
Coke	75	0.20
Concrete, stone	147	0.19
Concrete, cinder	105	0.18
Cork	15	0.48
Corundum	247	0.20
Cotton, baled	93	0.32

TABLE 24 (*continued*)

	Weight, Pounds Per Cubic Foot	Specific Heat
Cotton, loose	30	0.32
Cottonseed oil	57.8	0.47
Ebonite	72	0.35
Fats	58	0.46
Glass, common	164	0.199
Glass, crystal	184	0.117
Glass, flint	188	0.117
Glass, plate	161	0.161
Graphite	126	0.201
Gypsum, loose	70	0.26
Gypsum, compressed	152	0.26
Ice, at 14° F	57.5	0.530
Ice, at —4° F	—	0.480
Ice, at —76° F	—	0.392
Kaolin	160	0.224
Kerosene	50	0.476
Litharge	—	0.21
Magnetite	310	0.176
Mica	—	0.10
Paper	58	0.324
Paraffin 4° to 40° F	—	0.377
32° to 68° F	—	0.694
Plaster Paris	103	1.14
Porcelain	250	0 26
Pyrites	300	0.129
Rubber	59	0.48
Saltpeter	67	0.90
Sandstone	145	0.22
Serpentine	165	0.26
Soapstone	170	0.21
Sodium carbonate	91	0.27
Sodium nitrate	141	0.28
Sodium sulfate	167	0.21
Sugar	100	0.22
Sulfur	126	0.17
Tar	75	0.35

TABLE 25

SPECIFIC HEAT OF LIQUIDS

Acetic acid	0.51
Alcohol (absolute)	0.58
Aniline	0.49
Benzol	0.40
Chloroform	0.23
Ether	0.54
Fusel oil	0.56
Gasoline	0.50
Glycerin	0.58
Hydrochloric acid	0.60
Kerosene	0.50
Naphthalene	0.31
Machine oil	0.40
Mercury	0.033
Olive oil	0.40
Paraffin oil	0.52
Petroleum	0.50
Sulfuric acid	0.336
Sea water	0.94
Toluene	0.40
Turpentine	0.42
Molten metals:	
Bismuth (535–725° F)	0.036
Lead (590–680° F)	0.041
Sulfur (246–297° F)	0.235
Tin (460–660° F)	0.058

These tables reprinted from A.S.H.V.E. Guide.

TABLE 26

SPECIFIC AND LATENT HEAT OF VARIOUS FOOD PRODUCTS TEMPERATURE, ° F AND HUMIDITY

Article	Composition Water	Composition Solid	Specific Heat Above Freezing	Specific Heat Below Freezing	Latent Heat	Storage Temperature, °F	Relative Humidity
Fruit							
Apples	63.5	36.5	0.708	0.39	91.5	32–36	82
Banana (Edible Portions)	75.5	24.5	0.804	0.426	108.5	34	78
Berries (Fresh)	86.5	13.5	0.892	0.46	124.5	36	74
Cranberries	89.0	11.0	0.912	0.467	128.0	33–36	76

TABLE 26 (*continued*)

Article	Composition Water	Composition Solid	Specific Heat Above Freezing	Specific Heat Below Freezing	Latent Heat	Storage Temperature, °F	Relative Humidity
Fruit							
Cantaloupes (Whole)	45.0	55.0	0.560	0.335	65.0	40	80
Dates, Figs, etc.	79.0	21.0	0.832	0.437	104.0	50–55	74
Fruits, Dried	15–30	70–85	0.29–0.47	0.215–0.320	21.6–43.2	35–40	58
Grapes (Whole)	58.0	42.0	0.664	0.374	83.5	34–36	80
Lemons (Whole)	62.5	37.5	0.700	0.387	90.0	33–45	85
Oranges (Whole)	63.5	36.5	0.708	0.390	91.5	34–45	82
Peaches (Whole)	73.5	26.5	0.788	0.420	106.0	34–36	80
Pears, Watermelons	76.0	24.0	0.808	0.428	109.0	34–36	82
Meats							
Brined	52.0	48.0	0.616	0.356	75.0	38	75–80
Beef (Fresh)	68.0	32.0	0.744	0.404	98.0	33	85
Beef (Fat)	51.0	49.0	0.608	0.353	73.5	30	82
Beef (Lean)	72.0	28.0	0.776	0.416	102.0	30	85
Beef (Dried)	5–15	95–85	0.22–0.34	0.195–0.265	7.2–21.6	36–40	70
Hams, Ribs (Not Brined)	60.0	40.0	0.680	0.380	86.5	20	82
Shoulders (Not Brined)	76.0	24.0	0.808	0.428	109.0	20	82
Livers	65.5	34.5	0.724	0.396	93.5	20–30	82
Lamb	58.0	42.0	0.664	0.374	83.5	32	85
Pork (Edible Portion)	60.0	40.0	0.680	0.380	86.5	29–32	82
Pork (Fat)	39.0	61.0	0.512	0.317	56.2	30–35	80
Tenderloins, Butts, etc.	67.0	33.0	0.736	0.401	96.5	33	—
Veal	63.0	37.0	0.704	0.390	91.0	32–33	84
Fish							
Fresh Fish	70.0	30.0	0.760	0.410	101.0	20–28	80–85
Dried Fish	45.0	55.0	0.560	0.335	65.0	36	70–75
Oysters in Shell	80.38	19.62	0.843	0.441	115.7	30–40	—
Oysters in Tubs	87.0	13.0	0.896	0.461	125.1	25–35	—
Canned Goods							
Fruits			Same As Fresh			35–40	
Meats			Same As Fresh			35–40	
Sardines	70.0	30.0	0.760	0.410	101.0	35–40	
Butter, Eggs, etc.							
Butter	12.75	87.25	0.302	0.238	18.4	18–20	78
Cheese	35.0	65.0	0.480	0.305	50.5	34	80
Eggs	70.0	30.0	0.760	0.410	100.0	31	80–83
Milk	87.5	12.5	0.900	0.462	124.0	35	85
Vegetables							
Asparagus	94.0	6.0	0.952	0.482	134.0	34–35	82
Cabbage	91.0	9.0	0.928	0.473	131.0	34–35	78
Carrots	83.0	17.0	0.864	0.449	119.5	34–35	80
Celery (Edible Portion)	94.0	6.0	0.952	0.482	135.0	34–35	80
Onions	87.5	12.5	0.900	0.462	126.0	36	80
Parsnips	83.0	17.0	0.864	0.449	119.5	34–35	80
Potatoes	74.0	26.0	0.792	0.422	106.5	36–40	80
Sauerkraut	89.0	11.0	0.912	0.467	128.0	35	85
Miscellaneous							
Poultry Dressed Iced	73.7	26.3	0.790	0.421	105.0	28–30	—
Poultry Dry Picked	65.0	35.0	0.720	0.395	93.5	26–28	75–80
Poultry Scalded	75.0	25.0	0.800	0.425	108.6	20	—
Game Frozen	60.0	40.0	0.680	0.380	86.5	15–28	55
Poultry Frozen	60.0	40.0	0.680	0.380	86.5	15–28	55
Nuts (Dried)	3–10	97–90	0.21–0.294	0.195–0.244	4.3–14.4	35–40	68–70

Courtesy of Am. Soc. Pract. R. E.

TABLE 27

MEAN SPECIFIC HEAT OF DRY AIR AT STANDARD PRESSURE (29.92 IN. HG)

Temperature °F t	Specific Heat C_p	Temperature °F t	Specific Heat C_p	Temperature °F t	Specific Heat C_p
−96	0.2399	32	0.2400	160	0.2403
−64	0.2399	64	0.2400	192	0.2405
−32	0.2399	96	0.2401	224	0.2406
0	0.2400	128	0.2402	256	0.2408

Courtesy of A.S.H.V.E.

The values indicate that an average figure of 0.24 Btu per pound can be used in ordinary air-conditioning calculations between 32° F and 96° F.

TABLE 28

SPECIFIC HEAT OF WATER VAPOR AT CONSTANT PRESSURE

Temperature °F	Specific Heat at Constant Pressure	Mean Ratio C_p/C_v
32	0.4655	1.274
212	0.421	1.33
356	0.51	1.305

For ordinary calculations the specific heat of water vapor is assumed to be 0.444 for the ranges of temperature common in air-conditioning practice.

51. Specific Heat of Superheated Steam

Superheated steam is steam at a temperature higher than the temperature of the boiling point corresponding to the pressure. It is sometimes called steam gas. If water were mixed with superheated steam this water would be evaporated as long as the steam remained superheated. Superheated steam may have any temperature higher than that of the boiling point up to the critical conditions of 3206.2 lb per square inch absolute and 1200° F superheat temperature (saturation temperature 705.34° F). When steam is raised to any considerable temperature above the boiling point it follows very closely the perfect gas laws. The equation for superheated steam, considered as a perfect gas, is approximately

$$pv = 85.51\ T$$

The ratio of specific heat at constant pressure C_p to specific heat at constant volume C_v approximates 1.3. The specific heat of superheated steam for ordinary calculations may be assumed to have a value of 0.47 up to about 800° F.

TABLE 29

SPECIFIC HEAT OF WATER VAPOR AT CONSTANT PRESSURE

Temperature, °F	Pressure, pounds per square inch 1000	2000	4000	6000
200	1.0	1.0	0.99	0.98
400	1.07	1.06	1.05	1.04
600		1.45	1.77	1.21
800				1.45

TABLE 30

DENSITY AND SPECIFIC HEAT OF WATER AT ATMOSPHERIC PRESSURE

Temperature °C	°F	Density kilograms per liter	Specific heat
0	32	0.99987	1.00874
3.98	39.16	1.0000 (max)	——
10	50	0.99973	1.00184
20	68	0.99823	0.99859
30	86	0.99567	0.99745
40	104	0.99224	0.99761
50	122	0.98807	——
60	140	0.98325	——

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52. Heat Transmission

When the laws which regulate the distribution of heat are discussed, the subject falls into two distinct categories:

(a) Heat transfer by contact of one body with another
(b) Heat transfer by radiation through space

Category (a) may be conveniently divided into two classifications:

1. Conduction
2. Convection

The transfer of heat from one place to another by these methods are defined as follows:

By conduction: The transfer of heat from one part of a body to another part of the same body, or from one body to another in physical contact with it, without displacement of the particles of the body, is defined as conduction.

By convection: Convection is the transmission of heat from one part of a gas or liquid to another by the movement of the hotter portion of the fluid toward the colder.

By radiation: Transmission of energy by means of electromagnetic waves of very long wave-length. Radiant energy of any wave-length, may, when absorbed, become thermal energy and result in an increase in the temperature of the absorbing body. If two bodies, one hotter than the other, are placed within an enclosure, there is a continual exchange of energy between them. The hotter body emits more energy than it absorbs, the colder body absorbs more than it radiates. Even after equilibrium of temperature is established the process continues, each body radiating and absorbing energy.

53. Conduction

The fundamental unit in solving conduction problems is the thermal conductivity k, which is a specific property of any substance. Conductivity is defined as the time rate of heat flow through unit area of a homogeneous substance under the influence of a unit temperature gradient.

Let H_t = heat flow in Btu per hour
A = area in square feet
L = thickness in inches
t_1 = temperature, ° F at hot end of path
t_2 = temperature, ° F at cold end of path
k = Btu per hour per square foot per degree difference per inch thickness

Then
$$H_t = \frac{kA(t_1 - t_2)}{L}$$

Factors for obtaining values of k in various other common units are given in Table 31. In the case of radial flow through a thick-walled cylinder, A should be taken as the logarithmic mean of the outer and inner surfaces.

It is sometimes convenient to obtain a heat transfer coefficient of conductance (C) in Btu per hour per square foot per ° F difference for the thickness of a solid material under consideration, such as the metal wall of standard thickness pipe. This is done by dividing the conductivity by the thickness,

$$C = \frac{k}{L}$$
$$H_t = CA(t_1 - t_2)$$

TABLE 31

CONVERSION FACTORS FOR CONDUCTIVITY

Unit of Heat Flow	Unit of Area	Unit of Temperature Difference	Unit of Thickness	Divide k by
Btu per hour	square feet	°F	inches	1
Btu per hour	square feet	°F	feet	12
calories per second	square centimeters	°C	centimeters	2903
calories per hour	square meters	°C	meters	8.064
watt	square centimeters	°C	centimeters	694
watt	square inches	°C	inches	273
kilowatt	square feet	°F	inches	3415

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TABLE 32

THERMAL CONDUCTIVITY, *k*, OF VARIOUS SOLIDS

Btu per hour per square foot per inch thickness per ° F

Substance	*k*
Concrete, cinder	2.4
Concrete, stone	8.0
Fiber, sheet	2.0
Glass, window	5.5
Glass, Pyrex	7.5
Ice at 32° F	15.0
Paper	1.0
Porcelain	10.0
Rubber, solid	1.2
Sand, dry	2.3
Sand, moist	8.0
Sawdust	0.35
Snow, fresh	0.75
Snow, old	3.5
Soil, dry	1.0
Soil, moist	10.0

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54. Conductivity of Gases

For all gases *k* increases with temperature, but is practically independent of pressure except at very low or very high pressures. If the values of *k* are used to compute heat flow across a gas layer, it is to be noted that a considerable quantity of heat may be transferred by convection and radiation as parallel processes.

TABLE 33

THERMAL CONDUCTIVITY *k* OF GASES

(In Btu per hour per square foot per inch thickness per ° F difference in temperature)

Gas or vapor	Temperature ° F	*k*	k_{212}/k_{32}
Air	32	0.163	1.29
	212	0.211	

TABLE 33 (*continued*)

Gas or vapor	Temperature ° F	k	k_{212}/k_{32}
Ammonia	32	0.141	1.46
	212	0.206	
Carbon dioxide	32	0.097	1.37
	212	0.133	
Chlorine	32	0.053	
Ethane	32	0.126	1.78
	212	0.224	
Ethyl chloride	32	0.060	1.73
	212	0.104	
Hydrogen	32	1.13	1.28
	212	1.44	
Methane	32	0.21	
Methyl chloride	32	0.058	1.76
	212	0.102	
Nitrogen	32	0.163	1.29
	212	0.210	
Oxygen	32	0.165	1.30
	212	0.215	
Steam	212	0.162	
	572	0.254	
Sulfur dioxide	32	0.056	

Reprinted from A.S.R.E. Data Book.

55. Conductivity of Liquids and Solutions

The conductivity of most organic liquids decreases slightly with increasing temperature. For water k increases to a maximum at 270° F and then decreases until its value at 570° F is the same as at 40° F. The conductivity of aqueous solutions generally is lower than that of water; this reduction is roughly proportional to the concentration of the solute.

TABLE 34

THERMAL CONDUCTIVITY, k, OF LIQUIDS

Btu per hour per square foot per inch thickness per ° F

Liquid	Temperature ° F	k
Acetic acid	68	1.20

TABLE 34 (*continued*)

Liquid	Temperature ° F	k
Acetone	68	1.24
Alcohol (methyl)	68	1.44
Alcohol (ethyl)	68	1.26
Aniline	32	1.25
Benzene	68	1.18
Carbon bisulfide	68	1.11
Carbon tetrachloride	68	0.76
Carbon dioxide	68	1.45
Glycerin	68	1.98
Kerosene	68	1.05
Oil, lubricating	68	1.16
Oil, petroleum	55	1.03
Sulfur dioxide	68	2.34
Toluene	68	1.05
Water	32	3.85
	90	4.28
	140	4.54
	270	4.76
	420	4.55

TABLE 35

THERMAL CONDUCTIVITY, k, OF BRINES RELATIVE TO THAT OF WATER, k_w, AT 90° F

Brine	Spec. gravity	k/k_w
NaCl	1.09	0.97
	1.19	0.92
$CaCl_2$	1.13	0.95
	1.23	0.91
$MgCl_2$	1.10	0.95
	1.13	0.92
	1.20	0.89
	1.27	0.85

These tables reprinted from A.S.R.E. Data Book.

56. Conductivity of Metals and Alloys

According to the Weidemann-Franz-Lorenz Law (See *J. Inst. Metals,* **39, 337**) the thermal conductivity k in Btu per hour per square foot per ° F per inch of any metal or alloy can be obtained from the relation

$$Kw = 58\ T$$

Where w = electrical resistivity, in ohms per circular-mil-foot, and

T = temperature, ° R.

Except at very low temperatures, this relation has been found to hold fairly well. For most alloys, the value of k increases,

TABLE 36

THERMAL CONDUCTIVITY, k, OF METALS AND ALLOYS

Btu per hour per square foot per inch thickness per ° F

Metal	Temperature ° F	k
Aluminum	32	1475
	400	1560
Brass, yellow	68	770
	400	1010
Brass, red	32	715
	212	836
Copper	32	2680
	400	2605
Iron, pure	64	468
	212	438
Iron, cast	32	350
Iron, cast	400	220
Iron, wrought	32	423
Iron, wrought	400	360
Steel (1% C)	32	310
	400	300
Lead	64	241
	212	235
Monel Metal	90	242
Nickel	32	408
	400	390
Tin	32	450
	212	425
Zinc	32	775
	212	750

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but for the ferrous metals it decreases with rising temperatures.

In general the thermal conductivity of non-metallic materials increases considerably with temperature, density and moisture content.

57. Convection

The convection coefficient f, sometimes called the film coefficient, is expressed in Btu per hour per square foot of surface per degree difference in temperature between the surface and the main body of the fluid. In free convection, the value of the coefficient f is affected very little by the nature of the surface, but depends principally upon the size, temperature difference and physical properties of the fluid. There are two cases to be considered:

(a) Free convection in air
(b) Free convection in liquids

Case (a) For vertical plane surfaces in air

$$f = 0.30 \left(\frac{\theta}{L}\right)^{1/4} = \text{film coefficient}$$

where

θ = temperature difference
L = height in feet
f = Btu per hour per square foot per °F

The convection coefficient from a horizontal surface facing upward is about 30% greater, and facing downward is 30% less, than in the vertical position.

The convection coefficient for horizontal or vertical pipes in air is given as

$$f = 0.42 \left(\frac{\theta}{d}\right)^{1/4}$$

where d = pipe diameter in inches.

Case (b) Free convection in liquids (water)

In this case the coefficient f represents the total heat transfer, since radiation is not involved. For bodies of ordinary size in water

$$f = 0.165 \ (t_w + 30)\sqrt{\theta}$$

where t_w = water temperature in ° F.

This formula does not apply when the temperature is above 212° F. Values of f for various conditions are given in Table 37.

TABLE 37

HEAT TRANSFER COEFFICIENTS f FOR SURFACES IN STILL WATER

Water Temperature ° F	Temperature Difference, θ, ° F: 10	30	50	70	90
	Btu per hour per square foot per ° F difference				
40	37	63	82	96	110
60	47	81	105	124	141
80	57	100	128	152	172
100	68	117	152	179	203
120	78	136	175	207	235
140	89	154	198	235	—
160	99	172	222	—	—
180	110	190	—	—	—

Courtesy of A.S.H.V.E.

58. Free Convection in Liquids (Oil)

With free convection, if z = viscosity in centipoises, at the average of the surface and oil temperatures

$$f = 17 \frac{\theta^{1/4}}{z^{0.4}}$$

59. Forced Convection

If the fluid is forced over the heat transfer surface the value of the coefficient f is governed by the velocity and physical properties of the fluid and by the size, shape, positioning and nature of the surface. In general, roughening the surface, or anything that promotes turbulence in the fluid flow will increase the heat transfer coefficient. In liquids, the convection coefficient represents the total heat exchange. This is generally true of vapors as regards heat exchange between the liquid surface of radiation may be significant, especially at low velocities and high temperatures. The effect of humidity is negligible except when the temperature of the surface is below the dew point of the vapor. Condensation, which increases the rate of transfer above the freezing point, will then occur.

60. Heat Transfer to Boiling Liquids

Heat transfer coefficients from metal surfaces to boiling liquids are subject to extremely wide variations, depending chiefly on the nature of the surface, temperature difference and nature and temperature of the liquid. The value of f is affected so greatly by such factors as the amount of air dissolved in the liquid, the design of the apparatus, and the presence of scale on the surfaces, that it is not possible to obtain exact formulas. In boilers and evaporators f usually ranges from 400 to 2000 for water, and, in the case of refrigerants and organic liquids, may vary from 50 to 500, the higher values being associated with high heating rates.

The presence of even a small amount of a foreign gas, such as air, in a vapor may have a marked effect in reducing the heat transfer effect in steam and refrigeration condensers.

61. States of Thermal Equilibrium

Any homogeneous substance, if allowed to stand, reaches a state of thermal equilibrium. If a portion of the substance, or of the wall of its confining vessel, is heated (or cooled) slowly, the substance will be in a state of thermal equilibrium at any instant.

If the temperature change is rapid there is no longer a state of equilibrium. The temperature of the substance does not remain the same throughout. Such a condition obtains in the case of water falling through a cooling tower or liquid flowing in a double pipe condenser or brine cooler. The quantity of heat per unit area of surface varies. Part of the heat is used up in changing the temperature of the successive layers and to transfer a certain quantity of heat, the mean temperature difference assumed is nearer to the geometric mean than the arithmetic mean and is called the logarithmic mean. This is described in the following topic.

62. Over-all Heat Transmission Coefficient U

When heat flows from a medium of constant temperature t_1 to another medium at a constant temperature t_2 through a path having a constant mean area A square feet the rate of heat transfer in Btu per hour is

$$q = UA\theta$$

where θ is the temperature difference $(t_a - t_e)$. The value of the over-all coefficient U may be computed from the individual coefficients K, C or f, where U is the coefficient from medium to medium, through surface and scale.

In heat exchangers, where the temperature difference decreases from θ_a at one end of the apparatus to θ_e at the other end, an arithmetical mean $\frac{\theta_a + \theta_e}{2}$ may be used with little error, if $\frac{\theta_a}{\theta_e}$ is less than 2. Otherwise the logarithmic mean difference should be used.

Dr. Grashof in his work *Theoretische Maschinenlehre* introduced this formula for mean temperature difference:

$$\theta_m = \frac{\theta_a - \theta_e}{\ln \frac{\theta_a}{\theta_e}}$$

63. Calculation of True Mean Temperature Difference

The best example is that of a double-pipe brine and oil cooler.

Let T_a = initial temperature of warmer substance
t_a = initial temperature of cooler substance
T_e = final temperature of warmer substance
t_e = final temperature of cooler substance

Counter-current.

$T_a - t_e$ = temperature difference at one end
$T_e - t_a$ = temperature difference at other end

θ_a = greatest difference at either end
θ_e = smallest difference at either end

Formula

Log mean temperature difference, $\theta_m = \dfrac{\theta_a - \theta_e}{\ln \dfrac{\theta_a}{\theta_e}}$

This formula may be applied to either counter-current, parallel current or transverse flow.

Parallel Flow

$T_a - t_a$ = temperature difference at one end
$T_e - t_e$ = temperature difference at other end
θ_a = greatest temperature difference
θ_e = smallest temperature difference

Transverse Flow

Take the arithmetic mean of θ_m for counter-current and θ_m for parallel flow.

64. Rapid Calculation of Logarithmic Mean Temperature Difference

Hausbrand devised a table for rapid calculations, (See Table 38) but, before presenting it, an explanation will be given containing more symbols than the original:

MTD = θ_m = logarithmic mean temperature difference
$t_e - t_a$ = TR = temperature rise of cold substance
$T_a - T_e$ = TF = temperature fall of hot substance
$T_a - t_a$ = ITD = initial temperature difference between hot and cold substances
$T_a - t_a$ = FTD = final temperature difference between hot and cold substances
θ_a = GTD = greatest temperature difference at one end
θ_e = LTD = least temperature difference at other end

Case (a). Heated substance gives up heat with temperature remaining constant to a cold substance absorbing heat with rising temperature.

$$GTD = ITD$$
$$LTD = ITD - TR$$

This case includes: refrigeration condensers, steam condensers and boiler feed water heaters.

Case (b). Hot substance giving up heat with falling temperature to a cold substance absorbing heat with temperature remaining constant.

$$GTD = ITD$$
$$LTD = ITD - TF$$

This case includes: steam boilers, refrigeration brine coolers, refrigeration direct expansion coils in cold storage rooms, evaporator with hot liquid coils or jacket.

Case (c). Two substances, both changing temperature, one giving up heat with falling temperature to the other absorbing heat with rising temperature, either parallel or counter-current flow.

$$GTD = ITD$$
$$LTD = FTD$$

This case includes: Steam superheaters, economizers, brine coils in cold storage rooms, compressor intercoolers, cylinder jackets and oil coolers and water cooling coils in air conditioners.

$$MTD = \frac{GTD - LTD}{\ln \frac{GTD}{LTD}}$$

$$\text{Let } LTD = \text{percentage } GTD = \frac{P}{100} \times GTD$$

$$MTD = \frac{GTD\left(1 - \frac{P}{100}\right)}{\ln \frac{100}{P}}$$

By arranging the ratio $\frac{LTD}{GTD}$ as in Table 38 and tabulating the corresponding value of MTD when GTD equals unity, Hausbrand revised the means of obtaining a factor, by multiplying by actual GTD, which is always the greatest temperature difference, to obtain the actual logarithmic mean temperature difference which eliminates the necessity of referring to logarithm tables. *ln* or $\log_e$ is the Naperian or natural logarithm.

$$\log_e = 2.3025 \times \log_{10}$$
$$\log_{10} = 0.434 \times \log_e$$

This subject is easily visualized by referring to Figure 4 and Table 38.

TABLE 38

LOGARITHMIC MEAN TEMPERATURE DIFFERENCE FACTORS

LTD/GTD Col. A	MTD/GTD Col. B	LTD/GTD Col. A	MTD/GTD Col. B	LTD/GTD Col. A	MTD/GTD Col. B	LTD/GTD Col. A	MTD/GTD Col. B
0.0025	0.166	0.24	0.533	0.49	0.715	0.74	0.864
0.005	0.188	0.25	0.541	0.50	0.721	0.75	0.870
0.01	0.215	0.26	0.549	0.51	0.728	0.76	0.874
0.02	0.251	0.27	0.558	0.52	0.734	0.77	0.879
0.03	0.277	0.28	0.566	0.53	0.740	0.78	0.886
0.04	0.298	0.29	0.574	0.54	0.746	0.79	0.890
0.05	0.317	0.30	0.582	0.55	0.753	0.80	0.896
0.06	0.334	0.31	0.589	0.56	0.759	0.81	0.902
0.07	0.350	0.32	0.597	0.57	0.765	0.82	0.907
0.08	0.364	0.33	0.604	0.58	0.771	0.83	0.913
0.09	0.378	0.34	0.612	0.59	0.777	0.84	0.918
0.10	0.391	0.35	0.619	0.60	0.783	0.85	0.923
0.11	0.403	0.36	0.626	0.61	0.789	0.86	0.928
0.12	0.415	0.37	0.634	0.62	0.795	0.87	0.934
0.13	0.427	0.38	0.641	0.63	0.801	0.88	0.939
0.14	0.438	0.39	0.648	0.64	0.806	0.89	0.944
0.15	0.448	0.40	0.655	0.65	0.813	0.90	0.949
0.16	0.458	0.41	0.662	0.66	0.818	0.91	0.955
0.17	0.469	0.42	0.669	0.67	0.823	0.92	0.959
0.18	0.478	0.43	0.675	0.68	0.829	0.93	0.964
0.19	0.488	0.44	0.682	0.69	0.836	0.94	0.970
0.20	0.497	0.45	0.689	0.70	0.840	0.95	0.975
0.21	0.506	0.46	0.695	0.71	0.848	0.96	0.979
0.22	0.515	0.47	0.702	0.72	0.852	0.97	0.986
0.23	0.524	0.48	0.709	0.73	0.858	0.99	0.995

To determine MTD (Logarithmic mean temperature difference) find ratio LTD/GTD in Col. A. Find corresponding ratio MTD/GTD in Col. B. Then actual value MTD = Ratio from Col. B multiplied by actual GTD.

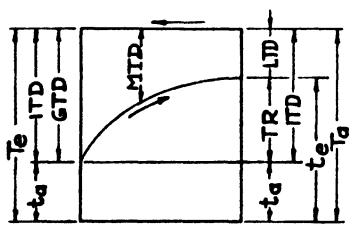
LOGARITHMIC MEAN TEMPERATURE DIFFERENCE

SYMBOLS

MTD — Logarithmic Mean Temperature Difference
GTD — Greatest temperature difference at either end.
LTD — Least temperature difference at either end.
ITD — Initial temp. diff. between hot & cold fluids.
FTD — Final temp. diff. between hot & cold fluids.
TR — Temperature rise of cold fluid.
TF — Temperature fall of hot fluid.
T_a — Initial temperature of hot fluid.
T_e — Final temperature of hot fluid.
t_a — Initial temperature of cold fluid.
t_e — Final temperature of cold fluid.

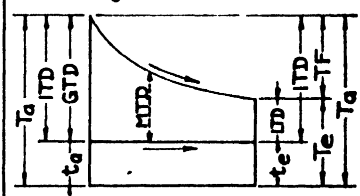
$$MTD = \frac{GTD - LTD}{\log_e \frac{GTD}{LTD}}$$

CASE A: Refrigeration or steam condensers; boiler feed water h'trs.



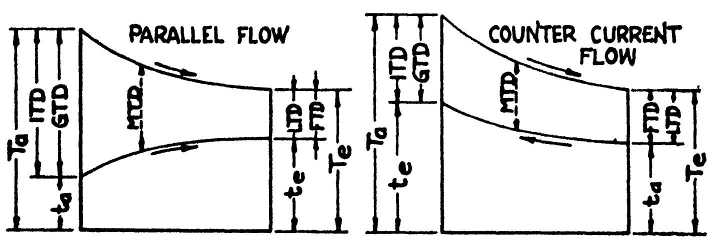
GTD = ITD
LTD = ITD − TR

CASE B: Steam boilers, refrigeration brine coolers, refrig. direct expansion coils, evaporator with hot liquid coils or jacket.



GTD = ITD
LTD = ITD − TF

CASE C: (Parallel flow or counter current flow): Brine oil coolers, brine cooling coils in cold storage rooms, water cooling coils in air conditioners, steam superheaters and economizers.



GTD = ITD LTD = FTD

FIGURE 4

65. Steam in Coils Submerged in Liquid

D = pounds steam required to heat F lb liquid with specific heat S from t_{fk} to t_{fu}

L = latent heat of one pound of steam

$$D = \frac{FS(t_{fu} - t_{fk})}{L - \frac{t_{fu} + t_k}{2}}$$

L = approximately 1000

H_e = square feet of heating surface

$$H_e = \frac{C}{U\theta_m}$$

C = heat supplied, Btu per hour

U = Btu per hour per square foot per degree difference over-all

$U = 28\sqrt{V_d}$ to $56\sqrt{V_d}$ without stirring

V_d = mean velocity of steam, feet per second. Mean velocity of liquid over coils may vary from 0.05 to 1 foot per second. The smaller figure is for large vessels and temperatures 140° F to 212° F. When liquids are agitated to 1 foot per second:

$$U = 46\sqrt{V_d} \times \sqrt{0.0126 + V_f}$$

V_d = steam velocity

V_f = liquid velocity

66. Steam-jacketed Kettles

$$H_e = \frac{C}{1400 \text{ to } 1800\theta_m} \quad \text{(no agitation)}$$

$$H_e = \frac{C}{3500\theta_m} \quad \text{(with agitation)}$$

H = heating surface, square feet

θ_m = logarithmic mean temperature difference

67. Steam "at Rest", Liquid Through Tubes

Consider steam at rest at $1\frac{1}{2}$ feet per second.

$U = 83\sqrt[3]{0.0126 + V}$ = Btu per hour per square foot per degree difference over-all

V = velocity of liquid

$$H_e = \frac{C}{\theta_m 83\sqrt{0.0126 + V}} = \text{surface, square feet}$$

68. Two Moving Liquids with Copper Wall Between

$$U = \frac{40}{\dfrac{1}{1 + 3.33\sqrt{v_1}} + \dfrac{1}{1 + 3.33\sqrt{v_2}}}$$

where

U = Btu per hour per square foot per degree difference over-all

v_1 and v_2 = feet per second

v_1 and v_2 vary from 0.003 to 6 ft per second

U varies from 23.8 to 190 for brass; with iron surface or viscous liquids increase surface 20%.

TABLE 39

HEAT TRANSFER FROM STEAM TO WATER IN CONDENSER TUBES

One Inch OD 18 BWG Tubes, Temperature Steam 130° F, Temperature Water 55° F.

	Water Velocity, Feet per Second			
	0.5	2	4	8
	Btu per Hour per Square Foot per ° F			
Clean tubes	150	400	600	900
Dirty tubes	135	325	460	600

69. Heat Transfer; Evaporators

In general, with non-viscous, non-scaling liquids, the horizontal-tube or standard vertical tube evaporator will have heat transfer coefficients between 200 and 500 Btu per square foot per hour per degree difference. The long-tube natural circulation evaporator and the forced circulation evaporator may yield a thermal transfer as high as 1000 to 1200 Btu per hour per square foot per degree difference. The transfer coefficient in an evaporator is the resultant of the steam film coefficient, the resistance of the metal wall (together with scale) and the resistance of the liquid film.

Since the liquid film is in natural or free convection it is practically impossible to compute the over-all heat coefficient.

70. Heat Transfer; Boilers

In heating boilers, practical rates of transfer between combustible gases and water will average approximately 3,300 Btu per square foot per hour for hand fired boilers; 4,000 Btu per square foot per hour for mechanically fired boilers. These values may be increased 50% at maximum load.

TABLE 40

CORRECTION FACTORS FOR DIRECT CAST IRON RADIATORS *

Reprinted by permission—Heating Ventilating Air Conditioning Guide 1940, Chapter 14, Page 265.

STEAM PRESSURE APPROXIMATELY: Gage Vacuum Inches Hg	Absolute Pounds Per Square Inch	Heating Medium Temperature °F Steam or Water	FACTORS FOR DIRECT CAST-IRON RADIATORS Room Temperature °F: 80	75	70	65	60	55	50
22.4	3.7	150	2.58	2.36	2.17	2.00	1.86	1.73	1.62
20.3	4.7	160	2.17	2.00	1.86	1.73	1.62	1.52	1.44
17.7	6.0	170	1.86	1.73	1.62	1.52	1.44	1.35	1.28
14.6	7.5	180	1.62	1.52	1.44	1.35	1.28	1.21	1.15
10.9	9.3	190	1.44	1.35	1.28	1.21	1.15	1.10	1.05
6.5	11.5	200	1.28	1.21	1.15	1.10	1.05	1.00	0.96
Pounds Per Square Inch									
1	15.6	215	1.10	1.05	1.00	0.96	0.92	0.88	0.85
6	21	230	0.96	0.92	0.88	0.85	0.81	0.78	0.76
15	30	250	0.81	0.78	0.76	0.73	0.70	0.68	0.66
27	42	270	0.70	0.68	0.66	0.64	0.62	0.60	0.58
52	67	300	0.58	0.57	0.55	0.53	0.52	0.51	0.49

* To determine the size of a radiator for a given space, divide the heat loss in Btu per hour by 240 and multiply the result by the proper factor from the above table.

To determine the heating capacity of a radiator under conditions other than the basic ones with the heating medium at a temperature of 215° F, and the room temperature at 70° F, divide the heating capacities at the basic conditions by the proper factor from the above table.

71. Heating Radiators (cast iron)

Cast iron steam radiators or hot water radiators are rated in square feet of surface emitting 240 Btu per square foot per pound steam at 215° F and standing free in a room at 70° F. The square foot of surface is theoretical, being the radiator Btu output divided by 240. The rating is known as *equivalent direct radiation* (E.D.R.). Since 215° F is the temperature of steam at 1 lb per square inch gage pressure the equivalent square foot of radiation has come to be called the *square foot of steam radiation*. At one time it was the custom to increase area by 60% when estimating hot water radiation, but the modern hot water system, equipped with a compression tank and relief valve, is no longer limited to a boiler temperature of 180° F. Accordingly, the term *square foot of radiation* has lost its meaning as a measurement of heat emitted by a hot water heating system. The surface which will emit 240 Btu per hour with steam at 215° F will also emit more or less than 240 Btu per hour as the average temperature of the water is raised or lowered. Since the modern hot water system is generally designed to lower the temperature of the water 20° F between inlet and outlet of the radiator, and the water enters the radiator at approximately boiler temperature, the average temperature of the radiator may be taken to be 10° F less than that of the boiler.

72. Heat Emission; Vertical Pipe Coils

The heat emission of vertical wall coils (pipes horizontal) containing steam at 215° F and surrounded by air at 70° F is given in Table 41.

For vertical wall coils (pipes vertical) the heat emission varies inversely as height of coil. It is customary to allow an average emission of 100 Btu per linear foot of 1¼ in. pipe 10 ft high.

TABLE 41

HEAT EMISSION, BTU PER LINEAR FOOT OF COIL
(not linear foot of horizontal pipe)

Size of Pipe	1 in.	1¼ in.	1½ in.
Single row	132	162	185
Four row	440	545	616
Eight row	651	796	907
Twelve row	812	1005	1135

TABLE 42

HEAT LOSSES FROM HORIZONTAL
BARE IRON PIPES

Expressed in Btu per linear foot per degree Fahrenheit difference in temperture between the pipe and surrounding still air at 70° F

Nominal Pipe Size (Inches)	Hot Water 120°F	150°F	180°F	210°F	Steam 227.1°F (5 Pounds)	297.7°F (50 Pounds)	337.9°F (100 Pounds)
	Temperature Difference 50°F	80°F	110°F	140°F	157.1°F	227.7°F	267.9°F
½	0.543	0.573	0.605	0.638	0.656	0.742	0.796
¾	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
1¼	0.979	1.02	1.087	1.15	1.184	1.345	1.450
1½	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
2½	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
3½	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
4½	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
9	4.80	5.05	5.350	5.71	5.885	6.824	7.400
12	6.25	6.62	6.995	7.46	7.670	8.900	9.670

TABLE 43

RADIATING SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Square Feet)	Nominal Pipe Size (Inches)	Surface Area (Square Feet)	Nominal Pipe Size (Inches)	Surface Area (Square Feet)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	9	2.519
1½	0.498	4	1.178	12	3.338

From A.S.H.V.E. Guide.

With a 10 mile per hour wind velocity the heat emission from bare pipe will be 1.5 to 2 times as great as in still air. Higher velocities increase the emission further. It will be noted that U varies from 2 to 4 Btu per square foot per degree difference per hour in still air.

73. Hot Blast Bare Pipe Coils

The temperature rise of a volume of air passing over heated surfaces in the form of iron pipes is dependent upon period of time of contact, which is proportional to number of coils deep in direction of flow. Each of the first three rows causes a rise of about 25° F with steam at 5 lb pressure, at 229° F and at a velocity of 1000 feet per minute, equivalent to about 500 feet per minute over net face area. If velocity is increased, the temperature rise is increased approximately one degree for every 100 feet per minute increased velocity. Although fin coils occupy less space than bare pipe coils, this advantage must be sacrificed in certain industrial air-conditioning installations where textile fibers, and other light substances floating in the atmosphere would clog up spaces between fins. With bare pipe coils, allow about 0.02 Btu to be absorbed per cubic foot of air per degree rise. This is based on an over-all heat transfer coefficient from steam to air of 8 Btu per square foot per hour per degree difference at 500 feet per minute velocity over net face area. The bare pipe coil is recommended where temperature conditions cause frost accumulation, and for cooling surface in spray systems.

74. Heat Transfer Through Fin Coils

The transfer of heat between the heating medium and the circulated air is determined by the following variables:

1. Temperature difference (logarithmic)
2. Design of surface
3. Velocity and character of air stream
4. Velocity and character of heating medium in tubes

Design of surface includes materials, thickness, height and spacing of fins and the ratio of fin surface to tube surface. Staggered tubes, as compared with the in-line arrangement, increase heat transfer, and corrugated fins are more effective than flat fins, although they clog up more quickly. The bond between fin and tube is important. The velocity of air usually considered is the coil face velocity. The heat transfer depends not only on fin design but upon velocity of the medium in the tubes. Heating (or cooling coils) are essentially heat exchangers and their performance depends on the following factors:

1. The over-all coefficient of heat transfer from the fluid within the coil to the air it heats (or cools)
2. The mean temperature difference between the fluid within the coil and the air flowing over the coil
3. The dimensions of the coil

The heating or cooling capacity of a coil for a definite room condition is expressed by the following formula:

$$Q = U \times MTD \times A \qquad (1)$$

where

Q = total heat transfer, Btu per hour

U = over-all coefficient of heat transfer, Btu per square foot of external coil surface per °F difference between the fluid within the coil and the air flowing over the coil

MTD = Logarithmic mean temperature difference between fluid and air

A = external area of surface in square feet

Because many variables enter into the design of fin coils, reliable performance information is generally based on tests of the coil under the expected conditions of operation. The results of many tests of various designs are recorded in the catalogs of manufacturers specializing in this branch of the air-conditioning industry. The catalogs are available to all contractors and engineers to assist them in the proper selection of heating or cooling coils.

75. Over-all Coefficient of Heat Transfer Through Coils

For a bare pipe coil the over-all coefficient of heat transfer for cooling or heating can be expressed by the basic formula

$$U = \frac{1}{\frac{R}{f_r} + \frac{x}{k} + \frac{1}{f_a}} \qquad (2)$$

where U = over-all coefficient, Btu per hour per square foot external surface per degree difference between fluid in coils and air

f_r = film coefficient of heat transfer between internal surface of the coil and fluid

f_a = film coefficient of heat transfer between air and external surface, Btu per hour per square foot external surface per ° F mean temperature difference

K = conductivity of the construction material of the bare pipe, Btu per hour per square foot per ° F per inch thickness

X = thickness of pipe wall, inches

R = ratio between external and internal surface of pipe, usually varying from 1.03 to 1.15. This ratio is inserted in the formula in order to place the internal fluid coefficient of heat transfer on the basis of external surface.

When pipe tubes are thin and of high conductivity the term X becomes negligible. Simplified, the bare pipe formula becomes

$$U = \frac{1}{\frac{R}{f_r} + \frac{1}{f_a}} \tag{3}$$

For finned coils the formula of H. B. Pownall is expressed as follows:

$$U = \frac{1}{\frac{R}{f_r} + \frac{1}{zf_a}} \tag{4}$$

in which the term z, called the fin efficiency, is introduced to allow for the resistance to heat flow encountered in the fins. The term R, in this case is the ratio of total extended surface to internal surface. For typical designs of finned coils for heating or cooling, the ratio R varies from 8 to 30. It will be noted that the R for fin coils is 8 to 30 times R for bare coils, which demonstrates that finned coil surface occupies much less space than bare pipe coils for equivalent surface. The following table gives averages of data collected from data supplied by various manufacturers of typical extended surface heaters.

Table 44

HEAT TRANSMISSION THROUGH FIN COILS

U = Btu per square foot extended surface per hour per degree difference between steam and air

Velocity Air Feet Per Minute	U
200	5.0
400	6.6
500	7.6
600	8.5
1000	12.0

76. Unit Heaters and Ventilators *

Unit ventilators are used to supply fresh air heated to slightly higher than room temperature. They are similar in design to unit heaters, and are provided with devices for proportioning the amount of recirculated air and outdoor air. If the unit is used only for circulating air, then radiators or some other equipment must be provided for heating the room. Unit ventilators are given two different capacity ratings, one established by measuring the volume of air with an anemometer and the other by the quantity of steam condensed. The A.S.H.V.E. standard code for testing and rating steam unit ventilators (*A.S.H.V.E. Trans.* **38,** 25, 1932) gives the following table and basic data referring to them.

TABLE 45

TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO

Cubic Feet of Air per Minute: Anemometer Rating	Cubic Feet of Air per Minute: Condensate Rating	Total Capacity in Square Feet, Equivalent Direct Radiation	Capacity Available for Heating the Room, Square Feet Equivalent Direct Radiation	Final Air Temperature °F
750	500	214	56	95
1000	750	320	84	95
1260	1000	427	112	95
1560	1250	534	141	95

If no direct heating surface (radiation) is installed to take care of the normal heat transfer losses, and the unit ventilator is to be used for both heating and ventilation, then the combined requirements must be taken care of by the unit ventilator. When all of the air handled by the unit is taken from the outside, the total heat to be supplied is obtained by means of Equations (5), (6) and (7).

* *Reprinted by courtesy of A.S.H.V.E.*

$$H_t = 0.24W(t_v - t_o) \tag{5}$$
$$W = d60Q \tag{6}$$
$$t_v = \frac{H}{0.24W} + t \tag{7}$$

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, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
t_v = temperature of the air leaving the unit, degrees Fahrenheit.
W = weight of air circulated, pounds per hour.
0.24 = specific heat of air at constant pressure.

From Equations (5), (6) and (7):

$$H_t = H + 0.24d60Q(t - t_o). \tag{8}$$

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70° F and all air is taken from the outside at zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system)?

Solution. Substituting in Equation (8)

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60 \times 1000(70 - 0)$$
$$= 99{,}600 \text{ Btu per hour}$$
$$t_v = \frac{24{,}000}{0.24 \times 0.075 \times 60 \times 1000} + 70 = 92.2° \text{ F}$$

When part of the air handled by the unit is taken from the room and the remainder from the outside,

$$H_t = 0.24W_o(t_y - t_o) + 0.24W_i(t_y - t) \tag{9}$$

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.

$$W_o = d_o 60Q_o \tag{10}$$

$$W_i = d_i 60Q_i \tag{11}$$

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.

$$t_y = \frac{H}{0.24(W_o + W_i)} + t \tag{12}$$

$$H_t = H + 0.24d_o 60Q_o(t - t_o) \tag{13}$$

Equations (9), (10), (11), (12) and (13) may be used in the same manner as is illustrated previously for Equations (5), (6), (7) and (8). It may be noted in Equation (13), 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 to 24,000 Btu per hour, 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 per hour. Units designed and operated on this principle show an average heat requirement and, therefore, a boiler capacity requirement of less than 50 per cent of that required for units taking all their air from the outside.

If all of the air is recirculated, the total heat required is the same as the heat loss of the room, or

$$H_t = H = 0.24W(t_y - t) \tag{14}$$

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:

$$H_v = 0.24W(t_y - t_o) \tag{15}$$

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.

77. Steam per Square Foot of Heating Surface *

The factors below are used in New York City for various classes of buildings. The factors are based on the attainment of an inside temperature of 70° F during 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:

* *Courtesy of A.S.H.V.E.*

1. *For 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.
2. *For office buildings* using steam during daylight hours only 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.
3. *For 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.
4. *For 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.
5. *For apartment houses* where high-class heating service is required. (Steam off at midnight). *Factor:* 650 lb per square foot of heating surface per season.
6. *For hotels* (commercial type) where very high-class service is required for 24 hours. *Factor:* 800 lb per square foot of heating surface per season.

One square foot of heating surface per 100 cubic feet of space is a fair ratio to use in rough computations of steam consumption per cubic foot of space. These data are more easily obtained than the actual area, in square feet, of the heating surface.

CHAPTER V

HEAT TRANSMISSION OF BUILDING MATERIALS

Wherever a temperature difference exists, nature makes every effort to destroy it by causing the heat to flow toward the colder place until the temperature becomes uniform. Where there are great temperature differences there is usually also a great difference in the quantity of water vapor in the warm and cold air. The warmer the air, the greater its moisture capacity. Both heat and moisture try to penetrate through the insulated wall into the cold compartment of a refrigerator, or through the wall of a heated home to the cold outside atmosphere. If the attempt succeeds, and moisture enters the insulation on the warm side, but is unable to get out on the cold side, it overloads the air near the cold surface and condenses into water. If this happens, the effectiveness of the insulation is impaired. Furthermore, when vapor condenses on a surface at a temperature below 32° F, it freezes, swells and even exerts enough pressure to break down the construction which contains the insulation. Heat and water vapor will flow through materials which will stop the flow of air or liquid water. The degree to which a building wall or insulation layer is able to withstand destructive moisture conditions and at the same time retard the flow of heat from the warm surface to the cold surface is a measure of its efficiency.

78. Symbols and Definitions

The following symbols which are in agreement with the recommendations of the National Research Council are used in the heat transmission equations in this chapter:

U = Thermal transmittance or over-all coefficient of heat transmission and is 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° F between the air on the inside and outside of the wall, floor, roof or ceiling.

k = Thermal conductivity and is 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° 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 and is 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° 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 and is 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° 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 surface or film-conductance and f_o the outside surface or film-conductance.

a = Thermal conductance of an air space and is 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° 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, the reciprocal of transmission, conductance or conductivity, *i.e.:*

$\frac{1}{U}$ = over-all or air-to-air resistance.

$\frac{1}{k}$ = internal resistivity.

$\frac{1}{C}$ = internal resistance.

$\frac{1}{f}$ = surface or film-resistance.

$\frac{1}{a}$ = air-space resistance.

x = thickness in inches.

The basic formula for the loss of heat by transmission through any surface is given in Equation (1).

$$H_t = AU(t - t_o) \tag{1}$$

where

H_t = heat loss transmitted through the wall, roof, ceiling, floor or glass, Btu per hour.

A = area of wall, glass, roof, ceiling, floor or other exposed surfaces, square feet.

U = coefficient of transmission, air to air, Btu per hour per square foot per degree Fahrenheit temperature difference (Chapter 4).

t = inside temperature near surface involved which may not necessarily be the so-called breathing line temperature, degrees Fahrenheit.

t_o = outside temperature, or temperature of adjacent unheated space or of the ground, degrees Fahrenheit.

Example 1. Calculate the transmission loss through an 8 in. brick wall having an area of 150 sq ft if the inside temperature (t) is 70° F and the outside temperature (t_o) is −10° F.

Solution. The coefficient of transmission (U) of a plain 8 in. brick wall is 0.50. The area (A) is 150 sq ft. Substituting in Equation (1):

$$H_t = 150 \times 0.50 \times [70 - (-10)] = 6000 \text{ Btu per hour.}$$

79. Calculation of Over-all Coefficient U

The reciprocals of the coefficients are treated as resistance units. This method simplifies combining the coefficients k, C or f of the individual parts of the wall whose over-all coefficient is desired. The formula for the over-all resistance R of a wall built up of homogeneous material of conductivity k and x inches thick is

$$R = \frac{1}{U} = \frac{1}{f_i} + \frac{x}{k} + \frac{x}{C} + \frac{1}{f_o} \tag{2}$$

If the coefficients f_i, k, C and f_o together with the thickness of the material x are known, the over-all coefficient U may be readily calculated as the reciprocal of the total resistance.

The formula for the total resistance of a compound wall built up of three homogeneous materials having conductivities k_1 k_2 and k_3 and thickness x_1, x_2 and x_3, respectively, and laid together without air spaces is

$$R = \frac{1}{U} = \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{1}{f_o} \tag{3}$$

In certain special forms of construction, such as tile with irregular air spaces, it is necessary to consider the conductance C of the unit as built instead of the unit conductivity K, and the resistance of the section is $\frac{1}{C}$ · When air spaces are present, equation (3) becomes

$$R = \frac{1}{U} + \frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{1}{f_o} + \frac{1}{a} \tag{3a}$$

The calculation of over-all heat coefficient U for a given wall is simple but the selection of proper coefficients k, C and f is often complicated. In such cases test methods are required. The results of the tests sponsored by the A.S.H.V.E. are generally accepted as authentic for heat loss calculations.

80. Surface Conductance Coefficients

Heat is transmitted to or from the surface of a wall by a combination of radiation, convection and conduction. The coefficient will be affected by any factor which has an influence on any one of these three types of transfer. The quantity of heat transmitted by radiation varies with character of surface and difference in temperature between it and surroundings. Convection transmission depends on air movement, surface, form and temperature difference. These variables may cause wide fluctuations in surface coefficients. These wide fluctuations necessitate the use of practical judgment in the selection of surface coefficients. Most tables giving coefficient of transmission U are based on the following assumptions: an outside wind velocity of 15 miles per hour; inside coefficient $f_i = 1.65$ Btu per square foot per degree difference per hour (4)
outside coefficient $f_o = 6.0$ Btu per square foot per degree difference per hour (5)
In single glass windows the surface resistance constitutes practically the entire resistance, and therefore is an important factor.

81. Air Space Conductance

Heat is conducted across an air space by a combination of radiation, conduction and convection. A brief table of conductances C for air spaces is given below.

In the calculation of the over-all coefficient U an average conductance of 1.10 Btu per square foot per degree difference per

TABLE 46

CONDUCTANCE OF AIR SPACES[a] AT VARIOUS MEAN TEMPERATURES

Mean Temperature °F	Conductances of Air Spaces for Various Widths in Inches 0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

[a] Thermal Resistance of Air Spaces by F. B. Rowley and A. B. Algren (A.S.H.V.E. *Trans.* 35, 165 (1929).

hour is generally used for spaces ¾ in., or more, wide. A large part of the heat transferred across air spaces bounded by ordinary building materials is transferred by radiation. Therefore, if such air spaces are faced with metallic surfaces such as aluminum foil or coated sheet steel, i.e., infra-red-reflective metal surfaces, the radiant heat transfer will be reduced, thus causing a major portion of the remaining heat to be transmitted by convection. To minimize the convection transfer the vertical air space should be at least three-quarters of an inch wide. A conductance of 0.46 may be used for a ¾ in. air space bounded by aluminum foil.

82. Practical Coefficients

Space limitations of this book will not permit the inclusion of all the combinations of materials that are used in building construction. However if the engineer has a table of fundamental conductivity values the proper coefficient may be estimated. Manufacturers of insulating and building materials publish complete tables in their catalogs. Attention is called to the fact that the conductivity values per inch of thickness do not afford a true basis of comparison between insulating materials as applied. The value of an insulating material is measured in terms of its heat resistance, which depends not only upon thermal conductivity per inch but also upon the thickness as installed and the manner of installation. Certain types of blanket insulations are designed to be installed between the studs of a frame building in such a manner as to give two air spaces. Each air space should be one inch thick and sealed at top and bottom to prevent the circulation of air from one space to the other.

TABLE 47

CONDUCTIVITIES (k) AND CONDUCTANCES (C) FOR BUILDING MATERIALS AND INSULATORS

The Coefficients are Expressed in Btu Per Hour Per Square Foot Per Degree Fahrenheit Temperature Difference. k is for One Inch Thickness. C is for Given Thickness.

Description Masonry	Conductivity or Conductance k	C	Resistivity Per Inch Thickness $1/k$	Resistance Thickness Given $1/C$
Brick—damp or wet	5.00		0.20	
Hollow tile 2 in.—½ in. plaster 2 sides		1.00		1.00
Hollow tile 4 in.—½ in. plaster 2 sides		0.60		1.67
Hollow tile 6 in.—½ in. plaster 2 sides		0.47		2.13
Hollow tile 8 in.—½ in. plaster 2 sides		0.52		1.92
Hollow tile 12 in.—½ in. plaster 2 sides		0.26		3.84
Concrete—sand and gravel aggregate	12.60		0.08	
Concrete blocks 8 in.—3 oval cores		0.90		1.10

TABLE 47 (*continued*)

Description	Conductivity or Conductance k	Conductivity or Conductance C	Resistivity Per Inch Thickness $1/k$	Resistance Thickness Given $1/C$
Masonry				
Gypsum—3 in. solid	2.41		0.42	
Cement plaster	8.00		0.13	
Gypsum plaster ⅜ in. thick		8.8		0.11
Roofing				
Asphalt composition or prepared		6.5		0.15
Asbestos shingles		6.0		0.17
Wood shingles		1.28		0.78
Built up—tar, felt, gravel ⅜ in. thick		3.53		0.28
Wood				
Cypress or douglas fir	0.67		1.49	
Yellow pine	0.86		1.16	
White pine	1.00		1.00	
Sawdust or shavings	0.42		2.44	
Insulation				
Semi-rigid—bats—hair felt	0.26		3.82	
Semi-rigid—vegetable fibers	0.27		3.70	
Rigid—mineral or vegetable fiber board	0.33		3.00	
Rigid—cork board	0.30		3.33	
Loose fill—mineral or rock wool	0.27		3.70	
Air space bounded by ordinary material	1.10		0.91	
Air space bounded by metal foil	0.46		2.17	
Frame Construction				
1 in. fir sheathing and building paper		0.86		1.16
1 in. pine sheathing building paper—pine lap		0.50		2.0
1 in. fir sheathing paper and stucco		0.82		1.2
Building Boards				
Compressed cement and asbestos sheets	2.70		0.37	
Pressed asbestos mill board	0.48		2.48	
Gypsum board ½ in.		2.82		0.35
Insulating board 25/32 in.		0.42		2.37
Plywood 5/16 in.		2.56		0.39
Surfaces				
Still air—non reflective—vertical		1.65		0.61
15 MPH wind velocity—vertical		6.00		0.17

TABLE 48

HEAT TRANSMISSION COEFFICIENTS U FOR BUILDING MATERIALS—WIND VELOCITY 15 MPH

Over-all thickness outside walls	2 in.	3 in.	4 in.	6 in.	8 in.	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	24 in.
	Btu Per Hour Per Square Foot Per Degree Difference Fahrenheit											
Brick												
Plaster, one side					0.46		0.34		0.27			
Air space, furred and plastered					0.22		0.18		0.16			
Plain					0.50		0.36		0.28			
Concrete—plain	1.21	1.08	0.99	0.79		0.62	0.60		0.48		0.41	
Furred, Lath and plaster				0.42		0.37	0.35		0.31		0.28	
4 in. brick veneer						0.57		0.48			0.39	
4 in. brick veneer and plaster						0.53		0.45			0.37	
4 in. brick veneer and plaster furred, lath						0.35		0.31			0.27	
Hollow tile—plain					0.42	0.41	0.32		0.27			
Plaster—one side					0.40	0.39	0.30		0.25			
Stucco and plaster					0.38	0.57	0.29		0.24			
Stucco, lath and plaster					0.28	0.27	0.22		0.19			
4 in. brick veneer						0.36	0.34	0.34	0.27			
4 in. brick veneer—plaster inside						0.34	0.33	0.32	0.26			
4 in. brick veneer—lath and plaster						0.25	0.25	0.24	0.21			

Table 48 (*continued*)

HEAT TRANSMISSION COEFFICIENTS U FOR BUILDING MATERIALS—WIND VELOCITY 15 MPH

Over-all thickness outside walls		2 in.	3 in.	4 in.	6 in.	8 in.	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	24 in.
		Btu Per Hour Per Square Foot Per Degree Difference Fahrenheit											
Stone—plain	Deduct 30% for 4 in. brick veneer					0.71		0.58		0.49		0.37	
Plastered						0.64		0.53		0.45		0.35	
Furred, lath and plaster						0.39		0.34		0.31		0.26	

Prepared surfaces—thickness, inches		1/8	1/4	1/2	3/4	1	1½	2	3	4
Asbestos—pressed mill board	K =	6.82	3.41	1.70	1.12	0.85				
Asbestos—corrugated	K =					0.48				
Celotex—fiber board	K =			0.66		0.33		0.17		
Gypsum board	K =	11.28	5.04	2.82	1.94	1.41				
Cork board	K =					0.30	0.22	0.15	0.10	0.075
(15 MPH air) cork board 8 in. to 12 in. masonry wall	K =						0.13			
Rock wool fill 8 in. to 12 in. masonry wall	K =						0.13			
Hair felt or semi-rigid insulation	K =					0.33		0.17		0.085

$$\text{Resistance } R = \frac{1}{U} = \frac{1}{f_i} + \frac{X}{k} + \frac{X_1}{k_1} + \frac{X_2}{k_2} \ldots + \frac{1}{f_o}$$

X = thickness k = conductivity $f_i = 6.0$ $f_o = 1.65$

U = over-all coefficient—air to air

TABLE 49

HEAT TRANSMISSION COEFFICIENTS *U* FOR BUILDING MATERIALS

Btu Per Hour Per Square Foot Per Degree Difference

Flat Roofs With Built Up Roofing	
1 in. wood—no insulation	0.49
1 in. wood 1 in. rigid insulation	0.20
1 in. wood lath and plaster ceiling	0.32
1 in. wood lath and plaster ceiling 1 in. insulation	0.16
2 in. concrete—no insulation	0.82
2 in. concrete 1 in. rigid insulation	0.24
2 in. concrete lath and plaster ceiling	0.42
2 in. concrete lath and plaster ceiling 1 in. insulation	0.19
6 in. concrete—no insulation	0.64
6 in. concrete 1 in. rigid insulation	0.33
6 in. concrete lath and plaster ceiling	0.37
6 in. concrete lath and plaster ceiling 1 in. insulation	0.18
1⅝ in. precast cement tile—no insulation	0.84
1⅝ in. precast cement tile 1 in. rigid insulation	0.24
1⅝ in. precast cement tile lath and plaster ceiling	0.43
1⅝ in. precast cement tile lath and plaster ceiling insulation	0.19
Flat metal roof—no insulation	0.95
Flat metal roof 1 in. insulation	0.25
Flat metal roof lath and plaster ceiling	0.46
Flat metal roof lath and plaster ceiling 1 in. insulation	0.19
Ceilings and Wood Flooring—¼ in. Linoleum	
¾ in. flooring—no ceiling—no insulation	0.34
Lath and plaster ceiling	0.24
½ in. rigid insulation and plaster	0.18
1 in. flexible insulation lath and plaster	0.12
Ceilings and Bare Concrete Floors	
6 in. concrete—no ceiling	0.59
6 in. concrete lath and plaster, furred	0.35
6 in. concrete lath and plaster, furred ½ in. insulation	0.23
6 in. concrete ½ in. plaster 1½ in. corkboard	0.14
Deduct 25% if wood flooring on sleepers	
For floors on ground—all thicknesses	0.10
Interior Walls	
4 in. clay tile—no plaster	0.45
4 in. clay tile plaster one side	0.42
4 in. hollow gypsum tile—no plaster	0.30
Studs—1 in. wood, lath and plaster—one side	0.62
Studs—1 in. wood, lath and plaster 2 sides—air space	0.34
Studs—1 in. wood, lath and plaster 2 sides—Rock wool	0.078
Studs—metal lath and plaster—one side	0.60
Studs—metal lath and plaster two sides (air space)	0.39
5/16 in. wall or plasterboard	1.00

Table 50

HEAT TRANSMISSION COEFFICIENTS *U* FOR BUILDING MATERIALS

Btu Per Hour Per Square Foot Per Degree Difference
(15 MPH Wind Velocity)

Windows and Skylights	
Single	1.13
Double	0.45
Triple	0.28
Doors and Partitions	
1 in. thick exposed	0.69
1 in. thick exposed with glass storm door	0.42
2 in. thick exposed	0.46
2 in. thick exposed with glass storm door	0.32
Hollow Glass Blocks	
Smooth surface 7¾ in. x 7¾ in. x 3⅞ in.	
Still air both sides	0.80
Still air inside, 15 MPH wind outside	0.98
Ribbed surface 7¾ in. x 7¾ in. x 3⅞ in.	
Still air both sides	0.76
Still air inside 15 MPH wind outside	0.92
Frame Building Construction	
Clapboard exterior—no plaster or insulation	0.65
Interior plaster on lath—no insulation	0.25
Interior plaster on lath ½ in. flexible insulation	0.17
Interior plaster on lath Rock wool fill	0.072
Interior plaster on 1 in. rigid insulation	0.15
Exterior stucco—interior plaster on lath—no insulation	0.30
Exterior stucco interior plaster on lath ½ in. flexible insulation	0.20
Exterior stucco interior plaster on lath Rock wool fill	0.076
Exterior stucco interior plaster on 1 in. rigid insulation	0.16
Exterior brick veneer interior plaster on lath—no insulation	0.27
Exterior brick veneer interior plaster on lath ½ in. flexible insulation	0.18
Exterior brick veneer interior plaster on lath—Rock wool	0.074
Exterior brick veneer interior plaster on lath 1 in. rigid insulation	0.15
Steel Exterior	
Sheet (marine bulkheads)	1.20
Corrugated	1.50
Corrugated asbestos protected	0.60
Channel, wire lath—plaster—both sides	0.34
Channel, wire lath—plaster Rock wool fill	0.21

83. Effect of Moisture on Insulation

Increase in conductivity with moisture is about that which is to be expected due to an increase in density. Conductivity tables generally refer to a bone-dry state, and, unless this state can be assured at the time the material is installed and hermetically sealed in place, allowance must be made for depreciation caused by increase of moisture content. Distinction is made between penetration of moisture by capillarity and by vapor or wind pressure. Various forms of moisture attacking insulating material are distinguished, e.g., water vapor, water film, water in quantity sufficient for submersion.

Cork is non-absorbent but adsorbs up to 20% water vapor by weight. Slag wool or glass wool fibers have a small and constant moisture content in air of varying humidity, but are of high capillarity. Moisture, once it has entered the insulating layer may go into chemical combination, remain free or become absorbed. When heat flows through insulation, the moisture content of the insulation crowds toward the cold surface. If the cold surface is not water-proof, exterior moisture will enter the refrigerator, increasing the moisture load, and deposit on coils.

84. Insulation

Aside from the question of price (which includes the factor of availability), seven factors determine selection of the proper insulating material:

1. Conductivity per unit volume
2. Conductivity per unit weight
3. Heat storage capacity referred to weight per unit volume
4. Performance temperature
5. Moisture content (permanent)
6. Moisture content (temporary)
7. Fire hazard

The temperature at which the insulation is to perform may be a factor for other reasons. Whereas aluminum foil, asbestos and glass fiber, will stand up under temperatures of several hundred degrees, certain vegetable fibers and hair felt insulation (otherwise highly desirable insulators), will begin to emit odors at temperatures above 125° F. Vegetable fiber insulation is often ruled out for use against the heat from steam, for fear of fire, but mineral products are acceptable.

85. Heat Loss Through Hot Pipe Covering

The heat transfer through uninsulated pipes and ducts is of considerable magnitude. Careful consideration must be given to this factor in a properly designed system, especially where pipes pass through an air-conditioned space. The thicknesses of insulation which ordinarily are specified under various high-temperature conditions are given in Table 51.

TABLE 51

THICKNESSES OF INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Pound Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation (1) Pipes Larger Than 4 In.	(2) Pipes 2 In. to 4 In.	(3) Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

The data in column (1) may be applied in insulating or lagging boilers, tanks or flat surfaces at high temperatures. The standard covering is usually applied in molded sections (about 3 ft long) the seams of which are staggered and filled with magnesia plaster. These sections are bound to the pipe by means

[a] All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

Courtesy of A.S.H.V.E.

TABLE 52

HEAT LOSSES FROM BARE AND INSULATED PIPE

Based on Still Air and Room Temperature of 90° F—Heat Losses Given in Btu Per Hour Per Linear Foot of Pipe—Coefficient "U" = 0.55 Btu Per Hour Per Square Foot Insulation Interior Surface Per Degree Difference vs. Steam and Outside Air for Thickness Required Which Varies from Standard 85% Magnesia to 4½ in. Special

Steam Temperature °F	Surface Temperature °F		Nominal Pipe Size													
			½	¾	1	1¼	1½	2	2½	3	3½	4	4½	5	6	8
270	125	Bare	102	127	160	202	231	288	348	425	485	546	607	675	804	1047
		Insulated	33	37	43	50	55	59	67	78	86	90	98	107	123	144
340	130	Bare	162	202	254	321	367	459	554	676	772	869	965	1073	1278	1665
		Insulated	47	53	61	71	78	84	96	113	123	127	139	152	176	204
390	135	Bare	215	269	336	425	487	608	735	897	1024	1152	1279	1423	1695	2207
		Insulated	57	65	74	87	95	84	95	109	119	110	118	130	147	180
550	140	Bare	462	577	722	913	1045	1305	1580	1924	2197	2471	2746	3055	3638	4735
		Insulated	75	84	95	108	118	126	131	149	163	165	179	194	220	250
800	145	Bare	1077	1347	1685	2131	2439	3047	3689	4490	5130	5773	6412	7132	8494	11050
		Insulated	120	134	149	169	183	155	171	185	207	215	219	228	258	307
1000	150	Bare	1760	2200	2750	3480	3990	4975	6020	7350	8370	9425	10450	11650	13880	18000
		Insulated	144	154	1157	162	164	151	148	146	151	148	145	143	143	146

Per Cent Increase in Heat Loss

Steam Temperature °F	Room Temperature 60°	70°	80°	90°
270	16.7	11	5	0
340	12	8	4	0
390	10	6.6	3.5	0
550	6.5	4.3	2.2	0
800	4.2	2.8	1.4	0
1000	3.3	2.2	1.1	0

Per Cent Heat Loss for Bare Pipe

Air Velocity Feet Per Minute	300	500	900
Surface Temperature	Per Cent Increase in Loss		
200	1.55	1.42	2.1
600	1.38	1.55	1.75
1000	1.22	1.85	1.4

Condensation: Btu loss does not change temperature but causes condensation of steam. Allow average of one lb condensation per 900 Btu.

of galvanized iron wire or netting, over which is wrapped a coat of rosin-sized paper, followed by 8 oz canvas securely sewed on. Standard covering consists of 65 to 85% carbonate of magnesia mixed with 15% of asbestos. Asbestos air cell covering consists of several layers of corrugated asbestos sheets forming a laminated covering with air spaces between corrugations.

86. Underground Hot Pipe Insulation

Tunnels are distinguished from conduits in that they contain sufficient space for men to pass along pipes for inspection and repairs. The estimation of proper thickness depends upon ground temperature and constants that apply to pipes exposed in tunnels do not apply to underground conduits. The theory is further complicated by the fact that the heat loss from a pipe in air occurs by conduction, convection and radiation, whereas in the case of a pipe buried in the ground the heat loss is by conduction only. Generally speaking, the economical thickness of insulation for underground pipe is reported to be approximately ½ in. less than for pipes in air. Insulation may consist of 3 in. thick magnesia block or loose fill as tabulated below.

TABLE 53

THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

Steam Pressures (Pound Gage) or Conditions	Steam Temperatures Degrees Fahrenheit	Minimum Thickness of Insulation in Inches: Steam Lines: Pipes Less Than 4 In.	Steam Lines: Pipes 4 In. to 10 In.	Steam Lines: Pipes Larger Than 12 In.	Return Lines: Pipes Less Than 4 In.	Return Lines: Pipes 4 In. and Larger	Minimum Distance Between Steam and Return
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

Courtesy of A.S.H.V.E.

87. Heat Transmission Through Uninsulated Ducts

The heat transmission through sheet metal duct walls is mainly a function of the surface characteristics of the metal, since the thickness of the metal itself is not enough to retard the flow of heat. Surface resistance is the dominating factor and not the thickness of metal. Black surfaces offer less resistance while bright surfaces offer greater resistance. For ducts in service at normal velocities and temperatures, the coefficient for black iron is approximately 1.6 Btu per hour per square foot per ° F difference between the mean temperature in the duct and the temperature of the surrounding air; for galvanized steel or iron ducts the coefficient averages approximately 1.1 as will be seen in Table 54.

TABLE 54

COEFFICIENT f OF HEAT TRANSFER AIR TO AIR THROUGH UNINSULATED GALVANIZED IRON DUCTS

(Btu per hour per square foot per °F)

Velocity Air in Duct	Temperature Difference, ° F			
	20	40	80	100
120	0.20	0.25	0.35	0.40
360	0.50	0.65	0 70	0.75
600	0.65	0.80	0.90	0.85
1000	1.00	1.05	1.15	1.19
1800	1.05	1.10	1.20	1.22

Increase 50% for black steel surfaces.

Formulas: Calculation of temperature change in uninsulated ducts.

The heat loss from a given length of uninsulated duct is expressed by

$$H = fPL\left[\left(\frac{t_1 + t_2}{2} - t_3\right)\right] \qquad (6)$$

the heat given up by the air is

$$H = 0.24M(t_1 - t_2) = \frac{14.4AS(t_1 - t_2)}{v} \qquad (7)$$

Equation (6) = Equation (7)

$$\frac{t_1 + t_2 - 2t_3}{t_1 - t_2} = \frac{28.8AS}{fPLv}$$

where

f = coefficient air to air through uninsulated duct Btu per square foot per hour per ° F difference
H = heat loss through duct walls Btu per hour
P = perimeter of duct in feet
t_1 = temperature of air entering duct
t_2 = temperature of air leaving duct
t_3 = temperature of air surrounding duct
M = weight in pounds of air per hour going through duct
A = cross sectional area of duct
S = velocity of air in duct, feet per minute at specified temperature t_1
L = length of duct in feet
v = cubic feet per pound dry air
0.24 = specific heat of air

CFM = cubic feet air per minute = $\dfrac{Mv}{60}$

v = 13.6 at 80° F

Problem. In a comfort cooling system the return air enters a galvanized iron uninsulated duct 100 ft long, 12 in. deep, 36 in. wide at a temperature of 80° F and travels at a velocity of 1000 ft per minute. The outside air is at 95° F. What will be the temperature rise due to lack of insulation?

Solution. Assume f = 1 and the answer is a rise of 3.5° F, a loss of approximately 4.5%.

88. Heat Transfer Through Insulated Ducts

The same formula is used as in topic 87 except that over-all heat coefficient U is substituted for f. Values of U are given in Table 55; the average value in ordinary installations amounts to $\frac{1}{10}$ Btu per square foot per hour per degree difference between air inside and still air outside of duct.

$$U = \frac{1}{R_f + R_d} = \frac{1}{3 + 0.75} = \frac{1}{3.75} = 0.26$$

for insulation one inch thick Table 55 may also be used.

TABLE 55

HEAT TRANSMISSION THROUGH DUCT WALLS INSULATED WITH MATERIALS OF VARYING CONDUCTIVITIES

Values are expressed in Btu per hour per square foot of flat surface per °F difference in temperature between air inside and still air outside at 90° F for cold air and 50° F for warm air in ducts

Conductivity of Insulation at 86°F Mean Temp.	Thickness of Insulation (Inches)	Cold Air 40°F	60°F	80°F	Warm Air 90°F	120°F	150°F	180°F
		Temperature Difference						
		50°F	30°F	10°F	40°F	70°F	100°F	130°F
0.200	½	0.319	0.323	0.328	0.324	0.330	0.337	0.344
	1	0.175	0.177	0.180	0.178	0.181	0.184	0.188
	1½	0.121	0.122	0.124	——	0.125	0.127	0.129
	2	0.092	0.093	0.095	——	——	——	——
0.250	½	0.382	0.387	0.392	0.390	0.397	0.404	0.412
	1	0.214	0.217	0.220	0.218	0.221	0.225	0.229
	1½	0.149	0.151	0.153	——	0.154	0.156	0.159
	2	0.114	0.115	0.117	——	——	——	——
0.300	½	0.440	0.445	0.450	0.448	0.457	0.466	0.475
	1	0.252	0.255	0.258	0.256	0.260	0.264	0.268
	1½	0.176	0.178	0.180	——	0.181	0.184	0.187
	2	0.135	0.137	0.139	——	——	——	——
0.350	½	0.494	0.499	0.505	0.502	0.511	0.521	0.530
	1	0.286	0.289	0.292	0.290	0.295	0.300	0.306
	1½	0.202	0.204	0.207	——	0.208	0.211	0.215
	2	0.156	0.158	0.160	——	——	——	——

TABLE 55 (*continued*)

Conductivity of Insulation at 86°F Mean Temp.	Thickness of Insulation (Inches)	Cold Air			Warm Air			
		40°F	60°F	80°F	90°F	120°F	150°F	180°F
		Temperature Difference						
		50°F	30°F	10°F	40°F	70°F	100°F	130°F
0.450	½	—	0.596	0.602	0.599	0.610	0.621	0.633
	1	—	0.356	0.360	0.358	0.364	0.370	0.376
	1½	—	0.254	0.257	—	0.259	0.263	0.267
	2	—	0.198	0.200	—	—	—	—
0.550	½	—	0.682	0.688	0.685	0.699	0.714	0.730
	1	—	0.417	0.422	0.418	0.425	0.432	0.440
	1½	—	0.302	0.305	—	0.307	0.312	0.317
	2	—	0.236	0.239	—	—	—	—

For round ducts less than 30 in. diameter, increase heat transmission values by the following percentages:

Thickness of Insulation (Inches)	½	1	1½	2
21 to 30 in. Duct Diameter	1%	2%	3%	4%
12 to 21 in. Duct Diameter	3%	5%	7%	9%

Data from A.S.H.V.E. Guide.

89. Calculation of Temperature Change in Insulated Ducts

The necessity of insulating cooled air ducts passing through warm spaces is demonstrated by solving for the temperature rise using the same quantities as in the preceding problem, and adding a 2 in. layer of insulation having a coefficient U of 0.25.

$$\frac{t_1 + t_2 - 2t_3}{t_1 - t_2} = \frac{28.8AS}{UPLv}$$

$$P = 8$$
$$v = 13.6$$
$$A = 3$$
$$S = 1000$$
$$U = 0.25$$
$$L = 100$$
$$t_1 = 80$$
$$t_3 = 95$$

$$\frac{80 + t_2 - 190}{80 - t_2} = \frac{28.8 \times 3 \times 1000}{0.25 \times 8 \times 100 \times 13.6} = 31.8$$

$$t_2 = 80.9° \text{ F}$$
$$t_2 - t_1 = 0.9° \text{ F rise}$$

Therefore under the same conditions, insulation prevents a loss of 3.41 degrees cooling effect. Since a total rise would be 15° F a saving of 22.8% results.

Chapter VI

HEAT TRANSMISSION FACTORS IN COOLING

In a discussion of air-conditioned spaces, when the outside or ambient dry-bulb temperature is lower than the dry-bulb temperature of the space to be conditioned, the process for maintaining the predetermined temperature differential is called *heating*. When the outside or ambient dry-bulb temperature is higher than the dry-bulb temperature of the conditioned space, the process to maintain the predetermined differential is called *cooling*. In common language *cooling* is associated with maintaining comfort conditions at an average temperature differential of 15° F. *Refrigeration* is generally associated with temperature differentials of over 100° F, to maintain refrigerators at temperatures ranging from considerably below zero to as high as 45° F in the domestic refrigerator.

Industrial air conditioning covers the whole range of refrigeration and heating.

90. Insulation of Refrigerated Spaces

Three properties of insulation are important:

(1) Conductivity per unit volume

(2) Conductivity per unit weight

(3) Heat storage capacity by weight per unit volume.

Aside from the question of price and availability, any one of the above factors can be of primary importance. Where the space occupied by insulation is most valuable, as in household or short-order refrigerators, the first item is of great importance. Where transportation enters the picture, as in a refrigerated car or truck, the second item may be most essential. Where fluctuations of temperature are to be encountered the third item may overshadow the others; e.g., comfort cooling of a house in summer or periodically used refrigerators.

An entirely satisfactory commercial method for applying insulation is difficult to find. The only precaution one can take is to demand painstaking workmanship. No perfectly waterproof material is absolutely vaporproof.

The author supervised construction of a submarine cable drying plant in which specifications called for air humidities "not exceeding one half of one per cent at 80° F." It was impossible to meet these specifications even though a waterproofing membrane which was considered commercially perfect was installed over and between the layers of cork insulation. Satisfactory results were obtained only by completely lining floor, walls and ceiling with 18 gage galvanized iron with welded joints; this created a hermetically vapor-sealed space. In general cold-storage practice, however, this expensive method is not considered justified. Low temperatures demand efficient vaporproofing in addition to efficient insulation.*

TABLE 56

PRACTICE IN INSULATION, PURE CORK BOARD IN COLD STORAGE

	Exposure	Thickness Inches
Exterior cooler walls		
	North	4
	Northeast	4
	Northwest	4
	South	5
	Southeast	5
	Southwest	5

* Cost of construction of insulation is given under the subject of Estimating Costs, Chapter XXIV.

(For walls facing east or west use 5 in. on that wall exposed to prevailing summer winds and 4 in. on the other.)

Exterior freezer walls		
	North	6
	Northeast	6
	Northwest	6
	South	7
	Southeast	7
	Southwest	7

(For walls facing east or west, use 7 in. on that wall exposed to the prevailing summer winds and 6 in. on the other.)

Partition walls on floors between freezers and coolers	6
Cooler floors exposed to usual atmospheric temperatures	5
Freezer floors similarly exposed	7
Roof over cooler space	6
Roof over freezer space	8

Courtesy of A.S.R.E.

91. Thickness of Refrigerated Space Insulation (Semi-rigid)

For refrigerated space insulation, the thickness of insulating material (exclusive of lining or floor covering) consisting of non-rigid type rock wool or felted coverings is indicated in the following table.

TABLE 57

THICKNESS OF REFRIGERATED SPACE INSULATION

Operating Temperature of Space, ° F	Thickness of Non-Rigid Insulation Inches
Below 5	10
5 to 19	8
20 to 39	6
40 and above	4

92. Thickness of Low Temperature Pipe and Tank Insulation

Pipes carrying Freon, ammonia, brine or other low temperature liquids or gases are insulated as indicated in Table 58:

TABLE 58

LOW TEMPERATURE PIPE AND TANK INSULATION

Temperature Range ° F	Total Thickness of All Layers, Inches
—30 to 0 (Heavy brine)	6
0 to 30 (Brine)	4.5
30 to 45 (Ice water)	3

93. Cylindrical Tanks for Cold Liquids

All cylindrical tanks, such as brine coolers, cold water storage tanks, accumulators, etc. are usually insulated as indicated in the following table:

TABLE 59

INSULATION THICKNESS OF CYLINDRICAL TANKS FOR COLD LIQUIDS

Temperature Range ° F	Thickness Inches
Below 5	6
5 to 19	5
20 to 34	4
35 to 50	3
Above 50	2

94. Insulation of Low-Temperature Pipes

When the surfaces of pipe are maintained at temperatures lower than the surrounding atmosphere insulation is required to reduce transmission losses and also to prevent condensation and frost, when the surfaces are below the dew point of the

Table 60
HEAT GAINS FOR INSULATED COLD PIPES
Based on materials having conductivity, $k = 0.30$

Nominal Pipe Size (Inches)	Ice Water Thickness: Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Square Foot Pipe Surface	Brine Thickness: Thickness of Insulation (Inches)	Btu Per Linear Foot	Btu Per Square Foot Pipe Surface	Heavy Brine Thickness: Thickness of Insulation (Inches)	Btu per Linear Foot	Btu Per Square Foot Pipe Surface
½	1.5	0.110	0.502	2.0	0.098	0.446	2.8	0.087	0.394
¾	1.6	0.119	0.431	2.0	0.111	0.405	2.9	0.094	0.340
1	1.6	0.139	0.403	2.0	0.124	0.352	3.0	0.104	0.294
1¼	1.6	0.155	0.357	2.4	0.131	0.300	3.1	0.113	0.260
1½	1.5	0.174	0.351	2.5	0.134	0.270	3.2	0.118	0.238
2	1.5	0.200	0.322	2.5	0.151	0.244	3.3	0.134	0.214
2½	1.5	0.228	0.303	2.6	0.170	0.226	3.3	0.147	0.197
3	1.5	0.269	0.293	2.7	0.186	0.202	3.4	0.162	0.176
3½	1.5	0.295	0.282	2.9	0.191	0.183	3.5	0.176	0.167
4	1.7	0.294	0.248	2.9	0.209	0.176	3.7	0.182	0.154
5	1.7	0.349	0.239	3.0	0.241	0.165	3.9	0.202	0.138
6	1.7	0.404	0.233	3.0	0.259	0.150	4.0	0.228	0.130
8	1.9	0.455	0.201	3.0	0.318	0.140	4.0	0.263	0.116
10	1.9	0.559	0.198	3.0	0.383	0.135	4.0	0.309	0.110
12	1.9	0.648	0.194	3.0	0.438	0.131	4.0	0.364	0.108

surrounding atmosphere. The thickness of insulation required to prevent sweating is that required to raise the temperature of exposed surface of insulation a few degrees higher than a design dew point for the corresponding air temperature and relative humidity. Since the dew point corresponding to 95° F dry-bulb and 75° F wet-bulb is 66° F, to avoid condensation a design temperature of approximately 70° F is suggested.*

TABLE 61

DATA FOR ESTIMATING REQUIREMENTS TO PREVENT FREEZING OF WATER IN PIPES WITH SURROUNDING AIR AT —18° F

Nominal Pipe Size (Inches)	Number of Hours to Cool 42° F Water to Freezing Point			Water Flow Required at 42° F to Prevent Freezing, Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches (Conductivity, $k = 0.30$)					
	2	3	4	2	3	4
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.43
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.45	1.93

—*Courtesy of A.S.H.V.E.*

95. Anti-freeze Pipe Covering

Frozen water in pipes causes service interruptions and serious damage. Insulation can materially prolong the time required for water to give up its heat, and if velocity of flow is maintained freezing may be prevented. Column 4 in the following table gives the minimum quantity of water at 42° F that should be supplied each hour for every linear foot of pipe, in order to prevent the temperature of the water from dropping to the

* Costs of cold-pipe covering are discussed under the subject of Estimating, Chapter XXIV.

freezing point, 32° F. If water remains stationary longer than the time designated in the table it is necessary to install a steam line, hot water line or electric resistance heater along the exposed pipe surface.

96. Surface Cooling with Bare Pipe Coils

When air is forced, by means of a fan, over the surfaces of bare pipes, through which a refrigerant circulates, the air is cooled by contact with the cold surface. This type of cooler formerly was called a *dry coil bunker*, but now is called a *surface cooler*, as distinguished from a *spray type cooler.*

The effect is quite different from the cooling effect with a spray of water or brine into the air, the spray at the same time wetting the coils. Air leaves the spray chamber in a practically saturated condition at a predetermined dew point governed by the temperature of the water and the humidifying efficiency of the spraying apparatus. Air leaves the surface cooler in an unsaturated condition. Moisture, it is true, condenses but owing to the impossibility of perfect contact of all the air with all the surface for the same period of time complete saturation does not occur. However a portion of air attains a dew point within a few degrees of the refrigerant temperature. A surface cooler is generally utilized when close regulation of humidity is not required, the only requirement being to reduce moisture content as much as possible.

When air is passed over 14 banks of standard 1 in. pipe coils (no fins), with the pipes bent at 3 in. coil centers, at a velocity of 400 ft per minute over net face area, a heat transmission factor from refrigerant to air of 2 to 8 Btu per degree difference per square foot per hour may be expected, depending on the velocity of air over the coils.

It is to be noted that only sensible heat is accounted for by temperature difference. However if refrigerating capacity is sufficient to lower the temperature of the air below the dew point, the pipe surfaces will be moistened and the transmission coefficient will be increased. If the latent heat or moisture load does not exceed 33.3% of the sensible heat load or 25% of the total load additional coils to counteract the wet surfaces are not necessary. If the sensible heat factor (the ratio of sensible heat to total heat) is less than 66.7%, extra coils will be necessary. The transmission coefficient U may rise to 10 in rooms above 50° F. However, if room temperature is below 50° F, frost accumulates on the coils and U drops to approximately 5 Btu per square foot per hour per degree difference at 30° F room temperature. When the room temperature is 20° F, U may approach 3.5 and when it is 10° F, U may drop as low as 2 Btu per hour per square foot per degree difference.

When the moisture load exceeds 25% of the total heat load, an excess of frost will accumulate and interfere seriously with heat transfer. Under continuous operating conditions a spray of brine over the coils proves effective in removing frost at temperatures below freezing. For comfort cooling systems, where the refrigerant temperatures are 28° F to 32° F, a spray of water over the coils has proved effective. If a spray is not used extra coil surface is required.

TABLE 62

SURFACE PIPE COIL COOLER
STANDARD 1 IN. PIPE—NO FINS
12 BANKS OF COILS WITH 3 IN. CENTERS
CONSTANT VOLUME AIR CIRCULATION
FLOODED OR BRINE SYSTEM

Temperature ° F Refrigerant 20° F Mean Difference, ° F	Temperature ° F Refrigerant 40° F Mean Difference, ° F	Room Temperature ° F	Temperature of Air Leaving Cooler ° F	U, Btu Per Hour Per Square Foot Per Degree Difference, Fluid to Fluid
50	30	80	62	10
44	21.5	70	45	10
25		50	42	8
16		40	34	5
8		30	28	4.5
—1.5		20	19	2.0
—11.5		10	9	0.7

Velocity of air, 400 ft per minute
To calculate direct expansion, multiply U by 0.7

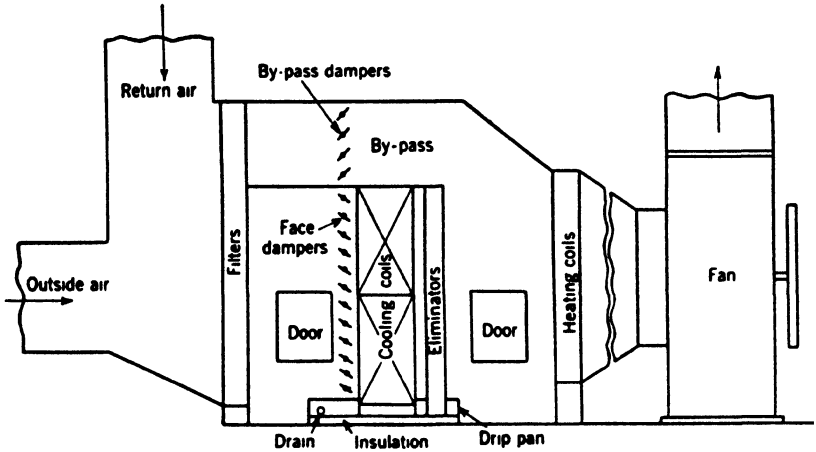


FIGURE 5

Typical Arrangement of Cooling Coils in a Central System
—*Courtesy of A.S.H.V.E.*

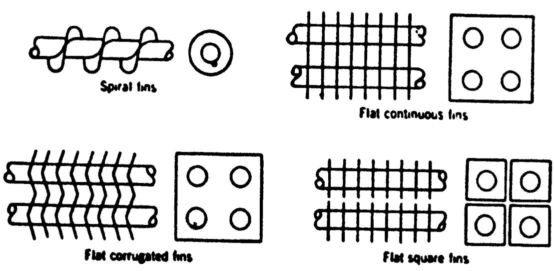


FIGURE 6

Types of Fin-Coil Arrangement
—*Courtesy of A.S.H.V.E.*

97. Shell and Tube Water Coolers (See Figures 71 and 72)

Refrigeration practice generally includes refrigerant temperatures below 32° F; air-conditioning refrigerant temperatures are generally above 32° F. The actual cooling effect is delivered at the evaporator in which the liquid refrigerant vaporizes continuously while the compressor is operating. The medium to be cooled is either air, brine or water. This topic is concerned largely with the cooling to about 45° F of water, which is to be delivered after cooling to the air-conditioning apparatus. There are two important factors to be considered in selecting the capacity of a water cooler. The heat transfer must be as high as possible and the cooler must be free from oil trouble. The quantitative cooling effect in an evaporator is expressed by the following formula

$$Q = wc\,(t_a - t_b) \tag{1}$$

where

Q = Btu per hour from the refrigerant
w = pounds of water circulated per hour
c = specific heat of water
t_a = entering temperature of water
t_b = leaving temperature of water

The following formula gives the heat transfer surface area required in the shell and tube evaporator.

$$Q = ADU \tag{2}$$

where

A = square feet outside pipe transfer surface
D = logarithmic mean temperature difference
U = over-all heat transfer coefficient, Btu per hour per square foot per degree

Equating equations (1) and (2); $c = 1$:

$$A = \frac{w(t_a - t_b)}{DU}$$

The factor *U* varies materially with various manufacturers and depends on temperature difference between refrigerant and water and water velocity. Conservative practice has established 150 ft per minute as water velocity. If refrigerant temperature is established at 35° F the water may be cooled to 40° F without danger of freezing. In a water-cooled remote unit system an economical temperature difference between inlet and outlet water is 15° F. In Table 38, Chapter IV, the logarithmic mean temperature difference is 10.8° F between refrigerant and water.

In Table 63 the over-all coefficient of transmission U for a shell water cooler, fin tube, flooded, is given as 135 Btu. However, the design of heat transfer surfaces is highly specialized, and manufacturers in the industry guarantee to meet any practical condition.

98. Heat Transmission Factors in Cooling

In the cooling of water or other substances in air conditioning processes, refrigerants are adjusted to an economical rise of 15° F. Table 63 gives examples of general practice in this field. The over-all factor U is given in Btu per hour per square foot per degree difference through the pipe surface, from fluid to fluid in the case of liquid coolers, and from fluid to air in the case of air coolers.

The heat transmission factor U of fin coils is given for purposes of comparison only and is of only academic interest to the designer, since fin coils are fabricated in standard sizes and rated on Btu transfer per square inch or square foot of frontal area instead of square feet of surface.

Fin coils occupy much less space than bare pipe coils for the same heat transfer capacity; in other words, bare pipe coils are rated on a square foot basis whereas fin coils may be rated on a *cubic foot basis.*

TABLE 63

OVER-ALL COEFFICIENT OF HEAT TRANSMISSION *U*

(Btu Per Hour Per Square Foot Per Degree Mean Temperature Difference) Operating Practice 10 to 15 Degrees Difference

Apparatus	Velocity of Fluid fpm	*U* Factor Btu
Condenser (double pipe)	100	150
Condenser (double pipe)	200	200
Condenser (double pipe)	300	250
Condenser (atmospheric)	Gravity	60
Condenser submerged coil	—	30–35
Condenser shell and water tube (flooded)	150	75–100
Shell water cooler, fin tube, flooded	500	135
Direct expansion pipe coil—baudelot	Gravity	40–60
Direct expansion pipe coil in brine tank	Still	10
Direct expansion pipe coil ice tank (rapid)	Circulation	15
Direct expansion double pipe brine cooler	100	80
Direct expansion double pipe brine cooler	200	130
Direct expansion double pipe brine cooler	400	175
Direct expansion double pipe brine cooler	800	220
Direct expansion shell and tube brine cooler	100	90–100
Direct expansion refrigerator coil—natural circulation	180 Air	2.5
Direct expansion refrigerator coil—forced draft (above 32° F)	500 Air	8–12
Direct expansion baudelot—oil chilling	Gravity	10
Direct expansion forced draft (air velocity)	100	3
Direct expansion forced draft (air velocity)	500	8
Brine coil—water—baudelot effect	Gravity	60
Brine coil—milk—baudelot effect	Gravity	70–75
Brine coil—cream—baudelot effect	Gravity	60
Brine shell and tube cooler	100	100
Brine coils in still air		3.5
Brine coils in air blast (above 32° F)	500	4.5
Finned brine coils in still air		4

Note: One inch frost will decrease *U* 25%. Area based on external pipe surface.

99. Refrigeration Pipe and Piping

Fluids used for refrigeration must be recirculated and conserved. Pipe coils are used for heat transfer purposes. Other piping must be used between the parts of the equipment to connect them and complete the system. Here the piping acts as a conduit to conduct the fluids in their cycles of operation. Piping and tubing for refrigerating purposes are made of wrought iron, steel, brass, copper, etc. in accordance with manufacturers' standard dimensions. Iron or steel pipe exclusively is used for ammonia to avoid corrosion since copper and brass are corroded by ammonia. Copper or brass tubing is generally used to conduct the majority of other refrigerants. To avoid the leaks in fittings which are caused by expansion and contraction due to changes in temperature, suitable provision must be made in the layout by proper location of swing joints, bends, anchors and offsets. Bent iron pipe coils, generally recommended for spray systems are made in pipe-bending shops to meet any design dimensions required.

The piping in refrigeration and air-conditioning units is commonly welded, although welding is a comparatively recent addition to the technique used in assembling a system.

TABLE 64

STEEL PIPE BENDS, HAIRPINS AND COILS—
LIMITING DIMENSIONS

Figures are distance on centers across U-bend, inches

Nominal pipe size *	¼		⅜		½		¾		1		1¼		2	
(Standard or extra heavy)	S	X	S	X	S	X	S	X	S	X	S	X	S	X
Usual diameter, inches	2½	2	3	2½	3	2½	3½	3	4	3	5	4	12	8
Limiting diameter, inches	2	1¾	2½	2¼	2½	2¼	3	2¾	3½	2¾	5	4	6	6

Nominal pipe size †	2½	3	3½	4	4½	5	6	7	8
Usual diameter, inches	20	32	40	44	48	54	60	72	86
Limiting diameter, inches	10	16	20	24	28	32	40	50	60

* Usually cold rolled; sometimes hot or requiring filling with sand.
† Usually hot rolled; Bessemer steel used mainly.

TABLE 65

MAXIMUM LENGTH OF ONE CONTINUOUS COIL, FEET

(Direct expansion NH_3 at velocity of gas 1000 ft per min; based on experience)

Evaporation Temperature °F / Pipe, Inches	—25	—15	—5	5	10	25
½	220	290	380	490	610	760
1	440	580	760	980	1220	1520
1½	740	990	1200	1670	2070	2560
2	970	1300	1710	2200	2770	3420

TABLE 66

SUITABLE FLUID VELOCITIES IN PIPING, FEET PER MINUTE

Fluid	Suction Side	Discharge Side
Ammonia	3000–4000	4000–5000
Carbon dioxide	1000–1200	1000–1200
Methyl chloride, Freon	3000–4000*	4000–5000*
	1000–2000†	2000–2500†
Sulfur dioxide	4000–5000*	5000–6000*
	1000–2000†	2000–2500†
Steam (saturated)	———	4000–5000
Water	60– 200	100– 250
Brine	60– 150	100– 200
Aqua Ammonia	30– 50	100– 250

* In large pipe sizes (manufacturing plants).
† In small tubing sizes (household refrigerators).
Tables 64, 65 and 66 by courtesy of A.S.R.E.

TABLE 67

STANDARD WROUGHT PIPE
(Welded Wrought Iron and Steel, and Seamless Steel)

Nominal Inches	Diameter External Inches	Internal Inches	Thickness Inches	Transverse Internal Area, Square Inches	Length of Pipe Per Square Foot of External Surface, Feet	Length of Pipe Per Square Foot of Internal Surface, Feet	Weight Per Foot Length, Pounds	Length of Pipe Containing 1 Cubic Foot	Number Threads Per Inch
1/8	0.405	0.269	0.068	0.057	9.431	14.15	0.25	2513.0	27
1/4	0.540	0.364	0.088	0.104	7.073	10.49	0.43	1383.3	18
3/8	0.675	0.493	0.091	0.191	5.658	7.73	0.57	751.2	18
1/2	0.840	0.622	0.109	0.304	4.547	6.13	0.85	472.4	14
3/4	1.050	0.824	0.113	0.533	3.637	4.635	1.13	270.0	14
1	1.315	1.049	0.133	0.864	2.904	3.645	1.68	166.9	11½
1¼	1.660	1.380	0.140	1.495	2.301	2.768	2.28	96.25	11½
1½	1.900	1.610	0.145	2.036	2.010	2.371	2.73	70.66	11½
2	2.375	2.067	0.154	3.355	1.608	1.848	3.68	42.91	11½
2½	2.875	2.469	0.203	4.788	1.328	1.547	5.82	30.10	8
3	3.500	3.068	0.216	7.393	1.091	1.245	7.62	19.50	8
3½	4.000	3.548	0.226	9.886	0.954	1.077	9.20	14.57	8
4	4.500	4.026	0.237	12.730	0.848	0.949	10.89	11.31	8
4½	5.000	4.506	0.247	15.947	0.763	0.848	12.64	9.02	8
5	5.563	5.047	0.258	20.006	0.686	0.757	14.81	7.20	8
6	6.625	6.065	0.280	28.891	0.576	0.630	19.19	4.98	8
*7	7.625	7.023	0.301	38.738	0.500	0.543	23.77	3.72	8
*8	8.625	8.071	0.277	51.161	0.442	0.473	25.00	2.81	8
8	8.625	7.981	0.322	50.027	0.442	0.478	28.81	2.88	8
*9	9.625	8.941	0.342	62.786	0.396	0.427	34.19	2.29	8

STANDARD WROUGHT PIPE (*continued*)

Nominal Inches	Diameter External Inches	Internal Inches	Thickness Inches	Transverse Internal Area, Square Inches	Length of Pipe Per Square Foot of External Surface, Feet	Length of Pipe Per Square Foot of Internal Surface, Feet	Weight Per Foot Length, Pounds	Length of Pipe Containing 1 Cubic Foot	Number Threads Per Inch
*10	10.750	10.192	0.279	81.585	0.355	0.374	32.00	1.76	8
*10	10.750	10.136	0.307	80.691	0.355	0.376	35.00	1.78	8
10	10.750	10.020	0.365	78.855	0.355	0.381	41.13	1.82	8
*11	11.750	11.000	0.375	95.033	0.325	0.347	46.25	1.51	8
*12	12.750	12.090	0.380	114.800	0.299	0.315	45.00	1.25	8
12	12.750	12.000	0.375	113.097	0.299	0.318	50.71	1.27	8

* Not included in U. S. Dept. of Commerce Simplified Practice Recommendations; all weights subject to 5% variation: mill test pressures, butt welded up to 2 in., 700 lb per square inch; lap welded 1¼ to 9 in., 1000; 10–12 in., 600 to 800.

TABLE 68

EXTRA STRONG PIPE

(Welded Wrought Iron and Steel and Seamless Steel)

Nominal Inches	Diameter External Inches	Internal Inches	Thickness Inches	Transverse Internal Area Square Inches	Length of Pipe Per Square Foot of External Surface, Feet	Internal Surface, Feet	Weight Per Foot Length Pounds	Number Threads Per Inch
1/8	0.405	0.215	0.095	0.036	9.431	18.632	0.31	27
1/4	0.540	0.302	0.119	0.072	7.073	12.986	0.54	18
3/8	0.675	0.423	0.126	0.141	5.658	9.070	0.74	18
1/2	0.840	0.546	0.147	0.234	4.547	7.046	1.09	14
3/4	1.050	0.742	0.154	0.433	3.637	5.109	1.47	14
1	1.315	0.957	0.179	0.719	2.904	4.016	2.17	11½
1¼	1.660	1.278	0.191	1.283	2.301	3.003	3.00	11½
1½	1.900	1.500	0.200	1.767	2.010	2.556	3.63	11½
2	2.375	1.939	0.218	2.953	1.608	1.975	5.02	11½
2½	2.875	2.323	0.276	4.238	1.328	1.649	7.66	8
3	3.500	2.900	0.300	6.605	1.091	1.328	10.25	8
3½	4.000	3.364	0.318	8.888	0.954	1.137	12.51	8
4	4.500	3.826	0.337	11.497	0.848	1.000	14.98	8
*4½	5.000	4.290	0.355	14.455	0.763	0.890	17.61	8
5	5.563	4.815	0.375	18.194	0.686	0.793	20.78	8
6	6.625	5.761	0.432	26.067	0.576	0.664	28.57	8
*7	7.625	6.625	0.500	34.472	0.500	0.576	38.05	8
8	8.625	7.625	0.500	45.663	0.442	0.495	43.39	8
*9	9.625	8.625	0.500	58.426	0.396	0.442	48.73	8
10	10.750	9.750	0.500	74.662	0.355	0.392	54.74	8
*11	11.750	10.750	0.500	90.763	0.325	0.355	60.08	8
12	12.750	11.750	0.500	108.434	0.299	0.326	65.42	8

* Not included in Simplified Practice Recommendations; weights subject to variations of 5%. Mill test pressures: up to 1 in., butt welded, 700; 1 to 3 in., butt welded, 1500; lap welded, 2500; above 3 in., lap welded, 2000 to 1000.

TABLE 69

DOUBLE EXTRA STRONG PIPE

Nominal Inches	Diameter External Inches	Internal Inches	Thickness Inches	Transverse Internal Area Square Inches	Length of Pipe Per Square Foot of External Surface, Feet	Length of Pipe Per Square Foot of Internal Surface, Feet	Weight Per Foot Length Pounds	Number Threads Per Inch
½	0.840	0.252	0.294	0.050	4.547	15.667	1.71	14
¾	1.050	0.434	0.308	0.148	3.637	9.049	2.44	14
1	1.315	0.599	0.358	0.282	2.904	6.508	3.66	11½
1¼	1.660	0.896	0.382	0.630	2.304	4.317	5.21	11½
1½	1.900	1.100	0.400	0.950	2.010	3.511	6.41	11½
2	2.375	1.503	0.436	1.774	1.608	2.561	9.03	11½
2½	2.875	1.771	0.552	2.464	1.328	2.176	13.70	8
3	3.500	2.300	0.600	4.155	1.091	1.672	18.58	8
3½	4.000	2.728	0.636	5.845	0.955	1.406	22.85	8
4	4.500	3.152	0.674	7.803	0.849	1.217	27.54	8
*4½	5.000	3.580	0.710	10.066	0.763	1.066	32.53	8
5	5.563	4.063	0.750	12.966	0.687	0.940	38.55	8
6	6.625	4.897	0.864	18.835	0.577	0.784	53.16	8
*7	7.625	5.875	0.875	27.109	0.500	0.650	63.08	8
8	8.625	6.875	0.875	37.122	0.437	0.565	72.42	8

* Mill test pressures: up to 1 in., butt welded, 700; 1¼ to 3 in., butt welded, 2200; 1½ to 3 in., lap welded, 3000; above 3 in., 2800 to 2000.

TABLE 70

WEIGHTS AND DIMENSIONS OF COPPER AND BRASS PIPE AND TUBES

	Copper Tubing							Copper and Brass Pipe—Regular Weight				
		Type K		Type L		Type M				Weight per Foot, Pound		
Nominal Size Inches	Outside Diameter Inches	Inside Diameter Inches	Weight per Foot Pound	Inside Diameter Inches	Weight per Foot Pound	Inside Diameter Inches	Weight per Foot Pound	Outside Diameter Inches	Inside Diameter Inches	67% Cop-per	85% Cop-per	100% Cop-per
1/8	0.250	0.186	0.085	0.200	0.068	0.20	0.068	0.405	0.281	0.246	0.253	0.259
1/4	0.375	0.311	0.134	0.315	0.126	0.325	0.106	0.540	0.375	0.437	0.450	0.460
3/8	0.500	0.402	0.269	0.430	0.198	0.450	0.144	0.675	0.494	0.612	0.630	0.643
1/2	0.625	0.527	0.344	0.545	0.284	0.569	0.203	0.840	0.625	0.911	0.938	0.957
5/8	0.750	0.652	0.418	0.666	0.362	0.690	0.263					
3/4	0.875	0.745	0.641	0.785	0.454	0.811	0.328	1.050	0.822	1.24	1.27	1.30
1	1.125	0.995	0.839	1.025	0.653	1.055	0.464	1.315	1.062	1.74	1.79	1.83
1 1/4	1.375	1.245	1.04	1.265	0.882	1.291	0.681	1.660	1.368	2.56	2.63	2.69
1 1/2	1.625	1.481	1.36	1.505	1.14	1.571	0.940	1.900	1.600	3.04	3.13	3.20
2	2.125	1.959	2.06	1.985	1.75	2.009	1.46	2.375	2.062	4.02	4.14	4.23
2 1/2	2.625	2.435	2.92	2.465	2.48	2.495	2.03	2.875	2.500	5.83	6.00	6.14
3	3.125	2.907	4.00	2.945	3.33	2.981	2.68	3.500	3.062	8.31	8.56	8.75
3 1/2	3.625	3.385	5.12	3.425	4.29	3.459	3.58	4.000	3.500	10.85	11.17	11.41
4	4.125	3.857	6.51	3.905	5.38	3.935	4.66	4.500	4.000	12.29	12.66	12.94
4 1/2								5.000	4.500	13.74	14.15	14.46
5	5.125	4.805	9.67	4.875	7.61	4.907	6.66	5.563	5.063	15.40	15.85	16.21
6	6.125	5.741	13.87	5.845	10.20	5.881	8.91	6.625	6.125	18.44	18.99	19.41
7								7.625	7.062	23.92	24.63	25.17
8	8.125	7.583	25.90	7.725	19.29	7.785	16.46	8.625	8.000	30.05	30.95	31.63

The National Bureau of Standards has recommended the elimination of the 3½" and 4½" pipe sizes.

100. Surface Cooling with Fin Coils

Fin coil surface area is dependent upon the following factors:

Coil

Surface-ratio:	Ratio of air-side surface area to refrigerant-side surface area.
Type of fins:	Round, square or continuous.
Shape of fins:	Plain or crimped, ribbon or wedge-shape.
Fin bond:	Integral, dipped, expanded, crimped, etc.
Material:	Copper, aluminum, steel, cast-iron, brass.
Depth and piping:	Depth of fins, number of tube-rows, tube spacing, counterflow, parallel flow, cross flow, mixed flow, etc.

Air

Velocity:	Mass velocity through free area, or linear velocity through face area.
Temperatures:	Dry-bulb, wet-bulb and dew-point.
Turbulence:	Reynolds number, local eddies, stratification and distribution.

Refrigerant

Type:	Liquid or direct-expansion vapor.
Velocity:	Velocity of liquid, gas or vapor in tube.
Turbulence:	Produced by velocity or by mechanical means, such as spirals or corrugated walls.

It is recommended that manufacturers' catalogs be consulted for rapid and accurate selection of fin coil surface capacities. Manufacturers' standard ratings for surface coolers are usually based on air volume for saturated air measured at a temperature of 70° F.

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% of the total heat.

101. Selection of Direct Expansion Coils

In the selection of direct expansion coils the following notes should be considered.

1. A direct expansion coil is selected to handle a given refrigeration load, cooling air to specifications. Consideration should be given to the capacity of the compressor and condenser.
2. Care should be exercised when refrigerant temperatures approach 32° F in order to avoid frost. One may either select a coil to operate at a higher back pressure or install a suction pressure regulator.
3. Direct expansion coils should be designed to drain, (what is known as "top feed") to assure return of oil; therefore the coil is used in a vertical position.
4. Counter-current flow is advised.

102. Rating Tables of Direct Expansion Coils

The catalogs of representative manufacturers are usually based upon the assumption that the refrigerant vapor is super-

TABLE 71

TONS OF REFRIGERATION PER SQUARE FOOT OF FACE AREA

(One Ton Equals 200 Btu per Minute 40° F Refrigerant—500 fpm Air Velocity)

		Initial Temperature Dry-bulb 80° F		
Initial Temperature Wet-bulb ° F	Rows of Tubes	Final Temperature Dry-bulb ° F	Final Temperature Wet-bulb ° F	Tons
67	2	67.1	60.8	0.83
67	3	62.6	58.2	1.16
67	4	58.6	55.8	1.43
67	5	55.3	53.6	1.68
67	6	53.0	51.7	1.88
67	8	49.1	48.6	2.19

heated 5° F (above saturation) when it leaves the coil. Design conditions recommended in New York City for cooling, are 75° F effective temperature for an occupancy of over 40 minutes; this corresponds to room conditions of 80° F dry-bulb, 50% relative humidity, 67° F wet-bulb and 60° F dew point, when outside air is 95° F dry-bulb, 75° F wet-bulb. To maintain the proper condition inside, air should be cooled to 57° F wet-bulb. The following table gives 4 rows of tubes to fulfill these conditions.

103. Dry Surface in Direct Expansion Coils

Assume initial conditions of 90° F dry-bulb, 65° F wet-bulb, 40° F refrigerant and 500 fpm face velocity. For the initial conditions stated above the following table shows that the first two rows will be dry and the others wet.

TABLE 72

TONS PER SQUARE FOOT FACE AREA

(One Ton Equals 200 Btu per Minute 40° F Refrigerant—500 fpm Air Velocity)

Initial Temperature Wet-bulb °F	Rows of Tubes	Initial Dry-Bulb Temperature 90° F Temperature Dry-bulb °F	Temperature Wet-bulb °F	Tons
65	2	dry 72	dry 58.7	dry 0.81
65	3	65.6	56.2	1.10
65	4	60.5	54.1	1.34
65	5	56.7	52.1	1.56
65	6	43.5	50.4	1.73
65	8	49.1	47.6	2.02

Catalogs give hundreds of ratings for various conditions similar to those above.

Table 73

RATIO OF FINAL TO INITIAL WET-BULB DEPRESSION FOR DIRECT EXPANSION COILS

Rows of Wet Tubes	Air Velocity—Feet Per Minute 300	400	500	600	700	800	900	1000	1200
2	0.440	0.472	0.497	0.512	0.527	0.543	0.554	0.566	0.583
3	0.292	0.323	0.346	0.368	0.387	0.399	0.411	0.423	0.445
4	0.194	0.221	0.244	0.264	0.281	0.295	0.307	0.320	0.340
5	0.129	0.153	0.172	0.188	0.204	0.217	0.228	0.239	0.259
6	0.085	0.104	0.121	0.135	0.148	0.160	0.170	0.179	0.198
7	0.057	0.071	0.085	0.097	0.108	0.118	0.126	0.135	0.151
8	0.038	0.049	0.060	0.069	0.078	0.086	0.094	0.101	0.114
10	0.016	0.023	0.030	0.036	0.042	0.047	0.052	0.057	0.067
12	0.007	0.011	0.015	0.018	0.022	0.025	0.029	0.032	0.039
14	0.003	0.005	0.007	0.009	0.012	0.014	0.016	0.018	0.023

From Trane Co. Catalog.

104. Dry Surface

No cooling coil can dehumidify air unless the temperature of its fins is lower than the dew point temperature of the entering air, i.e., the initial dew point temperature. The fin temperature is always at some point between the temperature of the air flowing past the fin and the temperature of the refrigerant inside the tubes. If the refrigerant is at constant temperature, the colder the air, the colder will be the fins. Therefore as the air travelling through the coil is cooled, the temperature of the fins progressively drops from the inlet face of the coil to its outlet face. The warmest fins are at the inlet face of the coil and the coldest ones at the outlet face. When the fin temperature has fallen to a point that is slightly below the dew point temperature of the entering air, dehumidification of the air will begin.

When the dry-bulb temperature of the air is high and the dew point temperature comparatively low, a considerable part of the coil is likely to be dry; only a small part of the coil will be cold enough to dehumidify the air. On the other hand, when the dry-bulb temperature of the entering air is low, there may be no dry surface whatever on the coil. The temperature of the fins at the inlet face of the coil may be, and often is, below the initial dew point temperature. In such a case, dehumidification of the air will begin at the instant the air strikes the coil, and there will be no dry surface whatever.

105. Determination of the Required Average Effective External Coil Surface Temperature

If a coil has an efficiency of 0.8 (80%) for the removal of sensible heat, it will, at the same time, remove 80% of the difference in moisture content between the entering air and saturated air at the surface temperature. This is due to the fact that 80% of the air particles contact the surface and attain a dew point temperature equal to the surface temperature.

Problem. Determine the required average effective external coil surface temperature.
Given:

(1) Air enters coil at 83° F dry-bulb and 69° F wet-bulb. Find the average external coil surface, if sensible heat pick up factor is 65%.

Solution.

Use the apparatus dew point empirical formula given in Chapter IX, Figure 12A.

$$t_{sz} = \frac{F_d(h_1 + 11.2) - 0.24t_1}{0.625F_d - 0.24}$$

where

t_{sz} = dew point

0.65 = F_d = sensible heat pick-up factor which is the ratio of the difference between sensible heat entering and leaving divided by the difference between the total heat entering and leaving

$33.12 = h_1 =$ total heat, Btu per pound air entering at wet-bulb temperature 69° F

$83 = t_1 =$ dry-bulb temperature of air entering

$$t_{sz} = \frac{0.65(33.12 + 11.2) - 0.24 \times 83}{0.625 \times 0.65 - 0.24} = 54° \text{ F approx.}$$

To obtain the exact dew point after obtaining the above trial value, compile the following table noting that h = sensible heat entering at dry-bulb t_1

Entering	Leaving	Difference	Condition
33.12	22.55	10.57	total heat
19.91 = h	12.95	6.96	sensible heat

$$\frac{6.96}{10.57} = 0.658$$

But 65% is the ratio desired

$0.65 \times 10.57 = 6.87$ difference required

$19.91 - 6.87 = 13.04$ which is true sensible heat and from table corresponds to 54.5° F dew point.

106. Year-round Air Conditioning of Hotel Rooms and Offices

Circulating cooled water as an efficient secondary refrigeration medium is recognized by progressive engineers in the design of the comfort cooling of individual hotel rooms, apartments and offices. The newest hotel constructed in Washington, D. C. supplies its guests with cold water air conditioning in summer and hot water air conditioning in winter, using the same coil and circulating piping, in addition to a positive supply of fresh, filtered, tempered air. Specifications and plans for many projects of this type of air conditioning are already prepared, and preliminary proposals of cost have been submitted on building projects being considered as far in the future as 1950. The expected postwar building boom will be shared by progressive air-conditioning contractors, who specialize in large building projects. See Figure 85, Chapter XIX.

107. Water Cooling-coil Ratings

Heat from the passing warm air is transmitted through the fins and tubes to the circulating water. Heat transmission depends upon several factors for any given type of surface. These factors are

(1) Air inlet dry-bulb temperature, ° F
(2) Air inlet relative humidity
(3) Air velocity, feet per minute
(4) Water inlet temperature, ° F
(5) Water velocity, feet per second
(6) Number of rows of tubing in direction of air stream

Consideration must be given to the relative proportion of latent and sensible heat for any combination of the above factors. The rating table in the catalog of a representative manufacturer covers 30,000 different capacities. Any of the above variables may be in combination with any of the others, and, as a result a large number of capacity variations is possible.

Latent heat transfer results when moisture in the air condenses on the finned surface because the air in close proximity to the finned surface reaches the dew point. Air at the dew point is saturated with moisture and this saturation occurs near the finned surface because the air cools off more rapidly there than at a more distant portion of the air stream. In close proximity to the finned surface the dry-bulb, wet-bulb and dew point temperatures become equal. Dehumidification occurs when the air in close proximity to the cold, finned surface reaches its dew point, therefore it is not necessary that the average air temperature at the exit of the coil drop that low. The principle of counter-current flow is utilized; the warmest air meets the warmest water and the coldest air meets the coldest water.

The fabrication of finned coils has become a major industry. Standard units are supplied in sizes from one row up to eighteen rows to meet the requirements. It is beyond the scope of this book to give capacity tables, dimension sheets, and other engineering data that apply to coils fabricated by representative manufacturers. Upon application, however, these manufacturers courteously supply engineers with very elaborate and authentic data referring to the subject.

108. Selection of Water Cooling Coils

In modern design, satisfactory installations incorporate the following general features:

1. Counter-current flow of water and air through the coil bank is used in order to obtain the maximum efficiency. Parallel flow is opposed to counter-current flow.

2. Excess coil surface area is avoided by allowing a reasonable difference of 6 to 8° F between the temperature of the entering water and the dry-bulb temperature of the leaving air.

3. Face velocity of air through a coil bank is limited to a range of 400 to 600 ft per minute; allow 400 ft with a long run of air ducts in order to hold air friction to a minimum. Eliminate the carry-over of moisture from fins by limiting the air velocity to a maximum of 600 fpm.

4. A narrow and long coil for a given face area is less expensive than a wide short coil. For instance a 12 in. × 60 in. coil (12 in. face width and 60 in. face length) has the same face area of 5 sq ft as a 24 in. × 30 in. coil but the 12 in. × 60 in. coil will be less expensive.

5. Maintain as high a water velocity as possible to obtain maximum heat transfer and at the same time maintain a reasonable limit to water friction losses. When water velocity falls below 4 fps, spiral baffles may be inserted inside the tubing to create additional turbulence. When velocity exceeds 4 fps baffles are not required. Where large quantities of water and a low water friction are specified use a specially designed coil,

giving a smaller number of passes. Maximum velocities are within range of 6 fps. The following data are offered as an average of modern practice utilizing ⅝ in. or ¾ in. outer diameter tubes.

109. Water-coil Selection Calculations

To convert cubic feet of air per minute at standard conditions to pounds of air per hour

$$\text{cfm} \times 4.5 = \text{pounds per hour} \tag{3}$$

To convert gallons of water per minute to pounds per hour

$$\text{gpm} \times 500 = \text{pounds per hour} \tag{4}$$

Problem. Given the following, select a water coil

(a) Air to be cooled, 5000 cfm
(b) Entering air dry-bulb 83° F
(c) Entering air wet-bulb 69° F
(d) Leaving air dry-bulb 61° F
(e) Leaving air wet-bulb 59° F
(f) Temperature of cooling water 50° F
(g) Quantity of water available 34 gpm

Solution. Operation 1

The total heat load is the difference in enthalpy between the entering and leaving wet-bulb temperatures, expressed in Btu per pound of air multiplied by the total air poundage circulated. (Refer to psychrometric table.)

Enthalpy at 69° F wet-bulb — 33.12 Btu per pound dry air
Enthalpy at 59° F wet-bulb — 25.70 Btu per pound dry air
7.42 Btu per pound dry air

Pounds of air per hour

$$5{,}000 \times 4.5 = 22{,}500 \text{ lb per hour}$$

$$\text{Total heat load} = 22{,}500 \times 7.42 = 167{,}050 \text{ Btu per hour}$$

Operation 2

Temperature rise (T_R) of water through coil

$$500 \times \text{gpm} = \text{pounds water per minute}$$
$$500 \times 34 = 17{,}000$$

$$T_R = \frac{167{,}050}{17{,}000} = 10^\circ \text{ F (approximately)}$$

The leaving water temperature is $50^\circ F + 10^\circ F = 60^\circ F$

Operation 3

Coil face area required

$$FA = \frac{cfm}{FV}$$

FA = face area
cfm = cubic feet air per minute
FV = face velocity
(allow 500 fpm as a reasonable face velocity)

$$\text{Face area} = \frac{5000}{500} = 10 \text{ sq ft}$$

Select a section from manufacturer's catalog of 10 sq ft face area

Operation 4

Water velocity through coil

The velocity varies from 2 to 6 fps depending upon face width of coil selected; this determines the length of header. In this problem, allow 3 fps.

Operation 5

Allowance for increase in heat transfer caused by moistening of surface

This factor is determined from the difference between the dry-bulb temperature of the entering air and the initial water temperature and the difference between the dew point temperature of the entering air and the initial water temperature.

In this problem the temperatures assumed are

83° F dry-bulb	62° F dew point
50° F entering water	50° F entering water
33° F difference	12° F difference

refer to Table 74 for approximate factor of 1.15

TABLE 74

INCREASE IN HEAT TRANSFER CAUSED BY MOISTENING OF SURFACE

Difference, ° F Between Dew Point of Entering Air and Temperature of Entering Water	10	20	30	40
Difference Between Dry-bulb Temperature of Entering Air and Temperature of Entering Water	Increase in Heat Transfer			
20	1.25	1.7		
30	1.13	1.44		
40	1.08	1.29	1.65	
50	1.05	1.2	1.45	
60	1.02	1.12	1.35	1.63

Operation 6

Mean temperature difference

The MTD can be found by referring to Figure 4 Case C and Table 38, Chapter IV. It amounts to 15.45° F.

Operation 7

To find heat transfer of water cooling coils in Btu per hour per degree mean difference per square foot of face area per row of tubes.

TABLE 75

BTU PER SQUARE FOOT OF FACE AREA PER HOUR PER DEGREE MEAN TEMPERATURE DIFFERENCE PER ROW OF TUBES

Water Velocity Feet Per Second	Air Velocity Feet Per Minute 300	500	700	1000
1.5	110	135	155	170
2.00	115	145	170	185
3.00	125	160	185	210
4.00	130	170	200	230
5.00	135	180	210	245

Operation 8

Finding number of rows of tubes required

Total load	= 167,050 Btu per hour—	operation 1
Coil face area	= 10 sq ft	operation 3
Mean temperature difference	= 15.45° F	operation 6
Heat transfer factor	= 160 Btu per minute	operation 7
Increase due to wet surface	= 1.15	operation 5

$$\text{Rows of tubes} = \frac{\text{Total load in Btu per hour (1)}}{(3) \times (6) \times (7) \times (5)}$$

$$= \frac{167{,}050}{10 \times 15.45 \times 160 \times 1.15} = \frac{28{,}428}{167{,}050} = 5.9$$

A six-row coil will be required in the design of a central air-conditioning system. It is also necessary to know the friction of air through fin coils, as given in Table 76.

TABLE 76

AIR FRICTION THROUGH FIN COILS IN INCHES, WATER GAGE; FIRST TWO COILS DRY, BALANCE WETTED SURFACE

(10° F difference between initial air dew point and initial water temperature)

Rows of	Air Velocity in Feet Per Minute					
Tubes	300	400	500	600	800	1000
1	0.028	0.047	0.07	0.097	0.163	0.240
2	0.024	0.085	0.125	0.175	0.291	0.440
3	0.081	0.135	0.302	0.281	0.469	0.70
4	0.105	0.178	0.292	0.369	0.61	0.924
6	0.156	0.264	0.393	0.545	0.91	1.36
8	0.206	0.349	0.521	0.721	1.20	1.80
10	0.257	0.435	0.649	0.897	1.50	2.24
12	0.308	0.521	0.77	1.073	1.79	2.68
14	0.358	0.602	0.91	1.25	2.09	3.12

TABLE 77

WATER FRICTION THROUGH ⅝ IN. COILS SINGLE PASS

(inches water gage)

Velocity Feet Per	Coil Face Length in Inches								
Second	12	24	36	48	60	72	84	96	120
1	0.13	0.18	0.23	0.28	0.33	0.38	0.43	0.48	0.58
2	0.42	0.58	0.74	0.90	1.1	1.2	1.4	1.5	1.9
3	0.86	1.2	1.5	1.8	2.1	2.4	2.7	3.0	3.6
4	1.4	1.9	2.4	2.9	3.4	3.9	4.4	4.8	5.8

TABLE 78

WATER VELOCITIES THROUGH COOLING COILS

Gallons Per Minute	Feet Per Minute	Face Width, Inches
40	6.5	12
30	4.8	12
20	3.3	12
60	6.6	18
50	5.4	18
40	4.3	18
30	3.3	18
20	2.2	18
75	6.0	24
65	5.2	24
55	4.5	24
45	3.5	24
35	2.75	24
25	2	24
100	6.5	30
80	5	30
70	4.5	30
55	3.5	30
40	2.5	30

CHAPTER VII

RADIANT HEATING

Heating 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 there is a feeling of comfort. In convection heating, heat is transferred from a conventional heating unit, such as a steam or hot water radiator, to the air and thence from the air to the occupant. The main object of radiant heating however, 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. On a cold windless day, one may feel perfectly comfortable standing in the sunshine, but, when a cloud passes over the sun, one will instantly feel colder. A shielded thermometer will show no immediate reduction in temperature, so that one actually feels a cooling effect which an ordinary thermometer cannot register.

110. Radiant Heat

Heat that is radiated from a body may be considered similar to light. Radiant heat is always transferred from a body of high temperature to a body of lower temperature. The amount of heat radiated will depend upon the temperature difference and the composition of the bodies involved. Most of the problems of energy transfer by thermal radiation may be classified as follows:

(a) Radiant heat exchange between the surfaces of solids separated by non-absorbing mediums.

(b) Radiation from flames and gases.

(c) Radiation from surfaces of small particles.

Thermal radiation from the surface of a solid is best expressed as a ratio with the radiation from a so-called *ideal radiator or a black body.* A black body absorbs all of the incident radiation and reflects, transmits and scatters none.

The quantity of radiation Q emitted by a black body is determined completely by its temperature according to the Stefan-Boltzman law.

$$Q = \sigma A T^4 \qquad (1)$$

where

A = area, square feet
T = absolute temperature
σ = Stefan-Boltzman constant

The constant σ has the value

$$0.173 \times 10^{-8} \text{ Btu/(square foot)(hour)}(^\circ\text{R})^4$$

The ratio of the total radiating power of a non-black surface to that of a black surface at the same temperature is called the *emissivity* of the surface. The emissivity of a surface varies with its temperature, its degree of roughness, and, if a metal, its degree of oxidation. Tables 79 and 80 give the emissivities of various surfaces.

Table 79

EMISSIVITY OF SURFACES
METALS AND THEIR OXIDES

Surface	Temperature °F	Emissivity
Aluminum		
Polished	73	0.040
Rough	78	0.055
Copper		
Polished	176	0.018
Oxide coat	77	0.78
Iron		
Galvanized bright	75	0.28
Galvanized gray	82	0.23
Aluminum paint	212	0.27 to 0.67

Polished metal surfaces have an emissivity 15 to 20% higher than the normal. The *absorptivity* of a surface, i.e., the fraction of impinging radiation directly absorbed depends on the same factors that affect emissivity. Kirchoff's law states that a surface of relatively low radiating power is also a poor absorber (or a good reflector) of radiation, from a source at its own temperature.

TABLE 80

EMISSIVITY OF SURFACES
BUILDING MATERIALS, PAINTS, ETC.

Surface	Temperature ° F	Emissivity
Asbestos	100–700	0.27 to 0.67
Black lacquer, shiny	70	0.87 to 0.91
Planed oak	70	0.87 to 0.91
White enamel paint	70	0.87 to 0.91
Gypsum	70	0.87 to 0.91
Roofing paper	70	0.87 to 0.91
Lime plaster	70	0.87 to 0.91
Glazed porcelain	70	0.92 to 0.96
Polished marble	70	0.92 to 0.96
Rough red brick	70	0.92 to 0.96
Smooth glass	70	0.92 to 0.96
Water	70	0.92 to 0.96

111. Radiant Heating

The conventional heating system, usually installed in the United States, is of the radiator or warm air type which transmits heat principally by convection. The function of any heating system is not to supply heat to occupants but to maintain thermal equilibrium between production and loss of human body heat. The human body maintains its heat at a temperature of 98.6° F and is so regulated that the loss approximately balances the rate at which heat is produced. However, the degree of activity and physical condition of the body complicates the problem of regulation.

The primary factor controlling a person's feeling of comfort is a skin temperature of 90 to 94° F. Heat is transferred from the moist human body by convection, by radiation and by evaporation from the body surface and the respiratory tract. The rate of heat loss by convection depends upon the average temperature difference between the surrounding air and the surface and size of the body and the rate of air motion over it. The rate of heat loss by radiation depends upon the exposed surface area of the body, and upon the difference between the mean surface temperature of the body and the mean surface temperature of the surrounding walls or other objects. To this latter temperature has been assigned the symbol MRT or mean radiant temperature. The total skin surface of the average human body may be assumed as 19.5 sq ft for convection and 15.5 sq ft for radiation. The loss by evaporation depends upon the temperature and area of the moist surfaces of the body, air temperature, air movement and humidity. In air at a tempera-

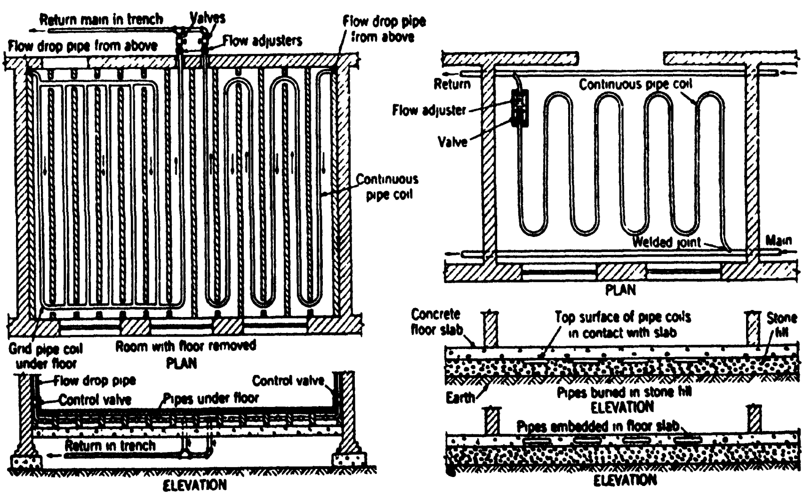


FIGURE 7
Radiant Heating Coils

Left, continuous coil in floor; right, coils embedded in floors.
Courtesy of A.S.H.V.E.

ture of 70° F, this loss for a person at rest will be approximately 90 Btu per hour. The loss by radiation and convection is approximately 300 to 310 Btu per hour. The proportion for greatest comfort seems to be about 190 Btu per hour by radiation and about 120 Btu per hour by convection. The surface temperature of the skin is 90 to 94° F; considering clothing and hair, a mean surface temperature of 80° F is assumed as a design factor.

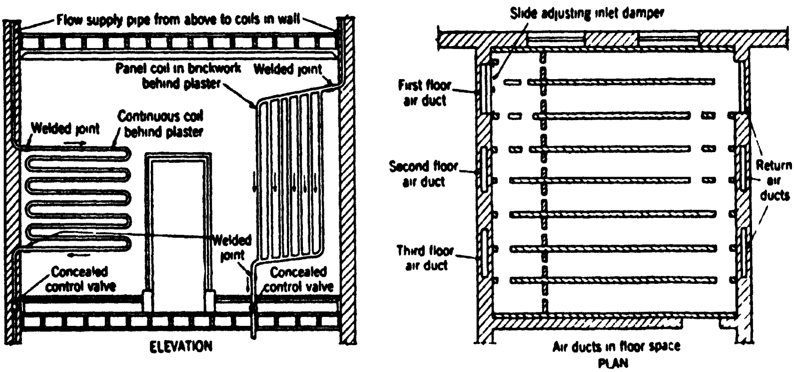


FIGURE 8
Radiant Heating
Left, coils in wall surfaces; right, air ducts for floor heating.
Courtesy of A.S.H.V.E.

112. Types of Radiant Heating Installations

Radiant heating may be used for heating a structure by several methods:

1. By warming the interior wall and ceiling surface of the building.
2. By circulating warm air through shallow ducts under the floor.
3. By placing hot water or steam pipes under the floor.
4. By attaching separate heated metal plates or panels to the interior surface.
5. By electrically heated metal plates or panels.

6. By electrically heated tapestry mounted on screens and on the wall.

113. Estimating the Radiant Heat Load

Calculations involved in radiant heating are entirely different from those for convection heating. Determination of heat loss from the building structure is the problem of convection heating but radiant heating involves regulation of heat loss from the human body. The order of operations suggested in estimating the heating load is:

1. Determine the desired MRT
2. Determine location of heating surfaces
3. Establish temperature of each heating surface
4. Compute area of each heating surface to produce the MRT
5. Calculate the actual heat loss from the room and provide additional convected heat to allow for air infiltration

114. Determination of MRT

The actual surface temperatures of a room vary with construction and exposure of the walls of the enclosure. If all the interior floor, wall and ceiling surfaces were at the same temperature, this would be the MRT. It is, therefore, necessary to estimate the MRT or mean interior surface temperatures, depending upon nature of occupancy. The following table is based on two classes of occupancy

(a) Schools, homes, buildings which are usually designed for an interior winter air temperature of 65 to 70° F dry-bulb

(b) Factories and foundries at 55 to 60° F dry-bulb

115. Inside Temperatures of Unheated Walls

Table 83 is based on inside conditions of 65° F air, 71° F MRT with still air inside and outside wind velocity of 15 mph. Over-all heat coefficient of transmission is U.

TABLE 81

MEAN RADIANT TEMPERATURE, ° F
(walls, floor and ceiling)

Class	Air Temperature, ° F	MRT
a	70	71
a	65	71
b	60	61
b	55	60

It will be noted that the design MRT may be a few degrees above air temperature.

TABLE 82

HEAT RADIATION TO SURROUNDINGS FROM HEATED PANEL
(e = 0.90)

Surface Temperature, ° F	Radiation, Btu per Square Foot per Hour
30	89.4
40	96.8
50	105.0
60	113.3
70	122.3
80	131.9
90	142.0
100	152.6

Note: Surroundings are assumed to be at absolute zero.

The factors in Table 82 are calculated from the formula

$$Q = \frac{0.1723 \times T^4}{1,000,000}$$

where

Q = total radiation Btu per square foot per hour
e = emissivity
T = absolute temperature ° R

TABLE 83

INSIDE SURFACE TEMPERATURE, ° F OF OUTSIDE VERTICAL UNHEATED WALLS

	Outside Air Temperature			
U	-20	0	20	40
0.2	55	56	58	60
0.4	45	49	54	59
0.6	35	42	49	55
1.0	17	26	35	45
2.0				27
3.0				8

TABLE 84

SAFE TEMPERATURES FOR HEATING PANELS

Type of Panel	Surface Temperature, ° F
Plastered ceiling (Pipes embedded)	70–110
Plastered walls (Pipes embedded)	75–115
Floor (Any method)	70–90
Iron (Hot water)	150–160
Iron (Steam)	160–180
Electrically heated panels	180–200

116. Calculation of Mean Radiant Temperature (MRT)

The mean radiant temperature of unheated interior surfaces is calculated by first determining the surface temperature of the individual interior surfaces. Multiply the area of each surface by the surface temperature and add the products. This sum is divided by the total surface to determine the mean surface temperature of the unheated surfaces. Assume partition walls to have surface temperature equal to air temperature when unheated.

117. Heat Release from Radiant Heat Panels

The heat release is determined by dividing the total heat loss of the room by the area of the ceiling, floor or panel which is to be heated. For example if the heat loss of the room is 5000 Btu per hour and the ceiling area is 200 sq ft the heat release of the ceiling will have to be 5000 ÷ 200 or 25 Btu per square foot per hour to meet the heating requirements.

TABLE 85

HEAT RELEASE BY RADIATION FROM PANELS, BTU PER HOUR
(Emissivity = 0.95)

Mean Radiant Temperature, °F	50	70	90
Surface Temperature °F	Btu Per Hour Released		
80	25	10	
100	45	30	10
120	68	50	30
140	90	75	53
160	118	100	80
180	146	130	110
200	178	160	140

TABLE 86

HEAT RELEASE BY CONVECTION FROM RADIANT HEAT PANELS

Surrounding air temperature ° F	50	70	90
Surface Temperature °F	Btu Per Hour Released		
80	26	10	
100	50	28	8
120	75	50	27
140	105	75	50
160	135	105	78
180	168	120	104
200	200	165	135

Note: This table is based on floor position. For vertical position multiply by 0.80. For ceiling position multiply by 0.48.

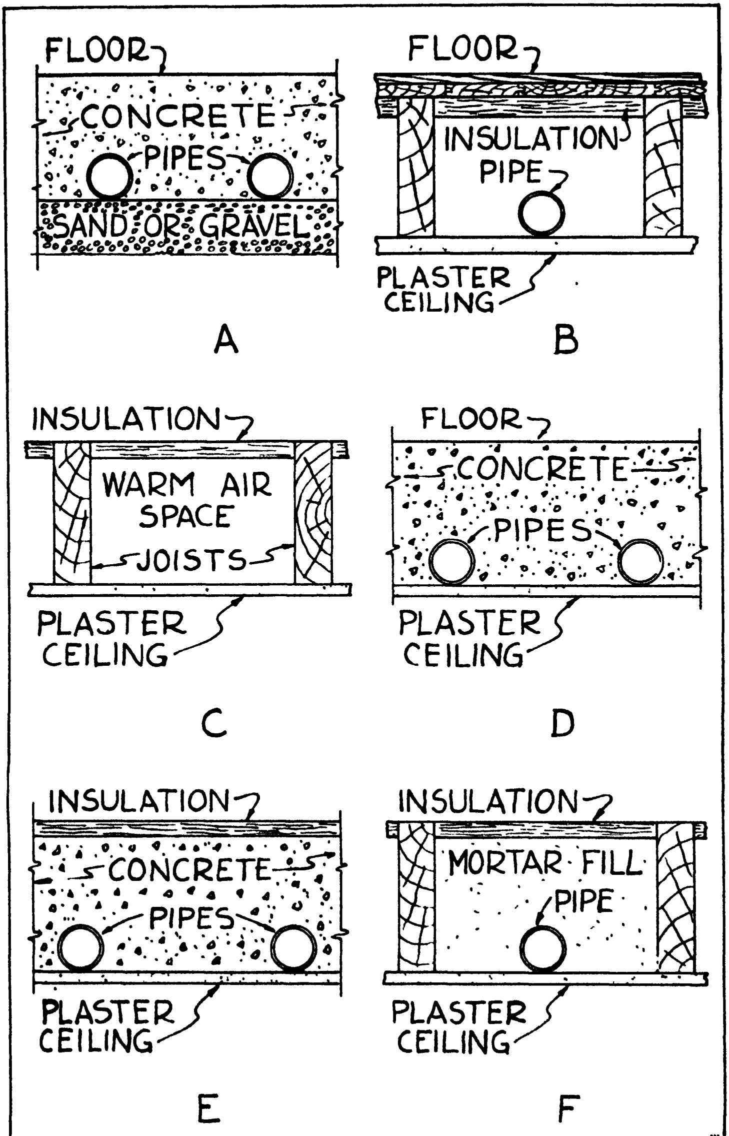


FIGURE 9
Radiant Heating

118. Design of Radiant Heating Surface

American practice is either to heat the floor with steam or hot water coils buried in it, or to heat ceiling panels with hot water or steam pipes buried in the plaster or mortar or located in the air space above the plaster. Warm air ducts in the space between the ceiling and the floor above are also used. Wall heating is not uncommon.

119. Pipe Coils in Concrete Floor (Figure 9-A)

Pipe from ¾ in. to 2 in. in diameter or from 6 in. to 4 ft centers is embedded in the concrete slab or in the gravel or sand base between concrete and ground. The steam or hot water coils heat the floor surface from 75° F to 85° F maximum.

where

t_c = temperature of concrete in coil plane
t_f = floor surface temperature
h_f = heat release of floor, Btu per square feet per hour
U_f = heat transmission coefficient between coils and floor

$$t_c = \frac{h_f}{U_f} + t_f$$

For concrete $U_f = 12$ Btu per square foot per degree difference per hour per inch thickness.
For determination of heat loss to ground beneath

$$h_g = U_g \ (t_c - t_g)$$

where

h_g = heat loss to ground in Btu per square foot per hour
U_g = transmission coefficient for moist earth
t_g = temperature of concrete slab at coil
t_c = temperature of ground.

U_g = about 10 Btu per square foot per degree difference per hour per inch thickness for moist earth. A ground temperature of 70° F may be expected 5 ft below the slab. Note that for 2 in. thickness the heat loss is $\frac{10}{2} = 5$ Btu; for 3 in. thickness $= \frac{10}{3}$ Btu.

120. Temperature of Steam or Hot Water

$$t_s = \frac{h_f}{U_f}\sqrt{d^2 + 0.058C^2} + t_f$$

where

t_s = temperature of steam or hot water
h_f = heat release of floor in Btu per square foot per hour per in. thickness per degree
d = thickness of floor above pipe coils
c = distance between pipes in inches

121. Quantity of Steam or Hot Water

A temperature drop of 10 to 20° F is allowed for hot water

where

h_f = total floor release [redacted]
h_g = total ground release [redacted]
t_i = water temperature inlet
t_o = water temperature outlet

$$\frac{h_f + h_g}{t_i - t_o} = \text{pounds hot water per hour}$$

For steam

$$\frac{h_f + h_g}{1000} = \text{pounds steam per hour}$$

1000 Btu = approximate latent heat of steam.

122. Pipes in Ceiling Air Space (Figure 9-B)

Iron or steel pipes exposed to air between ceiling joists with natural air movement transfer 2 Btu per square foot exterior surface per ° F difference per hour and the following table is based on this factor.

TABLE 87

HEAT RELEASE THROUGH IRON PIPES IN AIR

Temperature Difference Heating Medium and Air	Nominal Pipe Size, Inches					
	½	¾	1	1¼	1½	2
°F	Btu Per Hour Per Lineal Foot					
10	4	5	7	8	10	12
20	9	11	14	17	21	25
30	14	17	22	27	31	38
40	19	23	29	36	42	51
50	23	29	37	46	53	64
60	30	36	45	56	64	77

Figure 9–B shows insulation above pipes, as the intention is to avoid heating the floor above. The routine followed in designing the heat coil is:

1. Determine heat release and ceiling temperature as previously explained

2. Calculate temperature of air in joist space by solving equation

$$t_a = \frac{h_c}{U_c} + t_s$$

where

t_a = air temperature in ceiling space

h_c = required heat release of ceiling in Btu per square foot per hour

U_c = transmission coefficient of ceiling in Btu per square foot per degree difference per hour

t_s = ceiling surface temperature

3. Calculate the heat loss through top of air space as follows:

$$h_t = U_t\,(t_a - t)$$

TABLE 88

U_c FOR CEILINGS

Plaster board	0.80
Wood lath and plaster	1.00
Metal lath and plaster	1.2

where

h_t = heat loss through top of air space in Btu per square foot per hour

U_t = transmission coefficient in Btu per square foot per degree difference per hour

t_a = temperature of air in joist space

t = temperature of space above ceiling

Assume $U = 0.3$ for rigid board insulation.

4. The total heat release per square foot required is the sum of the ceiling heat release and the heat loss, or $h_c + h_t$.

If joists are on 12 in. centers this figure equals the heat release of pipe per lineal foot. If joists are on 16 in. centers multiply by 1.33. If joists are on 24 in. centers multiply by 2.

5. The temperature of hot water or steam is determined by referring to Table 87, and finding the pipe size and temperature difference. The heating medium temperature is the sum of air space temperature plus the temperature difference from Table 87.

6. The temperature of hot water and steam is determined as previously explained.

123. Pipes in Ceiling Space Room Above and Below Radiant-heated (Figure 9-B)

The method of determining pipe coil surface is practically the same as outlined in previous topics. However, determining heat loss through the top of the air space is not required. To provide the proper floor temperature the quantity of insulation in the floor must be determined as follows:

$$r_f = \frac{t_a - t_s}{h_f}$$

where

r_f = the resistivity of the floor
t_a = joist air temperature
t_s = floor surface temperature
h_f = heat release of floor

Resistivity is the reciprocal of U.

Assume approximately 1.95 as total resistivity of an ordinary floor. The difference between 1.95 and the calculated resistivity is the resistivity of the insulation to be added. Rigid insulating board has an approximate resistivity of 3 per inch of thickness and aluminum foil has an approximate resistivity of about 2.3.

124. Ceiling Radiant Heated by Warm Air (Figure 9-C)

General practice is to force warm air through ducts between the joists or through joist space, omitting ducts. When space above air space is not heated, insulation is required. The following calculations assume that space above is not heated but that the space below is heated. The fundamental problem is to find the volume flow of warm air per minute and the inlet and outlet temperature of the warm air. The routine to be followed in solving for these factors is given below:

1. The ceiling heat release and ceiling surface temperature are determined as previously explained.

2. The temperature of air in the joist space is found by solving the equation

$$t_a = \frac{h_c}{U_c} + t_s$$

where

t_a = air temperature in ceiling space

h_o = required heat release of ceiling in Btu per square foot per hour

U_o = transmission coefficient of ceiling in Btu per square feet per degree difference per hour

t_s = surface temperature

The heat transmission factor U_c for various types of building construction material is given in tables in Chapter V; however, approximate factors given previously may be used if the type of construction material is unknown at the time of making a preliminary estimate.

3. Calculate heat loss through top of insulated air space assuming $U_t = 0.3$, if type of construction is unknown.

$$h_t = U_t\ (t_a - t)$$

where

h_t = heat loss through top of air space in Btu per square foot per hour

U_t = coefficient in Btu per square foot per degree difference per hour (including film coefficient)

t_a = temperature of air in joist space

t = temperature space or material above floor.

4. To estimate volume of air

$$(h_c + h_t)\ A = \text{total heat release}$$

where

A = area, square feet

h_c = ceiling heat release in Btu per square foot per hour

h_t = floor heat release in Btu per square foot per hour

t_a = average temperature of air in space (previously calculated) assumed at exit from space

To avoid excessively large heaters and increased heat losses a temperature drop of 50° F is reasonable, therefore

$$\frac{(h_c + h_t)A \times 55}{50 \times 60} = \text{air flow, cubic feet per minute}$$

Note: One Btu will raise 1 cu ft of air 55° F; 1 cu ft of air at 70° F weighs approximately 0.075 lb, 13.34 cu ft weigh 1 lb.

Specific heat of air = 0.24
$0.075 \times 0.24 \times 55 = 1$ Btu (approximately)

125. Radiant Heating by Pipe Coils in Concrete Ceiling to Heat Only Room Below (Figure 9-D)

Pipe coils are embedded in concrete in the underside of the ceiling of the room to be heated and the surface is plastered over. Usual practice is to use ½ in. or ¾ in. copper, iron or steel pipe bent into coils at 4 to 6 in. centers. Routine calculations are given in order as previously explained.

1. Panel heat release and ceiling surface temperature is determined as demonstrated above.

2. Average temperature of concrete in plane of coils is determined by solving equation

$$t_p = \frac{h_c}{U} + t_s$$

where

t_p = temperature of concrete in plane of coils
t_s = ceiling surface temperature
h_c = heat release of ceiling in Btu per square foot per hour
U = coefficient, assumed as 12 Btu per square foot per degree difference per hour per inch thickness

3. The heat loss to the space above the coils is determined by solving the following equation

$$h_a = U\,(t_p - t)$$

where

h_a = heat loss, Btu per square foot per hour to space above
U = heat coefficient of space above the pipe plane to the pipe plane
t_p = temperature of concrete in pipe plane
t = temperature of air space above

In calculating U there is only one surface coefficient to be considered.

4. The average temperature of the heating medium in the pipe is determined by solving the following equation

$$t_m = 0.15\ h_c + t_s \text{ for pipe on 6 in. centers}$$
$$t_m = 0.12\ h_c + t_s \text{ for pipe on 4 in. centers}$$

where

t_m = average temperature of heating medium in pipe

t_s = average temperature of ceiling

h_c = heat release from ceiling, Btu per square foot per hour

5. Use the following formula to determine the quantity of hot water to be supplied each hour.

$$\frac{A(h_c + h_a)}{10 \text{ or } 20} = \text{pounds water per hour to be circulated}$$

where

h_c = heat release per square foot of ceiling

h_a = heat release to floor above

A = area

$A\ (h_c + h_a)$ = total heat release

A temperature drop of 10 to 28° F is usually allowed.

6. For steam coils divide total heat release by the latent heat of steam (usually 1000 Btu) at the required temperature to calculate pounds of steam to be supplied per hour.

126. Radiant Heating of Rooms Above and Below Concrete Slab with Embedded Coils (Figure 9-E)

The routine of determining pipe coil design for this application is practically the same as that outlined for ceiling coils heating only the room below. However the determination of heat loss to space above is not required. To determine the quantity of heating medium the total ceiling and floor heat release is considered. An additional allowance must be made to determine the amount of insulation which must be placed between the coils and the floor above to provide the required floor temperature. The required coefficient of heat transmission of the insulation construction above the pipe coils is determined by solving the equation:

$$U = h\ (t_p - t_f)$$

where

U = required transmission coefficient

h = floor heat release

t_p = average temperature in plane of pipe coil

t_f = average floor temperature desired

A construction can be selected after U is determined.

This design is popular in Great Britain and some design data are available. However, in using British data, consideration must be given to the local practice of using higher ceiling temperatures and lower room temperatures than American engineers recommend.

Research has shown that the heat release of the surface decreases as the spacing of the pipe increases even though the heat release per lineal foot of pipe increases.

TABLE 89

COMPARISON OF PIPE CENTERS FOR CONCRETE CEILING SLABS

Spacing of ½ in. Pipe	Heat Release from Panel Btu Per Square Foot Per Hour
4½ in.	115
6 in.	100 (English practice)
9 in.	70
12 in.	54
18 in.	38

The above table is based on 60° F difference in temperature between room air and hot water.

TABLE 90

HEAT RELEASE CONCRETE CEILING PANELS

Temperature Difference Air in Room to Water in Pipe	Heat Release Btu Per Hour Per Square Foot
20	23
30	41
40	59
50	79
60	100
70	120
80	140

The above data refer to general practice in Great Britain for coils embedded in concrete with ceiling plastered and hot water circulated at 120 to 130° F.

127. Pipe Coils Between Wood Beams Filled with Mortar (Figure 9-F)

Same method as previously outlined may be used, except that the transmission coefficient for mortar is 8 Btu per square foot per degree difference per hour per inch thickness.

Chapter VIII

ELEMENTARY THERMODYNAMICS AND STEAM

One of the greatest engineering problems is the transformation of naturally-occurring forms of energy into those forms which best serve the purposes of mankind. The form of energy which men find to be the most useful for their purposes is that form which we call mechanical energy. Unfortunately, the forms in which energy naturally occurs are seldom those which are immediately available to our purpose, or which may be transformed into work by simple mechanical appliances, such as windmills or water-wheels. The needs of society are usually such that man is commonly obliged to avail himself of that great store of natural energy found in the form of potential chemical energy in combustible substances. This form of energy can be liberated, so far as we know, only in the form of heat. In order to use natural sources of energy, therefore, it is necessary to transform them first into heat, and then, in most cases, to transform this heat into some more useful form of energy, such as work, electricity or light.

In doing this, we make use of certain forms of engineering apparatus which we may term *thermodynamic machines.*

Thermodynamics, in the sense in which it is used by physicists, is that branch of science which treats of the effects produced by heat and the phenomena accompanying its various manifestations. When used by the engineer, however, the term thermodynamics is understood to mean that branch of engineering which deals with the interconversion of heat and work, and the phenomena attendant thereon.

128. General Definitions

The thermodynamic state of a substance can be defined when its temperature, pressure, density, mass and composition are known. A substance having a definite mass and homogeneous composition is, in thermodynamics, termed a *body,* no matter what the physical state of that substance may be. A body is in *thermal equilibrium* when every part of it is at the same temperature. A body is in *thermodynamic* equilibrium when every part of it has the same temperature, pressure and density. An *isolated body* is one which is so situated that it neither gains nor loses heat. An *isolated system* (usually termed a *system*) is a group of bodies so situated that the system neither gains nor loses heat.

When a body undergoes a change in thermodynamic state, it is said to undergo a *process.* If, at every instant during this process, all parts of the body are in thermodynamic equilibrium the process is said to be *reversible.* If at any time during the process all parts of the substance are not at the same temperature, pressure, and density, the process is said to be *irreversible* or *sweeping.*

A substance which undergoes a reversible process can be brought back to its initial state by causing it to pass through each successive stage of the process in reversed order. This is called *reversing* the process. A substance which undergoes a sweeping process cannot be brought back to its initial state by reversing the sweeping process.

As an example of a reversible process, we may take the adiabatic expansion of a gas, a process which lowers its temperature and pressure and increases its volume. At every instant during the expansion every portion of the gas has the same temperature, pressure, and density. By raising the pressure upon the gas, its temperature and density will be increased, and its volume decreased, and it will be brought back to its initial state, after passing successively through each state through which it passes during its expansion.

As an example of a sweeping process, we may take the case of a body of gas confined under pressure, which is allowed to escape from the containing vessel through an orifice. While it is escaping, that portion of the gas which has passed through the orifice will be of a different temperature, pressure, and density from that portion yet within the containing vessel, and the gas is not in thermodynamic equilibrium during the process. Neither is there any possible method which will enable us to pass the gas from a region of low pressure back through the orifice into a region of high pressure, and yet have it pass in turn through the several conditions through which it passed during the progress of the sweeping process, and the process cannot be reversed.

When a substance is caused to undergo a series of processes, for any purpose, and is finally brought back to the thermodynamic state which it had initially, the substance is said to perform a *cycle.* The substance performing the cycle is called the *working substance,* and in case this substance is a fluid, as it usually is, it is called the *working fluid.* In case the series of processes which the working fluid undergoes in any cycle are all reversible processes, the cycle is said to be *reversible,* or *perfect.* In case one or more of the processes are sweeping processes, the cycle is said to be *irreversible,* or *imperfect.* The purpose of causing a working substance to perform a cycle is usually either to transform heat into work, or to transfer heat from a region of low temperature to a region of high temperature. A thermodynamic machine in which a cycle is performed for the purpose of transforming heat into work is called a *heat engine.* A machine in which a cycle is performed for the purpose of transferring heat from a region of low temperature to a region of high temperature is called a *refrigerating machine.*

When the temperature of the working substance remains constant an *isothermal change* takes place.

129. Heat as a Form of Energy

Energy may be defined as the capacity for doing work.

Energy may be in transition as work done by the action of a superior force, as heat transfer due to a temperature difference or as electrical energy due to an electrical potential difference.

Kinetic energy is possessed by a body when work can be done as a result of a change in velocity. Thus if a body of weight w has a velocity of V_a its kinetic energy equals $\frac{wV_a^2}{2g}$. If the velocity is changed to V_b in the process of doing work, the work done, W, will be expressed:

$$W = \frac{w(V_a^2 - V_b^2)}{2g}$$

Internal or *intrinsic energy* is the energy stored in a body above some datum state. This quantity will be denoted by U.

Work is the action of a force in displacing a resistance through a distance. It is calculated by multiplying the force by the distance.

130. Units of Energy

The conventional units of energy are derived from simple phenomena associated with the transition of energy. They are:

(a) *The foot-pound* (ft-lb), which is the amount of energy (exhibited as work) associated with the action of a force of 1 lb applied through a distance of 1 ft.

(b) *The British thermal unit* (Btu), which is $\frac{1}{180}$ of the energy required to raise the temperature of a pound mass of pure water from 32° F to 212° F at a constant atmospheric pressure of 14.696 lb per square inch.

(c) The horsepower hour (hp-hr) which is the transition of energy at the rate of 33,000 ft-lb per minute or 550 ft-lb per second.

(d) The Kilowatthour (Kwhr) which is the flow of energy at the rate of 1000 watts per hour and equivalent to 1.341 hp-hr, or 44,253 ft-lb min.

The relative magnitudes of the several units are:

$$\text{one Btu} = 778.6 \text{ ft-lb}$$
$$\text{one hp-hr} = 2543 \text{ Btu}$$
$$\text{one Kwhr} = 3411.5 \text{ Btu}$$

131. Enthalpy

Enthalpy (formerly called heat content, or total heat), defined as heat content per unit mass of a substance, is the sum of its internal energy U, in Btu per pound above some datum level and the product Pv, of its pressure and volume at the given state.

$$h = U + APv$$

where

h = Btu per pound

$A = \frac{1}{778.6}$

P = pressure, pounds per square foot

V = specific volume of 1 lb, cubic feet

132. The First Law of Thermodynamics

This law is a statement of the principle of the conservation of energy; energy may exist in many varied and interchangeable forms but may not be created or destroyed.

133. The Second Law of Thermodynamics

This law embodies the idea of the availability and degradation of energy, or the tendency of high-grade energy to degenerate into low-grade energy. The second law is an expression for the efficiency of the perfect elementary engine.

$$\frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1}$$

where

Q_1 = heat received

T_1 = corresponding absolute temperature

Q_2 = heat rejected

T_2 = corresponding absolute temperature

A conclusion of engineering significance derived from the second law is that the *maximum* work output obtainable from an engine equals the change of enthalpy for an adiabatic isentropic expansion of the fluid flowing through the engine.

134. Reversibility

A reversible process fulfills the following conditions:

(a) When the direction of the process is reversed, the system taking part in the process can assume in inverse order the states traversed in the direct process.

(b) The external actions are the same for the direct and reversed processes.

(c) Not only the system undergoing the change but all connected systems can be restored to initial conditions. (Goodenough).

Any process that fails to meet these requirements is an irreversible process. Three irreversible processes are of frequent occurrence:

(a) The direct conversion of work into heat through the agency of friction.

(b) The conduction of heat from a body at temperature t_1 to another at temperature t_2.

(c) The throttling or wire-drawing of a fluid in flowing through an orifice from a region of higher to a region of lower pressure.

135. The Pressure-Volume Diagram

This is a diagram in which the ordinates are pressures and the abscissas volumes. If a line is drawn on this diagram which represents the successive change of state of the substance in passing from state a to state b, the area under this line represents the work done on or by the substance during this change, provided that the change of state is mechanically reversible.

136. Adiabatic Process

An adiabatic process is any process in which no heat is transferred to or from the working substance from any outside source.

137. Reversible Adiabatic Process

This is a process occurring at constant entropy, i.e., an isentropic process. In this case no heat from an outside source is added to the working substance, nor is any heat lost to an outside body and no friction, either mechanical or fluid, exists. The reversible adiabatic processes make available the greatest amount of energy for doing external work and are the standards of reference in the Rankine cycle. In steam turbine calculations the ideal heat drop ($h_1 - h_2$) per pound of steam, and consequent maximum work, is found from an isentropic adiabatic diagram.

138. Irreversible Adiabatic Process

This is a process in which there is no transfer of heat from the outside to the working substances, but in which more or less of the ideally available energy of the working substance fails of utilization as work output due to the presence of fluid or mechanical friction.

139. Characteristic Gas Equation (Re-statement)

This equation is a combination of Boyle's and Charles's law and is expressed

$$Pv = wRT$$

where R is a constant for a given gas or mixture of gases.

140. Avogadro's Law (Re-statement)

The *mol* is that quantity of gas, at any given pressure and temperature, of which the weight in pounds numerically equals its molecular weight. At 32° F and an atmospheric pressure of 14.7 lb per square inch the volume of a mol of gas equals 358.7 cu ft. This volume of any gas weighs m pounds where m = molecular weight.

141. Universal Gas Constant

Let v in the characteristic equation of a gas = 1 mol at atmospheric pressure and 32° F. On solving

$$mR = \frac{Pv}{T} = \frac{(2116.4 \times 358.7)}{459.6 + 32} = 1544$$

This is called the *universal gas constant.*

For any gas,

$$R = \frac{1544}{m}$$

Also

$$\frac{R}{778.6} = \frac{1544}{778.6m} = \frac{1.985}{m} = C_p - C_v$$

or

$$R = \frac{778.6}{C_p - C_v} = K_p - K_c$$

$(K_p - K_c)$ is the difference between the specific heats of 1 lb of gas at constant pressure and at constant volume, expressed in foot-pounds.

Although gases have varying specific heats, the difference $m\ (C_p - C_v)$ remains a constant. The universal gas constant mR also is substantially constant for real gases. Hence $Pv = RT$ may be used for 1 lb of actual gas.

142. Internal Energy

Joule has shown that the internal energy of a perfect gas depends on temperature only, and is independent of volume. The change in internal or intrinsic energy of 1 lb of gas may be expressed as

$$778(U_1 - U_0) = K_v(T_1 - T_0) = \frac{R(T_1 - T_0)}{\gamma - 1}$$

The total internal energy of w pounds of gas *wv*, at pressure P_1 in pounds per square foot and volume v, in cubic feet with $\gamma = \frac{C_p}{C_v}$ would be $P_1 v_1 \ (\gamma - 1)$, measured above absolute zero, if the substance follows the perfect gas laws down to absolute zero and its specific heat is constant.

When air is expanded or compressed isothermally, Pv is a constant, and the internal energy remains constant; the work done in expansion equals the heat added, and the work done in compression equals the heat rejected.

143. Vapors

A substance which is a solid or liquid under ordinary conditions usually vaporizes upon the application of heat. A solid most frequently passes through the intermediate liquid state but occasionally passes directly into the gaseous state.

The gaseous state of matter when it coexists in a system with its non-gaseous phases, is known as *vapor*. It fills its container, thereby exerting a pressure as does a gas, but does not follow the laws of ideal gases even approximately. A liquid when heated at constant pressure reaches a temperature, depending on the pressure, at which it commences to change its physical state to a vapor at that constant temperature. At temperatures below the boiling point the water is *subcooled*. When the liquid reaches the boiling point under these conditions its energy is called the *saturated heat of the liquid*.

The heat added to complete evaporation at constant pressure and temperature at the boiling point is *the latent heat of vaporization*. When the last particle has been evaporated but the temperature has not risen above the boiling point, the vapor is dry and saturated. Any condition in which droplets of liquid in the form of fog are suspended in the vapor and are not completely evaporated produces a wet mixture. The mixture has the same temperature as dry saturated vapor at the same pressure. The ratio of completely evaporated liquid to total substance present in such a wet mixture is known as the *per cent quality*. If the dry steam-saturated vapor mixture is heated still further, its temperature rises and it becomes superheated. In this condition it approximately follows the laws of perfect gases. The degree of superheat is the difference between the actual temperature of the superheated vapor and the boiling temperature at the same pressure.

In every liquid there is a certain pressure at which the saturated liquid changes into vapor with no change in volume and without the addition of latent heat. This is the *critical state*.

A vapor that is not near the saturation point behaves like a gas under changes of temperature and pressure. If it is sufficiently compressed or cooled, it reaches a point where it begins to condense. It then no longer obeys the same laws as a gas, since its pressure cannot be increased by diminishing the size of the vessel containing it, but remains constant, except when the temperature is changed.

144. Water

Water (H_2O) is a chemical compound formed by the union of two volumes of hydrogen and one volume of oxygen, or two parts by weight of hydrogen and sixteen parts by weight of oxygen. 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 (231 cu in.) of water weighs approximately $8\frac{1}{3}$ lb, and a cubic foot of water is equivalent to 7.48 gal. The specific volume of water depends on the temperature because it is the reciprocal of its density.

TABLE 91

THERMAL PROPERTIES OF WATER

Temperature °F	Saturated Pressure Pounds Per Square Inch	Specific Volume, Cubic Feet Per Pound	Density, Pounds Per Cubic Foot	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	1.0200
300	67.01	0.01742	57.41	1.0260
350	134.62	0.01797	55.65	1.0440
400	247.25	0.01865	53.62	1.0670
450	422.61	0.01950	51.30	1.0950
500	681.09	0.02050	48.80	1.1300
550	1045.4	0.02190	45.70	1.2000
600	1544.6	0.02410	41.50	1.3620
700	3096.4	0.03940	25.40	——

Courtesy of A.S.H.V.E.

Water pressures are generally stated in feet of water, inches of water or pounds per unit area. At 62° F, when h is equal to the head in feet, the pressure of a column of water is 62.383 h lb per square foot, or 0.433 h lb per square inch. A column of water 2.309 ft (27.71 in.) high exerts a pressure of 1 lb per square inch at 62° F. The boiling point of water varies with the atmospheric pressure; it is lower at higher altitudes. A change in pressure will always be accompanied by a change in the boiling point, and there will be a corresponding change in the latent heat of evaporation. The specific heat of water, or the amount of heat (in Btu) required to raise the temperature of one pound of water 1° F, varies with the temperature, but it is commonly assumed to be the same at all temperatures. Steam tables are based on exact values. 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 in question.

145. Sensible and Latent Heat of Water

The heat required to raise the temperature of one pound of water from 32° F to the boiling point is known as *the heat content of the saturated liquid* (h_f) 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 saturated steam. The heat thus added is known as the *latent heat of evaporation.*

146. Properties of Steam

Steam is water which exists in a gaseous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the saturated liquid into vapor. This change takes place at a definite and constant temperature which is determined solely by the pressure of the steam. The specific volume of steam is the volume occupied by one pound of steam; this decreases as the pressure increases. The reciprocal of this, or the weight of one cubic foot of steam, is the density.

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 it is called dry saturated steam. If this is 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.

147. Abbreviations and Symbols Pertaining to Steam Tables

The symbols listed below have been selected from an approved list.

Change in specific volume during evaporation	V_{fg}
Density, weight per unit volume, specific weight	d or ρ
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
Heat content, Total heat, Enthalpy	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}
Internal energy, Intrinsic energy (Capital for any weight, small for unit weight)	U or u

Mechanical equivalent of heat	J
Quality of steam, Pounds of dry steam per pound of mixture	x
Saturated liquid at saturation pressure and temperature, liquid in contact with vapor	subscript f
Specific heat at constant pressure	c_p
Specific heat at constant volume	c_v
Specific volume, Volume per unit weight	v
Temperature F or C (Theta is preferable when t is used for time in same discussion)	t or θ
Temperature absolute	T
Vaporization values at constant pressure. Differences between values for saturated vapor and saturated liquid at same pressure	subscript fg
Volume (total)	V
Weight of a major item	W
Weight rate, weight per unit of power, or time	w
Work (total)	W

148. The Steam Table

A condensed steam table is presented here as an introduction to the use of psychrometric formulas, tables and charts.

149. Computation of Steam Table

Specific volume

The volume v_f of 1 lb of saturated liquid increases during evaporation to v_{fg} and becomes v_g in the case of the saturated vapor. These values are given in columns 3, 4 and 5 for various steam pressures. To find the volume of 1 lb of wet steam,

$$v_x = v_f + xv_{fg}$$

where x is the quality.

Table 92

SATURATED STEAM: PRESSURE TABLE

Abs. Press. Lb/Sq In.	Temp. Fahr.	Specific Volume Sat. Liquid	Specific Volume Evap.	Specific Volume Sat. Vapor	Enthalpy Sat. Liquid	Enthalpy Evap.	Enthalpy Sat. Vapor	Entropy Sat. Liquid	Entropy Evap.	Entropy Sat. Vapor	Abs. Press. Lb/Sq In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
0.0886	32.00	0.01602	3305.7	3305.7	0	1075.1	1075.1	0	2.1865	2.1865	**0.0886**
0.125	40.69	0.01602	2383.7	2383.7	8.74	1070.2	1078.9	0.0176	2.1388	2.1564	**0.125**
0.250	59.31	0.01603	1235.8	1235.8	27.38	1059.5	1086.9	0.0542	2.0414	2.0956	**0.250**
0.500	79.58	0.01607	641.71	641.73	47.60	1048.0	1095.6	0.0924	1.9434	2.0358	**0.500**
1	101.76	0.01614	333.77	333.79	69.72	1035.5	1105.2	0.1326	1.8443	1.9769	**1**
5	162.25	0.01641	73.584	73.600	130.13	1000.7	1130.8	0.2347	1.6090	1.8437	**5**
10	193.21	0.01659	38.445	38.462	161.17	982.1	1143.3	0.2834	1.5042	1.7876	**10**
14.696	212.00	0.01672	26.811	26.828	180.07	970.3	1150.4	0.3120	1.4446	1.7566	**14.696**
15	213.03	0.01672	26.303	26.320	181.11	969.6	1150.7	0.3135	1.4413	1.7548	**15**
20	227.96	0.01683	20.093	20.110	196.16	959.9	1156.1	0.3356	1.3959	1.7315	**20**
30	250.34	0.01700	13.746	13.763	218.83	945.2	1164.0	0.3680	1.3312	1.6992	**30**
40	267.24	0.01715	10.489	10.506	236.02	933.7	1169.7	0.3919	1.2844	1.6763	**40**
50	281:01	0.01727	8.505	8.522	250.09	923.9	1174.0	0.4110	1.2473	1.6583	**50**
60	292.71	0.01738	7.162	7.179	262.10	915 4	1177.5	0.4271	1.2166	1.6437	**60**
70	302.92	0.01748	6.193	6.210	272.61	907.9	1180.5	0.4409	1.1905	1.6314	**70**
80	312.03	0.01757	5.458	5.476	282.02	901.1	1183 1	0.4532	1.1677	1.6209	**80**
90	320.27	0.01766	4.880	4.898	290.57	894.8	1185.4	0.4641	1.1472	1.6113	**90**
100	327.83	0.01774	4.415	4.433	298.43	888.9	1187.3	0.4741	1.1287	1.6028	**100**
110	334.79	0.01782	4.032	4.050	305.69	883.3	1189.0	0.4832	1.1118	1.5950	**110**
120	341.26	0.01789	3.710	3.728	312.46	878.1	1190.6	0.4916	1.0963	1.5879	**120**

By courtesy of Combustion Engineering Co.

TABLE 92 (*continued*)

Abs. Press. Lb/Sq In.	Temp. Fahr.	Specific Volume			Enthalpy			Entropy			Abs. Press. Lb/Sq In.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
130	347.31	0.01796	3.437	3.455	318.81	873.2	1192.0	0.4995	1.0820	1.5815	130
140	353.03	0.01803	3.202	3.220	324.83	868.5	1193.3	0.5069	1.0686	1.5755	140
150	358.43	0.01809	2.998	3.016	330.53	863.9	1194.4	0.5138	1.0560	1.5698	150
160	363.55	0.01815	2.816	2.834	335.95	859.6	1195.5	0.5204	1.0442	1.5646	160
170	368.42	0.01821	2.656	2.674	341.11	855.2	1196.3	0.5266	1.0327	1.5593	170
180	373.08	0.01827	2.514	2.532	346.07	851.1	1197.2	0.5325	1.0220	1.5545	180
190	377.55	0.01833	2.386	2.404	350.83	847.2	1198.0	0.5382	1.0119	1.5501	190
200	381.82	0.01839	2.270	2.288	355.40	843.3	1198.7	0.5436	1.0021	1.5457	200
210	385.93	0.01844	2.165	2.183	359.80	839.6	1199.4	0.5488	0.9929	1.5417	210
220	389.89	0.01850	2.067	2.086	364.05	835.8	1199.9	0.5538	0.9838	1.5376	220
230	393.70	0.01855	1.9803	1.9989	368.16	832.2	1200.4	0.5585	0.9752	1.5337	230
240	397.40	0.01860	1.8990	1.9176	372.16	828.7	1200.9	0.5632	0.9669	1.5301	240
250	400.97	0.01866	1.8244	1.8431	376.04	825.4	1201.4	0.5677	0.9590	1.5267	250
260	404.43	0.01870	1.7555	1.7742	379.78	822.0	1201.8	0.5720	0.9513	1.5233	260
270	407.79	0.01875	1.6913	1.7101	383.43	818.8	1202.2	0.5761	0.9439	1.5200	270
280	411.06	0.01880	1.6316	1.6504	386.99	815.5	1202.5	0.5802	0.9365	1.5167	280
290	414.24	0.01885	1.5758	1.5947	390.47	812.4	1202.9	0.5841	0.9296	1.5137	290
300	417.33	0.01890	1.5237	1.5426	393.85	809.3	1203.2	0.5879	0.9228	1.5107	300
350	431.71	0.01912	1.3064	1.3255	409.70	794.7	1204.4	0.6057	0.8915	1.4972	350
450	456.27	0.0195	1.0123	1.0318	437.18	767.8	1205.0	0.6357	0.8382	1.4739	450
500	467.00	0.0197	0.9077	0.9274	449.40	755.5	1204.9	0.6488	0.8153	1.4641	500
550	476.94	0.0199	0.8217	0.8416	460.83	743.6	1204.4	0.6609	0.7939	1.4548	550
600	486.21	0.0201	0.7494	0.7695	471.59	732.0	1203.6	0.6721	0.7739	1.4460	600
650	494.90	0.0203	0.6879	0.7082	481.73	721.0	1202.7	0.6826	0.7553	1.4379	650

TABLE 92 (*continued*)

Abs. Press. Lb/Sq In.	Temp. Fahr.	Specific Volume: Sat. Liquid	Specific Volume: Evap.	Specific Volume: Sat. Vapor	Enthalpy: Sat. Liquid	Enthalpy: Evap.	Enthalpy: Sat. Vapor	Entropy: Sat. Liquid	Entropy: Evap.	Entropy: Sat. Vapor	Abs. Press. Lb/Sq In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
700	503.09	0.0205	0.6347	0.6552	491.49	710.1	1201.6	0.6925	0.7376	1.4301	**700**
750	510.83	0.0207	0.5884	0.6091	500.8	699.4	1200.2	0.7019	0.7206	1.4225	**750**
800	518.20	0.0209	0.5476	0.5685	509.7	689.1	1198.8	0.7108	0.7047	1.4155	**800**
850	525.23	0.0210	0.5116	0.5326	518.3	678.9	1197.2	0.7194	0.6893	1.4087	**850**
900	531.94	0.0212	0.4794	0.5006	526.6	669.0	1195.6	0.7276	0.6746	1.4022	**900**
950	538.38	0.0214	0.4503	0.4717	534.6	659.2	1193.8	0.7355	0.6605	1.3960	**950**
1000	544.56	0.0216	0.4240	0.4456	542.4	649.5	1191.9	0.7431	0.6468	1.3899	**1000**
1050	550.52	0.0218	0.4001	0.4219	550.0	640.0	1190.0	0.7504	0.6335	1.3839	**1050**
1100	556.26	0.0219	0.3783	0.4002	557.4	630.4	1187.8	0.7575	0.6205	1.3780	**1100**
1150	561.81	0.0221	0.3583	0.3804	564.6	621.0	1185.6	0.7644	0.6079	1.3723	**1150**
1200	567.19	0.0223	0.3397	0.3620	571.7	611.5	1183.2	0.7712	0.5955	1.3667	**1200**
1250	572.39	0.0225	0.3228	0.3453	578.6	602.2	1180.8	0.7777	0.5835	1.3612	**1250**
1300	577.43	0.0227	0.3067	0.3294	585.4	592.9	1178.3	0.7840	0.5717	1.3557	**1300**
1350	582.32	0.0229	0.2918	0.3147	592.1	583.7	1175.8	0.7902	0.5602	1.3504	**1350**
1400	587.07	0.0231	0.2780	0.3011	598.6	574.6	1173.2	0.7963	0.5489	1.3452	**1400**
1450	591.70	0.0233	0.2652	0.2885	605.0	565.5	1170.5	0.8022	0.5379	1.3401	**1450**
1500	596.20	0.0235	0.2530	0.2765	611.4	556.3	1167.7	0.8081	0.5269	1.3350	**1500**
1550	600.59	0.0237	0.2416	0.2653	617.7	547.1	1164.8	0.8138	0.5160	1.3298	**1550**
1600	604.87	0.0239	0.2309	0.2548	623.9	538.0	1161.9	0.8195	0.5054	1.3249	**1600**
1650	609.05	0.0241	0.2207	0.2448	630.0	528.8	1158.8	0.8250	0.4948	1.3198	**1650**
1700	613.12	0.0243	0.2111	0.2354	636.1	519.6	1155.7	0.8304	0.4843	1.3147	**1700**
1750	617.11	0.0245	0.2020	0.2265	642.1	510.4	1152.5	0.8359	0.4740	1.3099	**1750**
1800	621.00	0.0247	0.1933	0.2180	648.0	501.3	1149.3	0.8412	0.4639	1.3051	**1800**
1850	624.82	0.0249	0.1850	0.2099	653.9	492.0	1145.9	0.8465	0.4537	1.3002	**1850**

TABLE 92 (*continued*)

Abs. Press. Lb/Sq In.	Temp. Fahr.	Specific Volume Sat. Liquid	Specific Volume Evap.	Specific Volume Sat. Vapor	Enthalpy Sat. Liquid	Enthalpy Evap.	Enthalpy Sat. Vapor	Entropy Sat. Liquid	Entropy Evap.	Entropy Sat. Vapor	Abs. Press. Lb/Sq In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
1900	628.55	0.0252	0.1770	0.2022	659.9	482.5	1142.4	0.8517	0.4434	1.2951	**1900**
1950	632.20	0.0254	0.1695	0.1949	665.8	473.0	1138.8	0.8569	0.4332	1.2901	**1950**
2000	635.78	0.0257	0.1622	0.1879	671.7	463.5	1135.2	0.8620	0.4231	1.2851	**2000**
2100	642.73	0.0262	0.1486	0.1748	683.4	444.2	1127.6	0.8722	0.4029	1.2751	**2100**
2200	649.42	0.0267	0.1359	0.1626	695.0	424.4	1119.4	0.8823	0.3826	1.2649	**2200**
2300	655.87	0.0274	0.1240	0.1514	706.7	404.3	1111.0	0.8923	0.3624	1.2547	**2300**
2400	662.09	0.0280	0.1130	0.1410	718.5	382.9	1101.4	0.9025	0.3413	1.2438	**2400**
2500	668.10	0.0287	0.1026	0.1313	730.7	360.3	1091.0	0.9127	0.3195	1.2322	**2500**
2600	673.91	0.0295	0.0924	0.1219	743.1	337.0	1080.1	0.9232	0.2973	1.2205	**2600**
2700	679.54	0.0305	0.0818	0.1123	756.1	312.2	1068.3	0.9342	0.2740	1.2082	**2700**
2800	684.98	0.0316	0.0716	0.1032	770.0	284.6	1054.6	0.9458	0.2486	1.1944	**2800**
2900	690.26	0.0329	0.0612	0.0941	785.2	252.9	1038.1	0.9586	0.2199	1.1785	**2900**
3000	695.37	0.0346	0.0503	0.0849	802.6	216.7	1019.3	0.9731	0.1876	1.1607	**3000**
3100	700.29	0.0372	0.0380	0.0752	824.6	169.4	994.0	0.9916	0.1460	1.1376	**3100**
3200	705.04	0.0443	0.0153	0.0596	871.3	75.3	946.6	1.0311	0.0647	1.0958	**3200**
3206.2*	705.34	0.0541	0	0.0541	910.3	0	910.3	1.0645	0	1.0645	**3206.2***

* Critical pressure

This formula must be used for pressures above 250 lb per square inch. At low pressures v_f is relatively small as compared to v_g and may be neglected in exhaust steam calculations.

Then

$$v_x' = xv_g$$

150. Enthalpy or Total Heat

Enthalpy or total heat is calculated above 32° F and incorporates three elements

(a) The total heat of saturated liquid h_f, which is the heat required to raise the temperature of 1 lb of water from 32° F to the boiling temperature corresponding to the pressure.

(b) The internal latent heat h_i, which is the heat required to evaporate completely the water at that temperature and pressure.

(c) The external work done by the steam in expanding against the pressure under which it is generated, APv_{fg},

where

$$A = \frac{1}{778.6}$$

P = pressure, pounds per square foot

v_{fg} = change in vapor volume, cubic feet per pound

The sum of the last two factors is the latent heat of steam, h_{fg}, or the total heat of evaporation per pound. Thus the heat required to generate 1 lb of steam at 212° F from water at 32° F and 14.696 lb per square inch is calculated below:

Total heat of saturated liquid at 212° F, h_f = 180.0 Btu

Internal latent heat of steam h_i = 897.8 Btu

$$\text{External work} = APv_{fg} = \frac{14.696 \times 144(26.82 - 0.017)}{778.6} = 72.4 \text{ Btu}$$

Total heat of evaporation, h_{fg} = 897.8 Btu
72.4 Btu

$$= \underline{\ \ 970.2 \text{ Btu}}$$

Enthalpy or total heat of saturated vapor, h_g = 1150.2 Btu

The enthalpy or total heat of wet steam at quality x is

$$h_x = h_f + xh_{fg}$$

The enthalpy or total heat of superheated steam is found by adding to the total heat of saturated vapor the energy added in the superheat.

CHAPTER IX

AIR AND VAPOR MIXTURES

When two or more perfect gases are brought into contact with one another, the particles of one tend to pass between the particles of the other. As a result, after a lapse of time, the two gases will form a homogeneous mixture. This process of mixing is known as diffusion, and the resulting mixture is, in every sense except a chemical one, a perfectly homogeneous body and will remain so, provided that it undergoes no chemical action and that its component particles are not separated by enclosing the mixture within a porous vessel. When a mixing of gases occurs, all of the constituents will come to the same temperature. The mass of the mixture will be the sum of the masses of the several constituents. If the mixture is confined within a vessel, each of the constituents will exert upon the walls a pressure equal to the pressure which that constituent would exert if it were confined separately within the vessel at the same temperature. Consequently the pressure of the mixture will be the sum of the pressures of the several constituents. If each of the constituents were confined at the pressure and temperature of the mixture, it would occupy a definite volume. If the constituents are all perfect gases the sum of these volumes will be the volume of the mixture. A mixture of a gas and a vapor, e.g., atmospheric air and the moisture in it, becomes homogeneous through diffusion in exactly the same way as do mixtures of two or more gases. If the vapor is saturated by virtue of the presence of its liquid, the pressure which it exerts will depend upon the temperature, but the pressure which the gas will exert will depend upon its volume and mass as well as on the temperature. The pressure exerted by the gas may be computed by subtracting from the pressure of the mixture the saturation pressure of the vapor at the temperature of the mixture. If the vapor is superheated, it will behave like a gas and the pressure exerted will be the sum of the pressures of each constituent of the mixture, since the pressure of the superheated vapor depends upon the mass of the vapor and the volume to which it is confined. In air-conditioning computations it is essential to consider the variation in the properties of air when varying quantities of water vapor are present. In such computations, it is necessary to consider air as a mixture of a gas of known properties with superheated water vapor or steam.

151. Psychrometric Equations, Tables and Charts

The basic laws governing many of the phenomena of atmospheric moisture have been reviewed in the preceding chapters. Accepted data used in psychrometric calculations in the past have been based largely on empirical formulas; from time to time they have been corrected by research. Recent investigators have determined the most important properties of water vapor with great accuracy. New information that is vitally important in meeting the present day demands of engineers who devote their time to heating, ventilating, refrigeration and air-conditioning practice is given below in the form of equations, tables, diagrams and charts.

The engineer may select any one of three methods as his routine in solving psychrometric problems.

(1) Algebraic equations
(2) Psychrometric tables
(3) Psychrometric charts

For the research engineer, or the engineer conducting a test with the object of certifying the performance of a system complying with specifications, the solution by algebraic equations is professionally the logical method of presenting data. As an example assume that 500 tons (100,000 Btu per minute) of refrigeration per minute are to be delivered to 12,000 lb of air, in order to supply cool air to an office building or theater, capacity to be determined by total heat in supply air at a wet-bulb temperature computed to be 56° F. If the average of the test readings gives 57° F, which is one degree above contract guartee, computations will prove a deficiency of 0.63 Btu per pound of air. This, translated into refrigeration units, amounts to a deficiency of 7560 Btu per minute or to 37.8 tons which is 6.5% less than guaranteed capacity. The accuracy of wet-bulb readings and of the psychrometric chart readings may be questioned by the parties making the test. Then the question will arise as to which chart is to be recognized as authentic.

Test codes are available and these are recognized by engineers as the authority to assist them in rendering a decision in cases similar to this one. The values in Table 93, sponsored by the A.S.H.V.E. are considered as reliable as the basic equations given below, for the simple reason that the equations are, in general, the basis of the calculations for the table. However, the psychrometric chart is very convenient for frequent use and its use is limited only by the degree of accuracy required; generally an error of ± 3% is to be expected. The equations are sometimes more convenient for equal accuracy or more accurate for equal convenience. Symbols and abbreviations adopted in this chapter are the latest recognized by the A.S.H.V.E. The psychrometric chart following is a graph based on values given in the psychrometric table (Table 93).

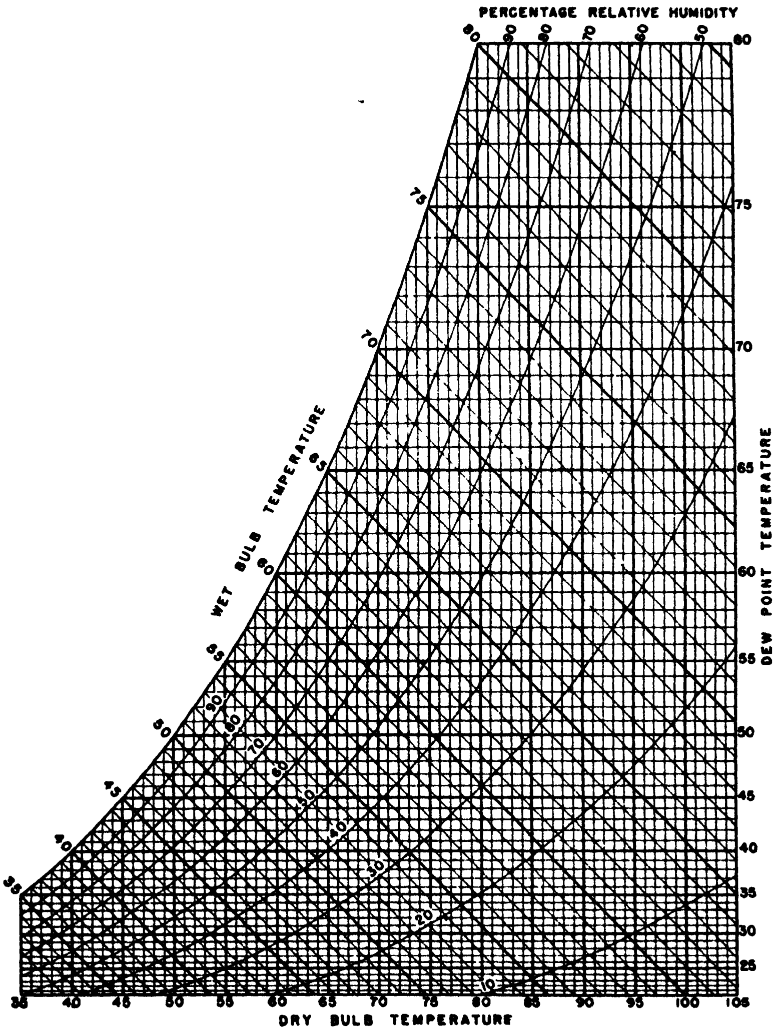


FIGURE 10
Psychrometric Chart

Barometric pressure = 29.92 in. Hg.
Wet-bulb lines are lines of constant heat content.

Table 93

THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds $\times 10^5$	Saturation Humidity Ratio: Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	Volume: v_{as} $(v_s - v_a)$	Volume: Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	Enthalpy: h_{as} $(h_s - h_a)$	Enthalpy: Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure $p_s \times 10^5$: In. of Hg	Saturation Pressure: Lb per Sq In.	Temp Deg F
—60	2.108	0.14756	10.07	0.00	10.07	—14.48	0.02	—14.46	—185.4	101.4	49.808	—60
—59	2.262	.15834	10.09	.00	10.09	—14 23	.02	—14.21	—185.0	108.8	53.443	—59
—58	2.418	.16926	10.12	.00	10.12	—13.99	.02	—13.97	—184.6	116.3	57.127	—58
—57	2.595	.18165	10.14	.00	10.14	—13.75	.03	—13.72	—184.2	124.8	61.302	—57
—56	2.773	.19411	10.17	.00	10.17	—13 50	.03	—13.47	—183.8	133.4	65.526	—56
—55	2.973	0.20811	10.19	0.00	10.19	—13.26	0.03	—13.23	—183.4	143.0	70.242	—55
—54	3.181	.22267	10.22	.00	10.22	—13.02	.03	—12.99	—182.9	153.0	75.154	—54
—53	3.399	.23793	10.24	.00	10.24	—12.78	.04	—12.74	—182.5	163.5	80.311	—53
—52	3.636	.25452	10.27	.00	10.27	—12.53	.04	—12.49	—182.1	174.9	85.911	—52
—51	3.888	.27216	10.29	.00	10.29	—12.29	.04	—12.25	—181 7	187.0	91.854	—51
—50	4.156	0.29092	10.32	0.00	10.32	—12 05	0.04	—12 01	—181 3	199.9	98.191	—50
—49	4.428	.30996	10.34	.00	10 34	—11 81	.05	—11.76	—180.9	213.0	104.63	—49
—48	4.738	.33166	10.37	.00	10.37	—11.57	.05	—11.52	—180 4	227.9	111.94	—48
—47	5.054	.35378	10.40	.00	10.40	—11 32	.05	—11.27	—180.0	243.1	119.41	—47
—46	5.395	.37765	10.42	.00	10.42	—11.08	.06	—11.02	—179.6	259.5	127.47	—46
—45	5.753	0.40271	10.45	0.00	10.45	—10 84	0.06	—10.78	—179.2	276.7	135.92	—45
—44	6.153	.42931	10.47	.00	10.47	—10 60	.06	—10.54	—178.7	295.0	144.90	—44
—43	6.543	.45801	10.50	.00	10.50	—10 35	.07	—10.28	—178.3	314.7	154.58	—43
—42	6.971	.48797	10.52	.00	10.52	—10.11	.07	—10.04	—177.9	335.3	164.70	—42
—41	7.435	.52045	10.55	.00	10.55	—9.872	.077	—9.795	—177.4	357.6	175.65	—41

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29 921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds $\times 10^5$	Saturation Humidity Ratio: Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	Volume: v_{as} ($v_s - v_a$)	Volume: Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	Enthalpy: h_{as} ($h_s - h_a$)	Enthalpy: Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure $p_s \times 10^5$: In of Hg	Saturation Pressure: Lb per Sq In	Temp Deg F
—40	7 907	0 55349	10 57	0 00	10 57	—9 629	0 082	—9 547	—177 0	380 3	186 80	—40
—39	8 431	59017	10 60	00	10 60	—9 388	088	—9 300	—176 6	405 5	199 18	—39
—38	8 965	62755	10 62	00	10 62	—9 146	093	—9 053	—176 1	431 2	211 81	—38
—37	9 548	66836	10 65	00	10 65	—8 905	100	—8 805	—175 7	459 2	225 56	—37
—36	10 16	71120	10 67	00	10 67	—8 663	106	—8 557	—174 8	488 4	239 90	—36
—35	10 80	0 75600	10 69	0 00	10 69	—8 422	0 113	—8 309	—174 4	519 5	255 18	—35
—34	11 49	80430	10 72	00	10 72	—8 180	120	—8 060	—174 0	552 4	271 34	—34
—33	12 20	85400	10 75	00	10 75	—7 939	127	—7 812	—173 5	586 5	288 09	—33
—32	12 97	90790	10 77	00	10 77	—7 698	136	—7 562	—173 1	623 7	306 36	—32
—31	13 76	96320	10 80	00	10 80	—7 457	144	—7 313	—172 6	661 8	325 08	—31
—30	14 58	1 0206	10 82	0 00	10 82	—7 216	0 152	—7 064	—172 2	701 0	344 33	—30
—29	15 43	1 0801	10 85	00	10 85	—6 975	161	—6 814	—171 7	742 2	364 57	—29
—28	16 45	1 1515	10 87	00	10 87	—6 734	172	6 562	—171 3	791 2	388 64	—28
—27	17 49	1 2243	10 90	00	10 90	—6 493	183	—6 310	—170 9	841 0	413 10	—27
—26	18 55	1 2985	10 92	00	10 92	—6 251	194	—6 057	—170 4	892 1	438 20	—26
—25	19 68	1 3776	10 95	0 00	10 95	—6 011	0 206	—5 805	—170 4	946 4	464 87	—25
—24	20 86	1 4602	10 97	00	10 97	—5 770	219	—5 551	—170 0	1003	492 67	—24
—23	22 13	1 5491	11 00	00	11 00	—5 529	232	—5 297	—169 5	1064	522 64	—23
—22	23 42	1 6394	11 02	00	11 02	—5 288	246	—5 042	—169 0	1126	553 09	—22
—21	24 79	1 7353	11 05	00	11 05	—5 047	260	—4 787	—168 6	1192	585 51	—21
—20	26 25	1 8375	11 07	0 00	11 07	—4 807	0 276	—4 531	—168 2	1262 0	619 89	—20
—19	27 81	1 9467	11 10	00	11 10	—4 566	292	—4 274	—167 7	1337	656 73	—19
—18	29 45	2 0615	11 13	00	11 13	—4 325	310	—4 015	—167 2	1416	695 54	—18
—17	31 12	2 1784	11 15	00	11 15	—4 085	327	—3 758	—166 8	1496	734 84	—17
—16	32 95	2 3065	11 18	00	11 18	—3 844	347	—3 497	—166 3	1584	778 06	—16

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds $\times 10^5$	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure $p_s \times 10^5$: In. of Hg	Lb per Sq In.	Temp Deg F
-15	34.84	2.4388	11.20	0.01	11.21	-3.604	0.367	-3.237	-165.9	1675.0	822.76	-15
-14	36.86	2.5802	11.23	.01	11.24	-3.363	.388	-2.975	-165.4	1772.	870.41	-14
-13	38.98	2.7286	11.25	.01	11.26	-3.123	.411	-2.712	-165.0	1874.	920.51	-13
-12	41.19	2.8833	11.28	.01	11.29	-2.883	.434	-2.449	-164.5	1980.	972.58	-12
-11	43.54	3.0478	11.30	.01	11.31	-2.642	.459	-2.183	-164.0	2093.	1028.1	-11
-10	45.98	3.2186	11.33	0.01	11.34	-2.402	0.485	-1.917	-163.6	2210.0	1085.6	-10
-9	48.58	3.4006	11.35	.01	11.36	-2.162	.513	-1.649	-163.1	2335.	1147.0	-9
-8	51.25	3.5875	11.38	.01	11.39	-1.921	.541	-1.380	-162.7	2463.	1209.8	-8
-7	52.06	3.6442	11.40	.01	11.41	-1.681	.570	-1.111	-162.2	2502.	1229.0	-7
-6	57.12	3.9984	11.43	.01	11.44	-1.441	.603	-0.838	-161.7	2745.	1348.3	-6
-5	60.30	4.2210	11.45	0.01	11.46	-1.201	0.637	-0.564	-161.3	2898.0	1423.5	-5
-4	63.57	4.4499	11.48	.01	11.49	-0.960	.672	-0.288	-160 8	3055.	1500.6	-4
-3	67.05	4.6935	11.50	.01	11.51	-0.720	.709	-0.011	-160.3	3222.	1582.6	-3
-2	70.69	4.9483	11.53	.01	11.54	-0.480	.748	+0 268	-159.9	3397.	1668.6	-2
-1	74.50	5.2150	11.55	.01	11.57	-0.240	.789	+0.549	-159.4	3580.	1758.5	-1
0	78.52	5.5000	11.58	0.01	11.59	0	0.832	+0.832	-158.9	3773.0	1853.3	0
	Pounds									p_s		
0	0.0007852	5.50	11.58	0.01	11 59	0 000	0.832	0.832	-158 9	0.03773	0.01853	0
1	.0008275	5.79	11.60	.02	11.62	.240	.877	1 117	-158.5	.03975	.01963	1
2	.0008714	6.10	11.63	.01	11.64	.480	.924	1.404	-158.0	.04186	.02056	2
3	.0009179	6.43	11.65	.02	11.67	.720	.974	1.694	-157.5	.04409	.02166	3
4	.0009671	6.77	11.68	.02	11.70	.960	1.026	1.986	-157.0	.04645	.02282	4

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
5	0.001017	7.12	11.70	0.02	11.72	1.200	1.080	2.280	—156.6	0.04886	0.02400	5
6	.001071	7.50	11.73	.02	11.75	1.440	1.137	2.577	—156.1	.05144	.02527	6
7	.001127	7.89	11.75	.02	11.77	1.680	1.197	2.877	—155.6	.05412	.02658	7
8	.001186	8.30	11.78	.02	11.80	1.920	1.260	3.180	—155.1	.05692	.02796	8
9	.001247	8.73	11.80	.03	11.83	2.160	1.336	3.486	—154.7	.05988	.02941	9
10	0.001311	9.18	11.83	0.02	11.85	2.400	1.395	3.795	—154.2	0.06295	0.03092	10
11	.001379	9.65	11.86	.02	11.88	2.640	1.468	4.108	—153.7	.06618	.03251	11
12	.001450	10.15	11.88	.03	11.91	2.880	1.544	4.424	—153.2	.06958	.03418	12
13	.001523	10.66	11.91	.02	11.93	3.120	1.622	4.742	—152.7	.07309	.03590	13
14	.001600	11.20	11.93	.03	11.96	3.359	1.705	5.064	—152.2	.07677	.03771	14
15	0.001682	11.77	11.96	0.03	11.99	3.599	1.793	5.392	—151.8	0.08067	0.03963	15
16	.001766	12.36	11.98	.03	12.01	3.839	1.883	5.722	—151.3	0.8469	.04160	16
17	.001855	12.99	12.00	.04	12.04	4.079	1.979	6.058	—150.8	.08895	.04369	17
18	.001947	13.63	12.03	.04	12.07	4.319	2.078	6.397	—150.3	.09337	.04586	18
19	.002043	14.30	12.06	.03	12.09	4.559	2.182	6.741	—149.8	.09797	.04812	19
20	0.002144	15.01	12.08	0.04	12.12	4.798	2.290	7.088	—149.3	0.0128	0.05050	20
21	.002250	15.75	12.11	.04	12.15	5.038	2.405	7.443	—148.8	.1078	.05295	21
22	.002361	16.53	12.13	.05	12.18	5.278	2.524	7.802	—148.3	.1132	.05560	22
23	.002476	17.33	12.16	.04	12.20	5.518	2.648	8.166	—147.8	.1186	.05826	23
24	.002596	18.17	12.18	.05	12.23	5.758	2.778	8.536	—147.3	.1244	.06111	24
25	0.002722	19.05	12.21	0.05	12.26	5 998	2.914	8.912	—146.8	0.1304	0.06405	25
26	.002853	19.97	12.23	.06	12.29	6.237	3.055	9.292	—146.4	.1366	.06710	26
27	.002991	20.94	12.26	.06	12.32	6.477	3.205	9.682	—145.9	.1432	.07034	27

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
28	.003133	21.93	12.28	.06	12.34	6.717	3.358	10.075	—145.4	.1500	.07368	28
29	.003283	22.99	12.31	.06	12.37	6.957	3.520	10.477	—144.9	.1571	.07717	29
30	0.003439	24.07	12.33	0.07	12.40	7.197	3.689	10.886	—144.4	0.1645	0.08080	30
31	.003601	25.21	12.36	.07	12.43	7.437	3.865	11.302	—143.9	.1722	.08458	31
32	.003771	26.40	12.38	.08	12.46	7.677	4.049	11.726	—143.4	.1803	.08856	32
33	.003931	27.52	12.41	.08	12.49	7.917	4.222	12.139	+ 1.0	.1879	.09230	33
34	.004094	28.66	12.43	.08	12.51	8.157	4.399	12.556	+ 2.0	.1957	.09610	34
35	0.004262	29.83	12.46	0.08	12.54	8.397	4.582	12.979	3.0	0.20360	0.1000	35
36	.004438	31.07	12.48	.09	12.57	8.636	4.773	13.409	4.0	.21195	.1041	36
37	.004618	32.33	12.51	.09	12.60	8.876	4.969	13.845	5.0	.22050	.1083	37
38	.004803	33.62	12.53	.10	12.63	9.116	5.169	14.285	6.0	.22925	.1126	38
39	.004996	34.97	12.56	.10	12.66	9.356	5.380	14.736	7.0	.23842	.1171	39
40	0.005194	36.36	12.59	0.10	12.69	9.596	5.595	15.191	8.0	0.24778	0.1217	40
41	.005401	37.80	12.61	.11	12.72	9.836	5.821	15.657	9.1	.25755	.1265	41
42	.005616	39.31	12.64	.11	12.75	10.08	6.05	16.13	10.1	.26773	.1315	42
43	.005840	40.88	12.66	.12	12.78	10.32	6.30	16.62	11.1	.27832	.1367	43
44	.006069	42.48	12.69	.12	12.81	10.56	6.55	17.11	12.1	.28911	.1420	44
45	0.006306	44.14	12.71	0.13	12.84	10.80	6.81	17.61	13.1	0.30031	0.1475	45
46	.006553	45.87	12.74	.13	12.87	11.04	7.08	18.12	14.1	.31191	.1532	46
47	.006808	47.66	12.76	.14	12.90	11.28	7.36	18.64	15.1	.32393	.1591	47
48	.007072	49.50	12.79	.14	12.93	11.52	7.64	19.16	16.1	.33635	.1652	48
49	.007345	51.42	12.81	.15	12.96	11.76	7.94	19.70	17.1	.34917	.1715	49

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
50	0.007626	53.38	12.84	0.15	12.99	12.00	8.25	20.25	18.1	0.36241	0.1780	50
51	.007921	55.45	12.86	.16	13.02	12.23	8.57	20.80	19.1	.37625	.1848	51
52	.008226	57.58	12.89	.17	13.06	12.47	8.91	21.38	20.1	.39051	.1918	52
53	.008534	59.74	12.91	.18	13.09	12 71	9.24	21.95	21.1	.40496	.1989	53
54	.008856	61.99	12.94	.18	13.12	12.95	9.60	22.55	22.1	.42003	.2063	54
55	0.009192	64.34	12.96	0.19	13.15	13.19	9.96	23.15	23.1	0.43570	0.2140	55
56	.009536	66.75	12.99	.20	13.19	13.43	10.34	23.77	24.1	.45179	.2219	56
57	.009890	69.23	13.01	.21	13.22	13.67	10.73	24.40	25.1	.46828	.2300	57
58	.01026	71.82	13.04	.21	13.25	13.91	11.14	25.05	26.1	.48538	.2384	58
59	.01064	74.48	13.06	.23	13.29	14.15	11.55	25 70	27.1	.50310	.2471	59
60	0.01103	77.21	13.09	0.23	13.32	14.39	11.98	26.37	28.1	0.52142	0.2561	60
61	.01144	80.08	13.11	.24	13.35	14.63	12.43	27.06	29.1	.54035	.2654	61
62	.01186	83.02	13.14	.25	13.39	14.87	12.89	27.76	30.1	.55970	.2749	62
63	.01229	86.03	13.16	.26	13.42	15 11	13.37	28.48	31.1	.57985	.2848	63
64	.01274	89.18	13.19	.27	13 46	15.35	13.86	29.21	32.1	.60042	.2949	64
65	0.01320	92.40	13.21	0.28	13.49	15.59	14.37	29.96	33.1	0.62179	0.3054	65
66	.01368	95.76	13.24	.29	13.53	15.83	14.90	30.73	34.1	.64378	.3162	66
67	.01417	99.19	13.26	.31	13.57	16.07	15.44	31.51	35.1	.66638	.3273	67
68	.01468	102.8	13.29	.31	13.60	16.31	16 00	32.31	36.0	.68980	.3388	68
69	.01520	106.4	13.31	.33	13.64	16.55	16.57	33.12	37.0	.71382	.3506	69
70	0.01574	110.2	13.34	0.34	13.68	16.79	17.17	33.96	38.0	0.73866	0.3628	70
71	.01631	114.2	13.37	.34	13.71	17.03	17.80	34.83	39.0	.76431	.3754	71
72	.01688	118.2	13.40	.35	13.75	17.27	18.43	35.70	40.0	.79058	.3883	72
73	.01748	122.4	13.42	.37	13.79	17.51	19.09	36.60	41.0	.81766	.4016	73

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
74	.01809	126.6	13.44	.39	13.83	17.75	19.76	37.51	42.0	.84555	.4153	74
75	0.01873	131.1	13.47	0.40	13.87	17.99	20.47	38.46	43.0	0.87448	0.4295	75
76	.01938	135.7	13.49	.42	13.91	18.23	21.19	39.42	44.0	.90398	.4440	76
77	.02005	140.4	13.52	.43	13.95	18.47	21.93	40.40	45.0	.93452	.4590	77
78	.02075	145.3	13.54	.45	13.99	18.71	22.71	41.42	46.0	.96588	.4744	78
79	.02147	150.3	13.57	.46	14.03	18 95	23.51	42.46	47.0	.99825	.4903	79
80	0.02221	155.5	13.59	0.49	14.08	19.19	24.32	43.51	48.0	1.0316	0.5067	80
81	.02298	160.9	13.62	.50	14.12	19.43	25.18	44.61	49.0	1.0661	.5236	81
82	.02377	166.4	13.64	.52	14.16	19.67	26.05	45.72	50.0	1.1013	.5409	82
83	.02459	172.1	13.67	.54	14.21	19.91	26.97	46.88	51.0	1.1377	.5588	83
84	.02543	178.0	13.69	.57	14.26	20.15	27.90	48.05	52.0	1.1752	.5772	84
85	0.02629	184.0	13.72	0.58	14.30	20 39	28.85	49.24	53.0	1.2135	0.5960	85
86	.02718	190.3	13.74	.60	14.34	20.63	29.84	50.47	54.0	1.2527	.6153	86
87	.02810	196.7	13.77	.62	14.39	20.87	30.87	51.74	55.0	1.2933	.6352	87
88	.02904	203.3	13.79	.65	14.44	21.11	31.91	53.02	56.0	1.3346	.6555	88
89	.03002	210.1	13.82	.66	14.48	21.35	33.00	54.35	57.0	1.3774	.6765	89
90	0.03102	217.1	13.84	0.69	14.53	21.59	34 11	55.70	58 0	1 4211	0.6980	90
91	.03205	224.4	13.87	.71	14.58	21.83	35.26	57.09	59.0	1.4661	.7201	91
92	.03312	231.8	13.89	.74	14.63	22.07	36.45	58.52	60.0	1.5125	.7429	92
93	.03421	239.5	13.92	.77	14.69	22.32	37.67	59.99	61.0	1.5600	.7662	93
94	.03535	247.5	13.94	.79	14.73	22.56	38.94	61.50	62.0	1.6088	.7902	94
95	0.03652	255.6	13.97	0.82	14.79	22.80	40.25	63 05	63.0	1.6591	0.8149	95

[a] Compiled by W. M. Sawdon and extended by John A. Goff.

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
96	.03772	264.0	13.99	.85	14.84	23.04	41.58	64.62	64.0	1.7108	.8403	96
97	.03896	272.7	14.02	.88	14.90	23.28	42.97	66.25	65.0	1.7638	.8663	97
98	.04024	281.7	14.04	.91	14.95	23.52	44.40	67.92	66.0	1.8181	.8930	98
99	.04156	290.9	14.07	.94	15.01	23.76	45.87	69.63	67.0	1.8741	.9205	99
100	0.04293	300.5	14.10	0.97	15.07	24.00	47.40	71.40	68.0	1.9316	0.9487	100
101	.04433	310.3	14.12	1.00	15.12	24.24	48.97	73.21	69.0	1.9904	.9776	101
102	.04577	320.4	14.15	1.03	15.18	24.48	50.58	75.06	70.0	2.0507	1.0072	102
103	.04726	330.8	14.17	1.08	15.25	24.72	52.25	76.97	71.0	2.1128	1.0377	103
104	.04879	341.5	14.20	1.11	15.31	24.96	53.96	78.92	72.0	2.1763	1.0689	104
105	0.05037	352.6	14.22	1.15	15.37	25.20	55.73	80.93	73.0	2.2414	1.1009	105
106	.05200	364.0	14.25	1.19	15.44	25.44	57.56	83.00	74.0	2.3084	1.1338	106
107	.05368	375.8	14.27	1.23	15.50	25.68	59.45	85.13	75.0	2.3770	1.1675	107
108	.05541	387.9	14.30	1.27	15.57	25.92	61.38	87.30	76.0	2.4473	1.2020	108
109	.05719	400.3	14.32	1.32	15.64	26.16	63.38	89.54	76.9	2.5196	1.2375	109
110	0.05904	413.3	14.35	1.36	15.71	26.40	65.46	91.86	77.9	2.5939	1.274	110
111	.06092	426.4	14.37	1.41	15.78	26 64	67.57	94.21	78.9	2.6692	1.311	111
112	.06292	440.4	14.39	1.46	15.85	26.88	69.82	96.70	79.9	2.7486	1.350	112
113	.06493	454.5	14.42	1.51	15.93	27.12	72.08	99.20	80.9	2.8280	1.389	113
114	.06700	469.0	14.45	1.55	16.00	27.36	74.40	101.76	81.9	2.9094	1.429	114
115	0.06913	483.9	14.47	1.61	16.08	27.60	76.80	104.40	82.9	2.9929	1.470	115
116	.07134	499.4	14.50	1.66	16.16	27.84	79.29	107.13	83.9	3.0784	1.512	116
117	.07361	515.3	14.52	1.72	16.24	28.08	81.84	109.92	84.9	3.1660	1.555	117
118	.07600	532.0	14.55	1.77	16.32	28.32	84.53	112.85	85.9	3.2576	1.600	118

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THERMODYNAMIC PROPERTIES OF MOIST AIR [a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
119	.07840	548.8	14.57	1.84	16.41	28.56	87.24	115.80	86.9	3.3492	1.645	119
120	0.08093	566.5	14.60	1.90	16.50	28.80	90.09	118.89	87.9	3.4449	1.692	120
121	.08348	584.4	14.62	1.96	16.58	29.04	92.97	122.01	88.9	3.5406	1.739	121
122	.08616	603.1	14.65	2.03	16.68	29.28	95.99	125.27	89.9	3.6404	1.788	122
123	.08892	622.4	14.67	2.10	16.77	29.52	99.11	128.63	90.9	3.7422	1.838	123
124	.09175	642.3	14.70	2.17	16.87	29.76	102.30	132.06	91.9	3.8460	1.889	124
125	0.09466	662.6	14.72	2.24	16.96	30.00	105.59	135.59	92.9	3.9519	1.941	125
126	.09770	683.9	14.75	2.31	17.06	30.24	109.02	139.26	93.9	4.0618	1.995	126
127	.1008	705.6	14.77	2.40	17.17	30.48	112.53	143.01	94.9	4.1718	2.049	127
128	.1040	728.0	14.80	2.47	17.27	30.72	116.15	146.87	95.9	4.2858	2.105	128
129	.1074	751.8	14.83	2.55	17.38	30.96	120.00	150.96	96.9	4.4039	2.163	129
130	0.1107	774.9	14.85	2.64	17.49	31.20	123.73	154.93	97.9	4.5220	2.221	130
131	.1143	800.1	14.88	2.73	17.61	31.45	128.81	159.26	98.9	4.6441	2.281	131
132	.1180	826.0	14.90	2.83	17.73	31.69	131.99	163.68	99.9	4.7703	2.343	132
133	.1218	852.6	14.93	2.92	17.85	31.93	136.31	168.24	100.9	4.8986	2.406	133
134	.1257	879.9	14.95	3.02	17.97	32.17	140.72	172.89	101.9	5.0289	2.470	134
135	0.1297	907.9	14.98	3.12	18.10	32.41	145.26	177.67	102.9	5.1633	2.536	135
136	.1339	937.3	15.00	3.23	18.23	32.65	150.02	182.67	103.9	5.2997	2.603	136
137	.1382	967.4	15.03	3.33	18.36	32.89	154.91	187.80	104.9	5.4402	2.672	137
138	.1427	998.9	15.05	3.45	18.50	33.13	160.01	193.14	105.9	5.5827	2.742	138
139	.1473	1031.1	15.08	3.57	18.65	33.37	165.24	198.61	106.9	5.7293	2.814	139
140	0.1521	1064.7	15.10	3.69	18.79	33.61	170.69	204.30	107.9	5.8779	2.887	140

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
141	.1570	1099.0	15.13	3.81	18.94	33.85	176.26	210.11	108.9	6.0306	2.962	141
142	.1622	1135.4	15.15	3.95	19.10	34.09	182.17	216.26	109.9	6.1874	3.039	142
143	.1675	1172.5	15.18	4.08	19.26	34.33	188.20	222.53	110.9	6.3482	3.118	143
144	.1730	1211.0	15.20	4.23	19.43	34.57	194.45	229.02	111.9	6.5111	3.198	144
145	0.1787	1250.9	15.23	4.37	19.60	34.81	200.95	235.76	112.9	6.6781	3.280	145
146	.1846	1292.2	15.25	4.53	19.78	35.05	207.66	242.71	113.9	6.8471	3.363	146
147	.1908	1335.6	15.28	4.68	19.96	35.29	214.73	250.02	114.9	7.0222	3.449	147
148	.1971	1379.7	15.30	4.85	20.15	35.53	221.90	257.43	115.9	7.1993	3.536	148
149	.2037	1425.9	15.33	5.02	20.35	35.77	229.43	265.20	116.9	7.3805	3.625	149
150	0.2105	1473.5	15.35	5.20	20.55	36.02	237.17	273.19	117.9	7.5658	3.716	150
151	.2176	1523.2	15.38	5.38	20.76	36.26	245.28	281.54	118.9	7.7551	3.809	151
152	.2250	1575.0	15.40	5.57	20.97	36.50	253.71	290.21	119.9	7.9485	3.904	152
153	.2327	1628.9	15.43	5.77	21.20	36.74	262.51	299.25	120.9	8.1460	4.001	153
154	.2407	1684.9	15.45	5.98	21.43	36.98	271.63	308.61	121.9	8.3476	4.100	154
155	0.2490	1743.0	15.48	6.19	21.67	37.22	281.12	318.34	122.9	8.5532	4.201	155
156	.2577	1803.9	15.50	6.43	21.93	37.46	291.05	328.51	123.9	8.7650	4.305	156
157	.2667	1866.9	15.53	6.66	22.19	37.70	301.24	339.04	124.9	8.9788	4.410	157
158	.2761	1932.7	15.56	6.90	22.46	37.94	312.08	350.02	125.9	9.1986	4.518	158
159	.2858	2000.6	15.58	7.16	22.74	38.18	323.18	361.36	126.9	9.4206	4.627	159
160	0.2961	2072.7	15.61	7.42	23.03	38.43	334.95	373.38	127.9	9.6486	4.739	160
161	.3067	2146.9	15.63	7.70	23.33	38.67	347.09	385.76	128.9	9.8807	4.853	161
162	.3179	2225.3	15.66	7.99	23.65	38.91	359.89	398.80	129.9	10.119	4.970	162
163	.3295	2306.5	15.68	8.30	23.98	39.15	373.19	412.34	130.9	10.361	5.089	163
164	.3416	2391.2	15.71	8.62	24.33	39.39	387.03	426.42	131.9	10.608	5.210	164

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	h_{as} $(h_s - h_a)$	Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Lb per Sq In.	Temp Deg F
165	0.3544	2480.8	15.73	8.96	24.69	39.63	401.71	441.34	132.9	10.860	5.334	165
166	.3677	2573.9	15.76	9.31	25.07	39.87	416.94	456.81	133.9	11.117	5.460	166
167	.3817	2671.9	15.78	9.68	25.46	40.11	433.00	473.11	134.9	11.379	5.589	167
168	.3964	2774.8	15.81	10.07	25.88	40.35	449.83	490.18	135.9	11.646	5.720	168
169	.4118	2882.6	15.83	10.48	26.31	40.59	467.52	508.11	136.9	11.919	5.854	169
170	0.4280	2996.0	15.86	10.91	26.77	40.83	486.08	526.91	137.9	12.196	5.990	170
171	.4451	3115.7	15.88	11.36	27.24	41.07	505.72	546.79	138.9	12.480	6.130	171
172	.4631	3241.7	15.91	11.83	27.74	41.32	526.36	567.68	139.9	12.770	6.272	172
173	.4821	3374.7	15.93	12.35	28.28	41.56	548.20	589.76	140.9	13.065	6.417	173
174	.5022	3515.4	15.96	12.88	28.84	41.80	571.25	613.05	141.9	13.366	6.565	174
175	0.5235	3664.5	15.98	13.45	29.43	42.04	595.74	637.78	142.9	13.674	6.716	175
176	.5459	3821.3	16.01	14.04	30.05	42.28	621.45	663.73	143.9	13.985	6.869	176
177	.5697	3987.9	16.03	14.68	30.71	42.52	648.83	691.35	144.9	14.303	7.025	177
178	.5949	4164.3	16.06	15.35	31.41	42.76	677.77	720.53	145.9	14.627	7.184	178
179	.6215	4350.5	16.08	16.07	32.15	43.00	708.39	751.39	146.9	14.954	7.345	179
180	0.6501	4550.7	16.11	16.83	32.94	43.24	741.24	784.48	147.9	15.290	7.510	180
181	.6805	4763.5	16.13	17.65	33.78	43.49	776.25	819.74	148.9	15.632	7.678	181
182	.7131	4991.7	16.16	18.52	34.68	43.73	813.72	857.45	149.9	15.981	7.849	182
183	.7481	5236.7	16.18	19.47	35.65	43.97	854.03	898.00	150.9	16.337	8.024	183
184	.7854	5497.8	16.21	20.46	36.67	44.21	896.93	941.14	151.9	16.697	8.201	184
185	0.8258	5780.6	16.23	21.55	37.78	44.45	943.48	987.93	152.9	17.066	8.382	185
186	.8693	6085.1	16.26	22.72	38.98	44.69	993.52	1038.21	153.9	17.440	8.566	186
187	.9162	6413.4	16.28	23.99	40.27	44.93	1047.58	1092.51	154.9	17.821	8.753	187

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THERMODYNAMIC PROPERTIES OF MOIST AIR[a], 29.921 IN. HG (*continued*)

Temp Deg F	Saturation Humidity Ratio W_s Weight of Water per Lb of Dry Air: Pounds	Saturation Humidity Ratio: Grains	Volume Cu Ft per Lb Dry Air: Dry Air v_a	Volume: v_{as} $(v_s - v_a)$	Volume: Saturated Mixture v_s	Enthalpy Btu per Lb Dry Air: Dry Air h_a	Enthalpy: h_{as} $(h_s - h_a)$	Enthalpy: Saturated Mixture h_s	Specific Enthalpy of Solid Water Btu per Lb h'_w	Saturation Pressure p_s: In. of Hg	Saturation Pressure: Lb per Sq In.	Temp Deg F
188	.9673	6771.1	16.31	25.36	41.67	45.18	1106.40	1151.58	155.9	18.210	8.944	188
189	1.0227	7158.9	16.34	26.70	43.04	45.42	1170.62	1216.04	157.0	18.605	9.138	189
190	1.083	7581.0	16.36	28.49	44.85	45.66	1239.71	1285.37	158.0	19.008	9.336	190
191	1.150	8050.0	16.39	30.29	46.68	45.90	1316.98	1362.88	159.0	19.419	9.538	191
192	1.224	8568.0	16.41	32.29	48.70	46.14	1402 21	1448.35	160.0	19.839	9.744	192
193	1.306	9142.0	16.44	34.49	50.93	46.38	1496 81	1543.19	161.0	20.266	9.954	193
194	1.397	9779.0	16.46	36.96	53.42	46.62	1601.66	1648.28	162.0	20.702	10.168	194
195	1.499	10493.0	16.49	39.71	56.20	46.86	1719.35	1766.21	163.0	21.144	10.385	195
196	1.613	11291.0	16.51	42.80	59.31	47.10	1850.76	1897.86	164.0	21.592	10.605	196
197	1.742	12194.0	16.54	46.31	62.85	47.34	1999.64	2046.98	165.0	22.048	10.829	197
198	1.890	13230.0	16.56	50.32	66.88	47.59	2170.29	2217.88	166.0	22.512	11.057	198
199	2.061	14427.0	16.59	54.95	71 54	47.83	2367.68	2415.51	167.0	22.984	11.289	199
200	2.261	15827.0	16.61	60.38	76.99	48.07	2598.34	2646.41	168.0	23.465	11.525	200

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152. Compilation of a Psychrometric Table

Tabulated values in psychrometric table 93 come under the following headings:

1. *Temperature, ° F.*

2. *Saturation humidity* ratio W_s = weight of water vapor per pound of dry air in fractions of a pound or in grains.

3. *Volume, cubic feet per pound dry air*

   (a) Dry air = v_a

   (b) Vapor = $v_{as} = (v_s - v_a)$

   (c) Saturated mixture = v_s

4. *Enthalpy, Btu per pound dry air*

   (a) Dry air enthalpy, h_a, in Btu per pound dry air generally termed sensible heat or "dry heat."

   (b) Water vapor enthalpy, h_{as} or $(h_s - h_a)$, in Btu per pound dry air, sometimes termed "moist heat."

   (c) Saturated mixture enthalpy, h_s, in Btu per pound dry air.

5. *Specific enthalpy of liquid water,* h'_w, in Btu per pound.

6. *Saturation pressure,* P_s

   (a) Given in inches of mercury (Hg)

   (b) Given in pounds per square inch

Note: The *A.S.H.V.E. Guide* gives the following definitions of enthalpy and latent heat. However the old expression "total heat" is used in this text from force of habit or because it seems more expressive.

LATENT HEAT: The most general interpretation is heat absorbed at constant temperature. More specifically the *latent heat of vaporization* is the difference between the specific enthalpies of saturated vapor and saturated liquid at the same temperature (and, for a pure substance, the same pressure). *Latent heat of sublimation* is the difference between the specific enthalpies of saturated vapor and saturated solid at the same temperature. *Latent heat of fusion* is the difference between the specific enthalpies of saturated liquid and saturated solid at the same temperature.

ENTHALPY: A thermodynamic property which serves as a measure of the quantity of thermal energy *convected* by a fluid in steady flow. In a non-flow process the increase of enthalpy equals the quantity of heat absorbed *provided pressure is constant.* Enthalpy was formerly called heat content, sometimes total heat. Specific enthalpy is the ratio of total enthalpy to total weight, that is, enthalpy per unit weight of substance, Btu per pound.

153. Saturation, Humidity Ratio

The weight ratio W_s is a convenient term to express the composition of moist air. Humidity ratio has been known by other names: absolute humidity, density ratio, proportionate humidity, mixing ratio. Humidity ratio is most suggestive of the meaning which it is desired to express, i.e., the weight of water vapor mixed with a unit weight of dry air.

$$W_s = 0.62193\left(\frac{p_s}{p - p_s}\right) \text{ pounds of water per pound dry air at saturation} \tag{1}$$

$$\frac{p_s}{p} = \frac{W_s}{0.62193 + W_s} \tag{2}$$

where

p_s = saturation pressure of pure water in inches of mercury

p = vapor pressure of the mixture, in inches of mercury

If p is assumed to be 29.92 in. Hg at atmospheric pressure and p_s the vapor pressure in inches of mercury at saturation

$$W_s = \frac{p_s}{48 - 1.6p_s} = \frac{0.62193p_s}{29.92 - p_s} \text{ pounds water per pound dry air at saturation} \tag{3}$$

$$W_s = \frac{4354p_s}{29.92 - p_s} = \text{grains water per pound dry air at saturation} \tag{4}$$

The gas constant for dry air when the partial pressure of the air is expressed in inches of mercury is 0.753, so that the humidity ratio or specific humidity is

$$W_s = \frac{p_s}{1.21(t + 460)} \div \frac{29.92 - p_s}{0.753(t + 460)}$$

154. Specific Volume; Cubic Feet per Pound Dry Air

The volume in cubic feet of one pound of dry air at atmospheric pressure is based on the laws of Charles and Gay-Lussac.

$$v_a = \frac{t + 460}{39.64} = \text{cubic feet per pound dry air} \qquad (5)$$

where

$t =$ temperature, ° F

the reciprocal of volume v_a is weight density d expressed in pounds per cubic foot.

155. Specific Volume of Water Vapor

The values of the specific volume of water vapor at corresponding pressures and temperatures can be calculated using the basic formula of Charles and Gay-Lussac. Usually, the information is given in steam tables. It is expressed by the equation

$$v = \frac{460 + t}{1.684p} = \text{cubic feet per pound vapor} \qquad (6)$$

where

$p =$ pounds per square inch
$t =$ temperature ° F

at atmospheric pressure $v = 18.5 + 0.04\ t$ approximately.

156. Weight of Moisture per Cubic Foot of Air

Humidity is the water vapor in the atmosphere. Water vapor, either saturated or superheated, at partial pressures lower than 4 in. Hg, may be treated as a gas with a gas constant R of 1.21 (when pressures are expressed in in. Hg) in the equation

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

Let

$$d = \frac{W}{V} = \frac{p_w}{1.21(t + 460)} \text{ pounds per cubic foot}$$

$$= \frac{5785p_w}{t + 460} \text{ grains per cubic foot} \qquad (7)$$

where

$p_w =$ actual partial pressure of vapor in inches of mercury
$t =$ dry-bulb temperature, ° F

157. Relative Humidity

Relative humidity (ϕ) is defined as the ratio of the partial pressure p_w of the water vapor in the air to the saturated pressure p_s of pure water corresponding to the actual dry-bulb temperature.

$$\phi = \frac{p_w}{p_s} \tag{8}$$
$$p_w = \phi p_s$$

The ratio of actual humidity ratio W to the saturation humidity ratio W_s corresponding to the actual temperature at the observed pressure is denoted by the Greek letter mu, (μ) and may be called degree of saturation or percent saturation

$$\mu = \frac{W}{W_s} \text{ which approximates } \phi \tag{9}$$

The relative humidity ϕ of a given mixture at a given temperature is not exactly the same as the degree of saturation μ at the same temperature. The relation between μ and ϕ is obtained directly from formula (1). Under Dalton's law the water vapor exerts a partial pressure p_w.

$$\mu = \frac{W}{W_s} = \frac{(p - p_s)p_w}{(p - p_w)p_s} = \left(\frac{p - p_s}{p - p_w}\right)\phi$$

At ordinary temperatures p_s and p_w are negligible and therefore μ is approximately equal to ϕ.
Referring to formula (3)

$$W_s = \frac{0.62193p_s}{29.92 - p_s} = \text{pounds water per pound dry air at saturation}$$

and

$$W_\phi = \frac{0.62193p_s\phi}{29.92 - p_s\phi} = \text{pounds moisture per pound dry air at } \phi \text{ relative humidity} \tag{10}$$

Problem. **Air is to be maintained at 80° F and 50% saturation when the outside air is at 90° F and 37% saturation. The ob-**

served pressure is 29.921 in. Hg. Using psychrometric Table 93, find the weight of water to be condensed per pound of dry air by a cooling process.

Solution.

0.0221 = pounds of moisture per pound dry air at 80° F saturated.

$0.50 \times 0.02221 = 0.0111$ pounds of moisture per pound dry air at 50% saturation.

0.03102 = pounds moisture per pound dry air saturated with moisture at 90° F.

$0.37 \times 0.03102 = 0.01148$ = pounds moisture per pound dry air at 37% saturation and 90° F.

$0.01148 - 0.0111 = 0.00038$ pound moisture per pound of dry air to be removed by cooling.

Problem. When outside air is at 0° F and 70% saturation, find the quantity of water to be added to a pound of dry air to maintain an inside room condition of 70° F dry-bulb and 40% saturation. The barometric pressure is assumed to be 29.921 in. Hg.

$$0.40 \times 0.01574 = 0.006296$$

where 0.01574 = pounds moisture per pound dry air at 70° F (saturated) (from table)

$$0.70 \times 0.0007852 = 0.00055$$

where 0.0007852 = pounds moisture per pound dry air at 0° F (from table)

$0.006296 - 0.00055 = 0.005746$ pound water added per pound dry air.

158. Measurement of Relative Humidity

The dew point apparatus for humidity measurements consists of a polished plated container cooled by the evaporation of a volatile liquid, e.g., ether. The temperature at which the first slight water vapor forms is the dew point. If the temperature is below 32° F the deposit will appear as frost. Another method of determining humidity is a chemical one; the water vapor is absorbed by anhydrous calcium chloride or some other drying agent, and weighed on a chemical balance. The sling psychrometer (see Figure 11) is the favorite and most practical instrument used in air-conditioning tests.

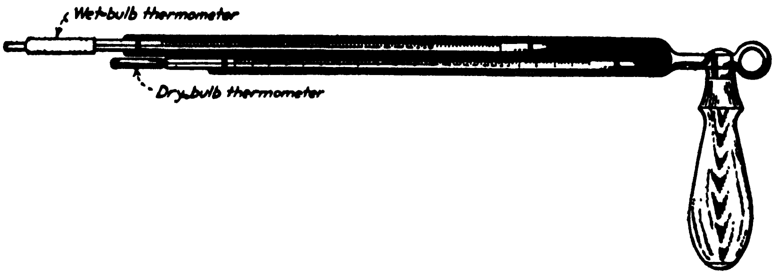


FIGURE 11
Sling Psychrometer

159. Dew Point Temperature

The temperature corresponding to saturation (100% relative humidity) for a given moisture content is called the dew point. If moist air is cooled at constant humidity ratio W and constant observed pressure P, the dew point will be reached when the air just becomes saturated and formation of a liquid (or solid) phase begins.

Problem. In the preceding problem find the dew point of inside and outside air.

Solution. The given humidity ratio of inside 80° F air temperature is 0.50×0.02221 or 0.0111 lb of moisture per pound of dry air, which is the saturation value between 60 and 61° F as given in the table. Therefore the dew point may be determined by interpolation. The given humidity ratio of the outside air 90° F, 37% saturation is 0.01148 lb moisture per pound dry air, which corresponds to a saturation value between 61 and 62° F. Again the dew point may be found by interpolation.

Problem (heating). It is desired to maintain a room condition of 70° F with a relative humidity of 40% when the outside air is at a condition of 0° F and 70% relative humidity. The barometric pressure is 29.92 in. Hg. Find the humidity ratio, weight in pounds of water vapor added to each pound of dry air, and the dew point temperature of the humidified air.

Solution. If a complete psychrometric table is not available, but a table of pressure-temperature relations of water vapor is, use equation (10).

$$W_s = \frac{0.62193 p_s \phi}{29.92 - p_s \phi}$$

$$\phi = 0.7$$

$$W_a = 0.622 \left(\frac{0.7 \times 0.0377}{29.92 - 0.0264}\right) = 0.000549 \text{ lb of water vapor per pound dry air at } 0° \text{ F.}$$

$$W_b = 0.622 \left(\frac{0.4 \times 0.738}{29.92 - 0.295}\right) = 0.00619 \text{ lb of water vapor or moisture per pound dry air at } 70° \text{ F.}$$

$$W_b - W_a = 0.00564 \text{ lb moisture added per pound dry air}$$

According to the psychrometric table, the temperature corresponding to 0.00619 lb moisture at saturation is 44.5° F; this is the dew point.

This result is approximated by referring to the psychrometric table

$$W_a = 0.7 \times 0.000785 = 0.000549 \text{ lb per pound dry air}$$
$$W_b = 0.4 \times 0.01574 = 0.006296 \text{ lb per pound dry air}$$

The water vapor added per pound of dry air is approximately 0.005746 lb. The dew point corresponding to 0.006296 is approximately 44.95° F.

Problem (cooling). Air entering the apparatus is at a temperature of 84° F dry-bulb, 70° F wet-bulb, i.e., at a relative humidity of 50%, at standard barometric pressure of 29.92 in. Hg. Find the dew point of the entering air and the weight in pounds of the vapor condensed if the air is cooled to 54° F.

Solution.

$$W_a = 0.622\left(\frac{0.5 \times 1.175}{29.92 - 0.588}\right) = 0.01248 \text{ lb per pound of dry air entering}$$

$$W_b = 0.622\left(\frac{0.42003}{29.92 - 0.42}\right) = 0.00887 \text{ lb per pound of dry air when cooled}$$

Inspection of the psychrometric table will show that 0.01248 lb of moisture corresponds to a saturation temperature of 63.5° F which is the dew point. The weight of water condensed is 0.01248 — 0.00887 or 0.00361 lb per pound of dry air.

160. Enthalpy; Sensible Heat of Air

Recent developments have made it possible to calculate the zero-pressure specific heat of air. The values indicate that 0.24 Btu per pound can be used in ordinary air conditioning calculations. As enthalpy is purely relative, any convenient state can be selected to which the value zero may be assigned. In general, most psychrometric tables are based on 0° F and 29.921 in. Hg.

$$h_a = 0.24t \tag{11}$$

where

h_a = Btu required to raise or lower one pound of dry air t degrees F.

t = temperature ° F, of dry air.

0.24 = specific heat of dry air.

161. Specific Enthalpy of Water Vapor

The reference point for water is usually chosen at 32° F to conform to steam table practice. For ordinary calculations the mean specific heat of saturated steam is assigned a value of 0.444 Btu per pound. The following equation is used in the solution of psychrometric problems

$$h^0_w = 0.444t + 1061 \tag{12}$$

where 0.444 t = sensible heat per lb steam from zero to t degrees F, in Btu

and 1061 = latent heat of vaporization of 1 lb water at partial atmospheric pressure of 0.03773 in. Hg at 0° F, in Btu
$h^0{}_w$ = total heat per pound saturated steam from 0° F to t, in Btu

From equation (3) $W_s = \frac{p_s}{48 - 1.6p_s} =$ pounds moisture per pound dry air at saturation

p_s = pressure, in. Hg at saturation

$h_{as} = W_s \times h^0{}_w$ = total heat in moisture per pound dry air at saturation. (13)

162. Enthalpy of Saturated Mixture

The enthalpy of a saturated mixture is the sum of the enthalpy of dry air and the specific enthalpy of water vapor expressed below:

$$h_s = h_a + h_{as} \tag{14}$$

$$h_s = 0.24t + W'_s \times h^0{}_w \tag{15}$$

where

h_s = enthalpy of saturated mixture, Btu per pound dry air plus the moisture required to saturate it.
$h^0{}_w$ = total heat in 1 lb saturated steam from zero to t degrees F.
h_a = 0.24t = sensible heat to raise 1 lb dry air to temperature t from 0° F.
$h_{as} = h_s - h_a$

163. Enthalpy of an Unsaturated Mixture

Dalton's law implies that the total heat of dry air and water vapor are independent of pressure and that the enthalpy h of an air mixture that is not saturated is expressed by a simple equation

$$h = h_a + \mu\, h_{as} \tag{16}$$

$$h = h_a + \mu\,(w_s h_w) \tag{17}$$

where

μ = per cent saturation
w_s = pounds moisture per pound dry air
h_s = enthalpy of mixture per pound dry air
h_a† = enthalpy of dry air in Btu per pound at given temperature and pressure
h_w = enthalpy per pound of vapor
h_{as}* = $h_s - h_a$

Problem. Find the enthalpy per pound of dry air at 90° F, 35.7% saturation and 29.921 in. Hg.

Solution. Using psychrometric table and referring to formula (16)

$h_s = 21.59 + (0.357 \times 34.11) = 33.76$ Btu per pound dry air.

164. The Energy Equation

This equation applies to heat transfer during various air-conditioning processes, and is conveniently explained by means of a diagram which may represent any type of air-conditioning apparatus. By the proper choice of the direction of arrows heating effects in humidifiers and driers and cooling effects in dehumidifiers and cooling towers, may be illustrated. The adiabatic saturation processes may also be visualized by referring to the same diagram and equations.

Assume that a mixture of air and water enters the apparatus at 1–1 and leaves at 3–3. Water is injected at a temperature t_2 and no other form of energy is delivered to or extracted from the steadily flowing air-vapor mixture. The following formulas give heating and cooling effects.

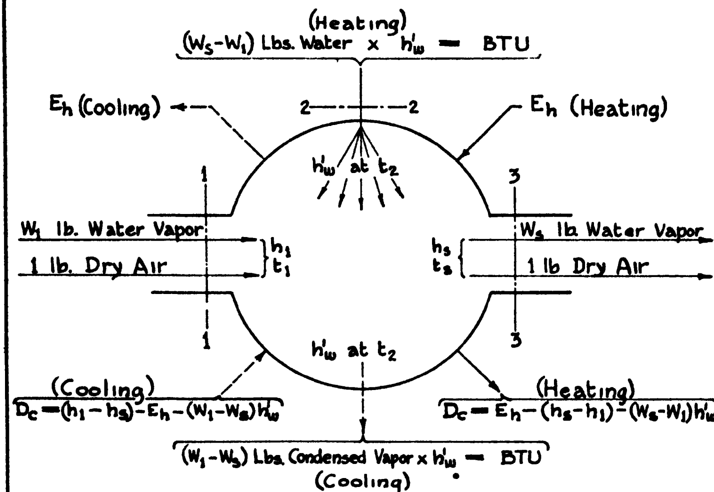
$$h_1 + E_h + (W_s - W_1)\,h'_w = h_s + D_c \qquad \text{heating} \quad (18)$$

$$E_h - D_c = h_s - h_1 - (W_s - W_1)\,h'_w \qquad \text{heating} \quad (19)$$

$$E_h + D_c = h_1 - h_s - (W_1 - W_s)\,h'_w \qquad \text{cooling} \quad (20)$$

† h_a is generally called *sensible heat* or *dry heat.*
* h_{as} is generally called *latent heat* or *moist heat.*

DIAGRAM OF TYPICAL AIR CONDITIONING APPARATUS – ENERGY EQUATION ILLUSTRATING HUMIDIFYING (HEATING) & DEHUMIDIFYING (COOLING)



Equation – Heating: $E_h - D_c = (h_s - h_1) - (W_s - W_1)\, h'_w$
= Net heat supplied per lb. dry air.

$(h_s - h_1)$ = Increase in enthalpy of mixture passing thru apparatus in heating.

Equation – Cooling: $E_h + D_c = (h_1 - h_s) - (W_1 - W_s)\, h'_w$
= Net heat supplied per lb. dry air.

$(h_1 - h_s)$ = Decrease in enthalpy of mixture passing thru apparatus in cooling.

SYMBOLS

W_1 = Weight of water entering per lb. of dry air.
W_s = Weight of water leaving per lb of dry air.
h'_w = Enthalpy of water above 32°F. at temperature t_2 in BTU per lb. water.
h_1 = Enthalpy of air-vapor mixture entering in BTU per lb dry air.
h_s = Enthalpy of air-vapor mixture leaving in BTU per lb. dry air.
t_1 = Dry bulb temperature air entering.
t_s = Dry bulb temperature air leaving.
D_c = Quantity of heat exchanged in apparatus – e.g. heat lost externally in heating effects; heat gained externally in cooling effects.
E_h = Quantity of heat supplied – BTU per lb dry air (Heating).
= Quantity of heat extracted – BTU per lb. dry air (Cooling).

EQUATIONS

$h_1 = 0.24\, t_1 + (0.444\, t_1 + 1061)\, W_1$
$h_s = 0.24\, t_s + (0.444\, t_s + 1061)\, W_s$
$h'_w = (W_s - W_1)(t_2 - 32)$ for heating; $(W_1 - W_s)(t_2 - 32)$ for cooling.

HH

FIGURE 12

where

E_h = Btu supplied per pound dry air

D_c = Btu lost per pound dry air in heating, gained in cooling, externally by heat transfer from the apparatus.

W'_1 = weight of water entering apparatus per pound dry air

W'_s = weight of water leaving apparatus per pound dry air

h'_w = enthalpy of water supplied at temperature t_2, Btu per pound

$h_s - h_1$ = increase in the enthalpy of the air-vapor mixture in passing through the apparatus, Btu per pound dry air (heating)

$h_1 - h_s$ = decrease in enthalpy when cooling

$$h_1 = 0.24t_1 + (0.444t_1 + 1061)\ W'_1$$

$$h_s = 0.24t_s + (0.444t_s + 1061)\ W_s \text{ when heating}$$

$$h_s - h_1 = 0.24\ (t_s - t_1) + W_s\ (1061 + 0.444t_s) - W_1\ (1061 + 0.444t_1) \text{ when cooling} \tag{21}$$

$$h_1 - h_s = W_1\ (1061 + 0.444t_1) - W_s\ (1061 + 0.444t_s) - 0.24\ (t_1 - t_s) \tag{22}$$

In equations (19) and (20) the factor $(W_s - W_1)\ h'_w$ is comparatively small, and in approximate calculations may be considered negligible, permitting the use of the simple equations

$$E_h - D_c = h_s - h_1 \text{ (heating)} \tag{23}$$

$$E_h + D_c = h_1 - h_s \text{ (cooling)} \tag{24}$$

When $t_1 = t_s$ and $h_s - h_1 = 0$ the result is a special condition of adiabatic saturation; i.e., no gain or loss of heat but an exchange between sensible and latent heat at the thermodynamic wet-bulb temperature.

165. Enthalpy Problem Using Equations

Given air at 95° F dry-bulb, 47% saturation, saturation pressure of 1.6591 in. Hg and barometric pressure of 29.921 in. Hg, what is the enthalpy per pound of dry air?

Solution (a)

From equation (16) $h = h_a + \mu h_{as}$

$h_a = 0.24 \times 95 = 22.80$ Btu specific enthalpy per pound dry air as in equation (11)

$h_{as} = W_s \times h_w{}^0 =$ total heat in moisture per pound dry air as in equation (13)

$h_w{}^0 = 0.444t + 1061 = 1103.18$ Btu enthalpy per pound moisture as in equation (12)

$$W_s = \frac{p_s}{48 - 1.6p_s} = \frac{1.6591}{48 - 1.6(1.6591)} = 0.03652 \text{ lb}$$

water per pound dry air

$h_{as} = 1103.18 \times 0.03652 = 40.25$ Btu at 95° F saturation

$\mu h_{as} = 0.47 \times 40.25 = 18.9175$ Btu, enthalpy of moisture per pound dry air

$h = 22.80 + 18.9175 = 41.7175$ Btu enthalpy of mixture per pound of dry air.

Solution (b)—using psychrometric Table 93

$$h = h_a + \mu\, h_{as}$$
$$h_a = 22.80$$
$$\mu\, h_{as} = 40.25 \times 0.47 = 18.9175$$
$$h = 22.80 + 18.9175 = 41.7175 \text{ Btu}$$

The convenience of using the psychrometric table for the solution of simple air-conditioning problems is now obvious.

166. Humidifying and Heating Problem

An inside condition of 70° F and 50% relative humidity is required when the outside air is at 0° F and 70% relative humidity.

Problem (a) Find weight of water vapor added to each pound of dry air.

Problem (b) Find net heat supplied if the heating medium is water at 50° F.

Solution (a)

At 70° F saturation 1 lb dry air contains 0.01574 lb moisture

At 70° F 50% RH 1 lb dry air contains $0.50 \times .01574 =$ 0.00787 lb moisture

At 0° F saturation 1 lb dry air contains 0.0007852 lb moisture

At 0° F 70% RH 1 lb dry air contains $0.7 \times 0.0007852 = 0.000549$ lb moisture

The water vapor added per pound dry air is approximately $0.00787 - 0.000549 = 0.00732$ lb.

Solution (b) Refer to energy equation (19) and typical apparatus diagram.

$$E_h - D_c = h_s - h_1 - (W_s - W_1)(50 - 32)$$

$W_1 = 0.000549$ lb moisture per pound dry air at 0° F and 70%

$W_s = 0.00787$ lb moisture per pound dry air at 70° F -50%

$W_1 = 0.00732$ lb moisture per pound dry air added.

$$h_1 = 0.24t_1 + (0.444t_1 + 1061)\,W_1$$

$$h_s = 0.24t_s + (0.444t_s + 1061)\,W_s$$

$$t_s = 70° \text{ F}$$

$$t_1 = 0° \text{ F}$$

$h_s = (0.24 \times 70) + [(0.444 \times 70 + 1061)\,0.00787)] =$ 25.43 Btu per pound at exit

$h_1 = (0.24 \times 0) + [(0.444 \times 0 + 1061)\,0.000549] =$ 0.5824 Btu per pound at entrance

$(W_s - W_1)(50 - 32) = 0.1317$

$E_h - D_c = 25.43 - 0.58 - 0.1317 = 24.72$ Btu per lb dry air net supply

This problem may be solved very rapidly using the psychrometric table exclusively, as follows:

Referring to equation (16) for an unsaturated air-vapor mixture

$$h = h_a + \mu\, h_{as}$$

h_a = enthalpy dry air (dry heat)
h_{as} = heat in moisture (moist heat)
μ = 50% saturation.

dry heat in saturated air at 0° F = 0
dry heat in air at 0° F 70% RH = 0
moist heat in saturated air at 0° F = 0.832 Btu
moist heat in air at 0° F 70% RH = 70 × 0.832 = 0.582 Btu per pound dry air
dry heat in saturated air at 70° F = 16.79 Btu
dry heat in air at 70° F 50% RH = 16.79 Btu
moist heat in saturated air at 70° F = 17.17 Btu
moist heat in air at 70° F 50% RH = 8.585 Btu per pound dry air

Enthalpy at 70° F and 50% RH = 16.79 + 8.585
25.375 Btu
Enthalpy at 0° F and 70% RH = 0 + 0.582 =
.582 Btu
Btu net heat supplied per pound dry air = 24.793
This is approximately same solution.

167. Cooling and Dehumidifying

Problem (a) When outside air is at a dry-bulb temperature of 85° F, a relative humidity of 50% and a barometric pressure of 29.92 in. Hg, find the dew point.

Problem (b). If the outside air specified above is cooled to 54° F calculate the quantity of vapor condensed per pound of dry air.

Problem (c). If the condensate is removed at 54° F compute the quantity of heat removed by referring to the energy equation and diagram.

Solution (a). Substituting in equation (10)

$$W_1 = \frac{0.622 p_s \phi}{29.92 - p_s \phi}$$

p_s = 1.2135 in. Hg saturation vapor pressure at 85° F.

then $W_1 = \dfrac{0.622 \times 1.2135 \times 0.50}{29.92 - 0.6067} = 0.01288$ lb moisture per pound dry air at 85°F and 50% relative humidity

Since $W_1 = W_t$ when $t = 64.3°$ F saturation, this is the dew point.

Solution (b) $W_s = \dfrac{0.622 \times 0.42}{29.92 - 0.42} = 0.00885$ lb moisture per pound dry air

$W_1 - W_s = 0.01288 - 0.00885 = 0.00403$ lb vapor condensed per pound of dry air.

Solution (c). See energy equations (20) and (22) and diagram.

$$E_h + D_c = h_1 - h_s - (W_1 - W_s)\, h'_w$$
$$h_1 = 0.24\, t_1 + W_1\,(0.444\, t_1 + 1061)$$
$$h_s = 0.24\, t_s + W_s\,(0.444\, t_s + 1061)$$

$h_1 = (0.24 \times 85) + [0.01288\,(0.444 \times 85 + 1061)] = 34.55$ Btu per pound of dry air entering

$h_s = (0.24 \times 54) + [0.00885\,(0.444 \times 54 + 1061)] =$ 22.56 Btu per pound of dry air leaving

$(W_1 - W_s)\, h'_w = 0.00403\,(54 - 32) = 0.08866$

h'_w = Btu per pound liquid water at 54° F from 32° F

$F_h + D_c = 34.55 - 22.56 - 0.08866 = 11.90134$ Btu removed per pound dry air.

168. Thermodynamic Wet-bulb Temperature

If an air stream is passed through or over liquid or solid water, respectively, the water will evaporate and thus increase the moisture content or humidity ratio of the air. A quantity of water may be injected to saturate the air. If the observed pressure is constant and the flow steady, if there is no gain or loss of heat to or from outside sources and if the temperature at which the air reaches saturation coincides with the temperature of the liquid (or solid) water added, the common temperature is called the thermodynamic wet-bulb temperature. When there is no gain of heat from or loss of heat to outside sources, and the only heat transferred is between the air and water the process is called an adiabatic process.

During the process of adiabatic saturation the dry-bulb temperature of the air drops to the wet-bulb temperature and no lower, the wet-bulb temperature remains constant and the weight of water vapor associated with each pound of dry air increases to W_s, where W_s is the weight of saturated vapor per pound of dry air for saturation at the wet-bulb temperature.

Problem. Find the thermodynamic wet-bulb temperature of dry air at 90° F.

Solution. Referring to energy equation (19) or (20)

$$E_h - D_e = h_s - h_1 - (W_s - W_1)\, h'_w$$

or

$$E_h + D_e = h_1 - h_s + (W_1 - W_s)\, h'_w$$

Inasmuch as the process is an adiabatic spray process, neither cooling nor heating

$$E_h = 0$$
$$D_e = 0$$
$$W_1 = 0 \text{ in problem}$$
$$h_s = h'_1 + W_s h'_w \qquad (25)$$

Therefore in solving for the temperature corresponding to h_s it is permissible to obtain a trial value by considering $W_s h'_w$ as negligible.

Solution (using equation $h_1 = h_s$)

h_1 = enthalpy at temperature t_1

h_s = enthalpy at wet-bulb temperature t_s

thus $h_1 = 0.24\, t_1 = 21.59$ Btu, sensible heat at 90° F,

$h_s = 0.24\ t_s + 1061\ W_s$ = enthalpy of air-vapor mixture

and $0.24\ t_s + 1061\ W_s = 21.59$ Btu, enthalpy of mixture at wet-bulb temperature

This is an indeterminate equation requiring a trial and error method of solution. Two methods may be used.

Solution (a). Find a temperature in the psychrometric table that nearly corresponds to an enthalpy of 21.59 Btu. This is 52° F and is the approximate wet-bulb. The true wet-bulb temperature is obtained by substituting in equation (25)

$$h_s = 21.59 + (0.008226 \times 20.1) = 21.755$$

where 0.008226 = lb of water per pound dry air at 52° F saturation

$h'_w = 20.1$ = specific enthalpy of liquid water at 52° F.

According to the table enthalpy at 52° F is 21.38.

According to the table enthalpy at 53° F is 21.95.

By interpolation true wet-bulb is found to be 52.65° F.

Solution (b). If the table is not available an empirical formula * may be used.

$$S = 0.84 - 0.005\ t_s \qquad (26)$$

where S = sensible heat factor, i.e., ratio of sensible heat to total heat.

t_s = temperature (saturation)

L = latent heat factor $= 0.16 + 0.005\ t_s$

For practical purposes equation (26) is true for temperatures between 35 and 100° F.

$h_1 = 0.24\ t_s$ = sensible heat at wet-bulb temperature

$h_s = 21.59$ = total heat at wet-bulb temperature

$$S = \frac{0.24t_s}{21.59} = 0.84 - 0.005t_s$$

$t_s = 52.1°$ F approximate thermodynamic wet-bulb temperature

* Devised by the senior author.

169. Adiabatic Saturation

For example, if ample quantities of water at a temperature of 432° F are sprayed into dry air at 90° F, a saturated mixture will be produced at an adiabatic saturation temperature.

From energy equation (18)

$$E_h - D_c = h_s - h_1 - (W_s - W_1)\, h'_w$$

Since $E_h = D_c$

$$h_s = h_1 + (W_s - W_1)\, h'_w$$

$W_1 = 0$ for dry air at 90° F

Therefore $h_s = h_1 + W_s h'_w$

h_s = total enthalpy at saturation temperature

$h_1 = 21.59$ Btu, enthalpy at 90° F

W_s = pounds moisture per pound dry air at saturation

$h'_w = 400$ Btu enthalpy of liquid water at 432° F

$$h_s = 21.59 + 400\, W_s$$

In the preceding problem water was injected at 52° F which corresponds to the thermodynamic wet-bulb temperature according to definition. Dry air at 90° F will vaporize 0.008226 lb water per pound dry air at the thermodynamic wet-bulb temperature. This is an adiabatic saturation process; i.e., no gain or loss of external heat occurs. In this problem water was injected at 432° F into dry air at the same temperature, 90° F and this is also an adiabatic mixing process.

Therefore $h_s = 21.59 + 400 \times 0.008226 = 24.88$ Btu, enthalpy per pound of dry air at 90° F after injection of water spray at 432° F.

According to the psychrometric table an enthalpy of 24.88 Btu for the mixture corresponds to a temperature, between 57 and 58° F, which is termed the adiabatic saturation temperature. It is possible to produce adiabatic saturation by mixing a stream of cold air with warm air.

Problem. Find the wet-bulb temperature corresponding to 95° F dry-bulb and 47% saturation using equation (16).

h_1 = enthalpy at 95° F and 47% saturation
h_s = enthalpy at thermodynamic wet-bulb temperature

From equation (16) giving enthalpy of an air-vapor mixture

$$h_1 = h_a + \mu\, h_{as}$$

$h_a = 0.24 \times 95 = 22.80$ Btu per pound dry air, sensible heat

$\mu h_{as} = 0.47 \times 40.25 = 18.9175$ Btu

where

$h_{as} = 40.25$ = total heat in moisture at saturation

μ = 47% saturation

$h_1 = 22.80 + 18.9175 = 41.7175$ Btu per pound dry air, enthalpy

h_s = enthalpy at wet-bulb t_s

$0.24\, t_s$ = sensible heat at wet-bulb

From equation (26)

$$\frac{0.24 t_s}{41.718} = 0.84 - 0.005 t_s$$

$t_s = 77.2°$ F approximately.

The solution is more conveniently obtained, if the psychrometric table is available, by finding the saturation temperature corresponding to an enthalpy of 41.71; this is 78.28° F, which is the true wet-bulb temperature.

Problem. Find the degree of saturation of moist air at 85° F dry-bulb, 70° F wet-bulb and 29.921 in. Hg.

Solution. The equation to be solved is the energy equation for an adiabatic process, i.e., equation (18).

Therefore $E_b = D_c$

and $h_s = h_1 + (W_s - W_1)\, h'_w$

where $h_s = 33.96$ Btu, enthalpy of saturated mixture at 70° F wet-bulb

$h_1 = 20.39$ Btu, enthalpy of dry air at 85° F

$W_s = 0.01574$ lb, moisture per pound dry air at 70° F saturation

$W_1 = W\mu = 0.02629\,\mu$

$W = 0.02629 =$ pounds water vapor per pound dry air at 85° F saturation

$h'_w = 38$ Btu, enthalpy of liquid water (from 32° F)

$$33.96 = 20.39 + 28.85\,\mu + (0.01574 - 0.02629\,\mu) \times 38$$

$\mu = 46.5\%$ saturation. In rough calculations, it is customary to ignore the term $(W_s - W_1)\,h'_w$.

170. Theory of Evaporation (Restatement)

The wet-bulb temperature is that constant temperature which a body of water will assume if free to evaporate into the atmosphere, provided the heat necessary for evaporation comes from an exchange between the atmosphere and the water, no heat entering from external sources and no heat being lost to external bodies. It is an adiabatic exchange of heat at small pressure differences.

In the compression of gases this process is known scientifically as an adiabatic process, sometimes called an isentropic process.

171. Cooling Towers, Air Washers, Adiabatic Saturation (Restatement)

When air moves through a spray of water as in a cooling tower or air washer, the air and water mutually exchange sensible and latent heat and the water arrives at a constant temperature equal to the wet-bulb temperature of the air if the exchange is perfect; practically, within 2 to 3 degrees of the wet-bulb temperature. When air at low humidity and high temperature is lowered in temperature and increased in humidity, the heat content per pound remaining the same, the air is said to be "adiabatically saturated."

Example

When outside air is at 90° F dry-bulb and 78° F wet-bulb, it is possible to cool water from 100° F to within about 3 degrees of the wet-bulb, or 81° F. This process is described in a later topic on cooling towers.

When dry air is adiabatically saturated, the temperature is reduced as the moisture is added to the air, and the decrease of the sensible heat of the air is equal to the simultaneous increase in the latent heat of the moisture in the air, due to evaporation. Inasmuch as the final temperature is the wet-bulb temperature, it may be assumed that the wet-bulb temperature is associated with air conditions at different dry-bulb temperatures, humidities, vapor pressures, latent heat contents and sensible heat contents, but that, nevertheless, the sum of the latent and sensible heats is always constant. In other words, the wet-bulb temperature is at points of constant heat.

During the process of adiabatic saturation, the sensible heat and latent heat vary but the total heat is constant. This must be the case, because it is assumed by definition of an adiabatic process that no heat can enter or leave. The only heat obtainable by the water vapor for its own evaporation is from the air. The air in losing its heat is cooled. It is to be noted here that while the total heat is constant the vapor pressure of the air is increased as its moisture content is increased.

172. The True Wet-bulb Temperature (W. H. Carrier Law)

The true wet-bulb temperature of air depends entirely on the total of the sensible and latent heats, and is independent of their relative proportions. In other words, the wet-bulb temperature of the air is constant provided the total heat of the air is constant.

Thermodynamic wet-bulb temperature is an important property of an air-vapor mixture because it is approximately the temperature indicated by the wet-bulb psychrometer. This instrument consists of a thermometer with its bulb covered with silk gauze and moistened with distilled water. It is whirled in the air until the thermometer reading is constant.

173. The Sling Psychrometer

This instrument (Figure 11) is generally used to determine relative humidity. An air-stream velocity of 1,000 ft per minute is recommended. Charts and tables are based on the standard barometric pressure of 29.92 in. Hg, therefore a correction for barometric pressure is necessary for extreme accuracy. This correction is made by multiplying the relative humidity, as determined from the chart, by the ratio of the observed barometric pressure and the standard barometric pressure. For temperatures below 32° F, the water on the wick is allowed to freeze; during this process the temperature will drop below the true wet-bulb temperature.

Problem. If air at 95° F and wet-bulb 75° F, is saturated adiabatically, what will be the final temperature and vapor content of the air? Use the psychrometric table.

Solution. The final temperature will be equal to the initial wet-bulb temperature or 75° F. Since the air is saturated at this temperature, psychrometric Table 93 gives $W_s = 0.01873$ lb of moisture per pound of dry air.

In the adiabatic saturation process, the heat given up by the dry air and associated vapor in cooling to the wet-bulb temperature is utilized in evaporating the water at the wet-bulb temperature. W. H. Carrier states in *Rational Psychrometric Formulae* that the equation for the process of adiabatic saturation is:

$$h'_{fg}\,(W_{t'} - W) = C_{pa}\,(t - t') + C_{ps}\,W\,(t - t') \qquad (27)$$

Let

$C_{pa} = 0.24$ specific heat of air
$C_{ps} = 0.45$ specific heat of water vapor

h'_{fg} = latent heat of vaporization at t'

$(W_{t'} - W)$ = increase in moisture content 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.

Then $h'_{fg}\,(W_{t'} - W) = (0.24 + 0.45\,W)(t - t')$ (28)

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

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

$$\Sigma = C_{pa}\,(t - 0) + W'\,[h'_{fg} + C_{ps}\,(t - t')] \qquad (29)$$

where

Σ = *total* heat of mixture, Btu per pound 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, ° F

t' = wet-bulb temperature, ° F

W = pounds of water vapor mixed with one pound of dry air

h'_{fg} = latent heat of vaporization at t'

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

$$\Sigma = C_{pa}(t' - 0) + Wt'h'_{fg} \qquad (30)$$

Equation (28) for the adiabatic saturation process is derived by equating (29) and (30).

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 furnishes a convenient method for finding the total heat of an air-vapor mixture by using a sling psychrometer to obtain the wet-bulb temperature. The total heat as given in equation (29) is not exactly equal to the true heat content or enthalpy as given in the psychrometric tables, since the heat content of liquids is not included. However for all practical purposes it is used for approximate calculations. To distinguish the total heat content in equation (29) from the true heat content or enthalpy, Carrier termed this psychrometric heat function the "sigma function."

174. Relation of Dew Point to Relative Humidity

An interesting relationship between the dew point and relative humidity has proved to be of great service in air-conditioning work. For a fixed relative humidity there is an approximately constant difference between the dry-bulb and dew point temperature over a considerable temperature range. The following table gives a clear idea of this relationship under various conditions.

This principle permits the use of a differential thermostat, responsive to both the room temperature and the dew point, to control the relative humidity in certain industrial installations, e.g., in textile mills and tobacco factories where relatively high humidities are carried and one of the main problems of design is the removal of the heat generated by machinery, etc.

175. Dew Point Control

The most positive and satisfactory control of relative humidity within a conditioned space is accomplished by fixing the dew point, i.e., the absolute humidity, at the conditioning unit. The relative humidity within the space is then set by controlling the degree of temperature rise above the dew point. This is done by varying the quantity of saturated air delivered to the space or by reheating the air by automatic control after it has been saturated in the humidifier.

TABLE 94

DEW POINT TEMPERATURE AND ROOM TEMPERATURE WITH HUMIDIFYING SPRAY TYPE AIR WASHER

Room Temperature	ϕ = Percentage Relative Humidity																					
	85%		80%		75%		70%		65%		60%		55%		50%		45%		40%		35%	
°F	T°	Di	T°	Di	T°	Di	T°	Di	T°	Di	T°	Di	T°	Di	T°	Di	T°	Di	T°	Di	T°	Di
65	61	4	59	6	57	8	55	10	53	12	51	14	48	17	45	20	43	22	40	25	37	28
70	65	5	64	6	62	8	60	10	58	12	55	15	53	17	50	20	48	22	45	25	41	29
75	70	5	69	6	66	9	64	11	62	13	60	15	58	17	55	20	53	22	49	26	46	29
80	75	5	73	7	71	9	69	11	67	13	65	15	62	18	60	20	57	23	53	27	50	30
85	80	5	78	7	76	9	74	11	72	13	69	16	67	18	64	21	61	24	58	27	54	31
90	85	5	83	7	81	9	79	11	77	13	74	16	72	18	69	21	66	24	62	28	59	31
95	90	5	88	7	86	9	84	11	81	14	79	16	77	18	73	22	70	25	67	28	63	32

Column T° gives dew point temperatures ($Di = 43.8 - 0.46\phi$).
Column Di gives difference between dew point and room temperature.

Cooling Capacity of Air Washers

Room Humidity %	Difference Versus Dew Point and Room Temperature	Cubic Feet Air Per Btu Cooling
50	20.4	2.71
55	17.8	3.10
60	15.3	3.64
65	12.7	4.31
70	10.7	5.10
75	8.8	6.28
80	6.8	8.11

176. Apparatus Dew Point Equation

This equation will prove exceedingly useful in solving for the dew point, a procedure often necessary in industrial air-conditioning work, especially if used in conjunction with the estimating form given at the end of Chapter XXII, Figure 94.

Problem. A store is to be maintained at 80° F dry-bulb and 67° F wet-bulb when outside conditions are 95° F dry-bulb and 75° F wet-bulb. The energy gain or sensible heat gain is as follows:

Conduction 16,000 Btu per hour
Sun effect 48,000 Btu per hour
Lights, power 13,900 Btu per hour

The ventilation requirement is 30,000 cu ft outside air per hour. The number of occupants is 50. Find (a) the apparatus dew point. (b) The refrigeration requirements.

Solution. (a) The condition of outside air

$$v_a = 13.97$$

$$v = 13.97 + 39\% \text{ of } 0.82 = 14.29 \text{ cu ft per pound}$$

$$h = 38.46 \text{ Btu per pound, total heat outside air}$$

$$h_a = 22.80 \text{ sensible heat at } 95^\circ \text{ F dry-bulb}$$

$$h - h_a = 15.66 \text{ Btu, latent heat of outside air.}$$

A dew point of 67.4° F corresponds to the latent heat of 15.66 Btu per pound dry air and a moisture content of 0.01437 lb

$$W = 0.0143 \text{ lb water per pound dry air (outside)}$$

$$\frac{30{,}000}{14.29} = 2{,}099 \text{ lb dry air per hour, entering with the ventilating air}$$

$$2{,}099 \times 38.46 = 80{,}727 \text{ Btu per hour, sensible heat}$$

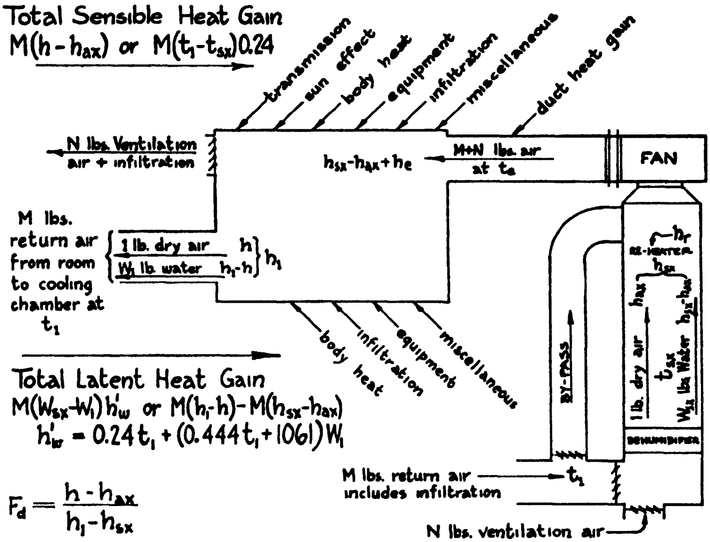
$$2{,}099 \times 0.01437 = 30.16 \text{ lb water per hour}$$

$$h'_w = 0.444t + 1{,}061 = \text{latent heat, Btu per pound vapor}$$

$$0.444 \times 95 + 1{,}061 = 1{,}103 \text{ Btu per pound water}$$

$$30.16 \times 1{,}103 = 33{,}266 \text{ Btu latent heat content of ventilating air.}$$

APPARATUS DEW POINT EQUATION & COOLING DIAGRAM



h_{sx} — Enthalpy or total heat BTU per lb. dry air leaving cooler.
h_{ax} — Sensible heat BTU per lb. dry air leaving dehumidifier.
h_1 — Enthalpy or total heat BTU per lb. dry air leaving room
h — Sensible heat BTU per lb. dry air leaving room.
W_{sx} — Lbs water per lb. dry air leaving dehumidifer.
W_1 — Lbs. water per lb. dry air entering dehumidifier - leaving room
t_{sx} — Temperature air leaving dehumidifier (saturated)
t_1 — Dry bulb air leaving room
F_d — Sensible heat differential factor
L_d — Latent heat differential factor $= 1 - F_d$
h_e — Sensible heat BTU per lb. dry air entering room $= h_r + h_{ax}$
$h_{sx} = 0.625\, t_{sx} - 11.2$ Empirical Equation (1)

$$t_{sx} = \frac{F_d(h_1 + 11.2) - 0.24\, t_1}{0.625\, F_d - 0.24} \quad \textbf{— Apparatus Dew Point (2)}$$

NOTE: The empirical equations (1) & (2) were developed by author as a matter of convenience & are sufficiently accurate for first trial dew point calculations. HH

FIGURE 12A

The properties of inside air are as follows

$$h_1 = 31.51 \text{ total heat at } 67^\circ \text{ F wet-bulb}$$
$$W_1 = 0.01115 \text{ lb water per pound dry air}$$

Therefore the sensible heat leaving the store with the circulating air is $2{,}099 \times 31.51 = 66{,}639$ Btu per hour; the weight of water leaving is $2{,}099 \times 0.01115 = 23.41$ lb per hour.

$0.444 \times 80 + 1{,}061 = 1{,}096 \times 23.41 = 25{,}657$ Btu, latent heat leaving with ventilating air.

From Table 109 in Chapter XI, and considering each occupant as a normal person at rest, it may be assumed that each person gives off 220 Btu per hour sensible heat and 180 Btu per hour latent heat

$$220 \times 50 = 11{,}000 \text{ Btu sensible heat}$$
$$180 \times 50 = 9{,}000 \text{ Btu latent heat}$$

177. Summary

Referring to apparatus dew point equation

A = pounds air per hour circulated

$A\,(h) = 66{,}639$ = Btu sensible heat returning to cooler

$A\,(h_{ax}) = 16{,}000 + 48{,}000 + 13{,}900 + 80{,}727 + 11{,}000 = 169{,}627$ Btu

$A\,(h - h_{ax}) = 169{,}627 - 66{,}639 = 102{,}988$ Btu, sensible heat gain

$A\,(h_1 - h) = 25{,}657$ Btu latent heat returning to cooler

$A\,(h_{sx} - h_{ax}) = 33{,}266 + 9{,}000$ Btu $= 42{,}266$ Btu, latent heat entering

$42{,}266 - 25{,}657 = 16{,}609$ Btu, latent heat gain

$102{,}988 + 16{,}609 = 119{,}597$ Btu total heat gain

$$\text{Sensible heat differential factor} = \frac{102{,}988}{119{,}597} = 0.865$$

Substituting in apparatus dew point equation

$$t_1 = 80^\circ \text{ F}$$

$$F_d = 0.865$$
$$h_1 = 31.51$$

$$t_{sz} = \frac{0.865(31.51 + 11.2) - 0.24 \times 80}{0.625 \times 0.865 - 0.24} = 59° \text{ F}$$

Apparatus dew point = 59° F

(b)

Total heat leaving at 67° F wet-bulb	=	31.51 Btu
Total heat entering at 59° F wet-bulb	=	25.70 Btu
Refrigeration	=	5.81 Btu per lb air

To find amount of air circulated per hour, *A*:

$$A = \frac{102{,}988 + 25{,}657}{5.81} = \frac{128{,}645}{5.81} = 22{,}000 \text{ lb}$$

$$\text{Refrigeration} = \frac{128{,}645}{12{,}000} = 10.7 \text{ tons}$$

CHAPTER X

DYNAMICS

In this work, an attempt has been made to place before the reader the elementary facts and principles of the science of air conditioning and also to offer some of its most practical applications. The method followed began with the study of well ascertained facts and proceeded onward to general principles, even though it might have been assumed that the reader was familiar with certain basic subjects.

The book has been divided into two parts. The first nine chapters of the book have dealt largely with the theoretical aspects of air conditioning. This chapter and those following it will cover the practical aspects.

178. Hydrodynamics

The study of a fluid in motion presents one of the most unsatisfactory branches of applied mechanics from a mathematical standpoint. The internal eddies, cross currents, and general intricacy of motion of the particles occurring in a pipe or duct transmitting a fluid or gas are almost entirely defiant of mathematical expression. In most practical cases we are forced to adopt as a basis for mathematical investigation the simple assumption that particles move side by side in such a way that those which at any instant form lamina or thin sheets perpendicular to the axis of the pipe or orifice, remain together as lamina during the further stages of the flow. It has been previously stated that a gas or vapor is a fluid the volume of which is limited only by the containing vessel.

179. Pumps ‡

Static head (h_s) on a pump is the vertical distance (in feet) between the free level of the source of supply and the point of free discharge, or the level of the free surface of the discharge liquid, or (in the case of delivery under pressure) the equivalent level of a free discharge.

Static suction lift (l_{ss}) is the vertical distance (in feet) from the level of the source of supply to the centerline of the pump, when pump is above source of supply.

Static suction head (h_{ss}) is the difference (in feet) between the level of the source of supply and the centerline of the pump, when the pump is below the source of supply; or (in the case of delivery under pressure) is the difference (in feet) between the level corresponding to the pressure and the centerline of the pump.

Friction head (h_f) is the pressure (in terms of feet of liquid) required to overcome the resistance to flow in pipe and fittings.

Velocity head (h_v) is the vertical distance a body would have to fall to acquire the velocity v. It corresponds to the static or pressure head which would cause that velocity.

$$h_v = \frac{v^2}{2g} = 0.0155v^2 = \frac{0.00259\ \text{gpm}^2}{d^4} = \frac{0.00127\ \text{bph}^2\ \dagger}{d^4}$$

Velocity head may be converted to equivalent pressure head by suitable means such as a Venturi tube. In any flowing liquid the sum of pressure head and velocity head remains constant.

Total dynamic head (h_{td}) is the difference (in feet) between the elevation corresponding to the pressure at the discharge flange of a pump and the elevation corresponding to the vacuum or pressure at the suction flange of the pump, corrected to the same datum plane, plus the velocity head at the discharge flange,

† Formulas for h_v and v involving gpm and bph apply only to circular conduits.

‡ Abstract from *Cameron Hydraulic Data*, G. V. Shaw and A. W. Loomis, editors.

minus the velocity head at the suction flange. $(h_{td})^* = (h_s) + (h_f) + (h_{vdisch.})$

Total dynamic suction lift (l_{ds}) is the difference (in feet) between the level corresponding to the vacuum at the suction flange (corrected to the pump centerline) and the pump centerline. $(l_{ds})^* = (l_{ss}) + (h_{fsuct}) + (h_{rsuct})$

Total dynamic suction head (h_{ds}) is the difference (in feet) between the level corresponding to the pressure at the suction flange (corrected to the pump centerline) and the pump centerline. $(h_{ds})^* = (h_{ss}) - (h_{fsuct}) - (h_{rsuct})$

180. Work Performed in Pumping

In order to determine the work required of a pump it is necessary to know the total dynamic head and the weight of liquid to be pumped in a given time. Usually weight is not given but rather volume in gallons per minute or barrels per hour, along with the density or specific gravity. The weight can be calculated from this information, but this is not necessary since the horsepower formulas are usually given in terms of gpm or bph. Theoretical horsepower of a pump is usually called the hydraulic horsepower.

$$\text{hydraulic horsepower} = \frac{(\text{gpm})\,(h_{td})\,(s)\,(8.33)}{33{,}000} = \frac{(\text{gpm})\,(h_{td})\,(s)}{3960} = \frac{(\text{bph})\,(h_{td})\,(s)}{5660}$$

$$\text{hydraulic horsepower} = \frac{(\text{gpm})\,(p_{td})}{1714} = \frac{(\text{bph})\,(p_{td})}{2450}$$

where p_{td} is expressed as pounds per square inch.

The actual or brake horsepower of a pump is greater than the theoretical or hydraulic horsepower by the amount of the losses incurred in the pump, through friction leakage, etc. The efficiency of a pump is, therefore, measured as:

$$\text{Pump efficiency} = \frac{\text{hydraulic horsepower}}{\text{brake horsepower}}$$

* These formulas apply when estimating before a pump is installed.

TABLE 95

PRESSURE DROP OF WATER THROUGH VALVES AND FITTINGS

Type of Fittings	½	¾	1	1¼	1½	2	2½	3	3½	4	4½	5	6
	Nominal Pipe Size—Inches; Equivalent feet of straight pipe having the same pressure drop												
Gate Valve—Open	0.3	0.4	0.5	0.7	0.8	1.1	1.4	1.8	2.1	2.5	2.0	3.3	4.1
Globe Valve—Open	13	18	24	34	42	57	71	93	111	131	151	173	218
Angle Valve—Open	6.3	9.0	12	17	21	28	36	47	56	65	75	87	109
Standard 45° Elbow	0.5	0.8	1.0	1.5	1.8	2.4	3.0	3.9	4.7	5.5	6.3	7.3	9.2
Standard 90° Elbow	1.1	1.6	2.2	3.1	3.7	5.1	6.4	8.4	10	12	14	16	20
Medium Sweep 90° Elbow	1.0	1.3	1.8	2.6	3.1	4.2	5.3	7.0	8.4	9.8	11	13	16
Long Sweep 90° Elbow	0.8	1.1	1.5	2.1	2.5	3.4	4.3	5.6	6.7	7.9	9.0	10	13
Square Elbow—90°	2.3	3.2	4.4	6.2	7.5	10	13	17	20	24	27	31	39
Close Return Bend	2.8	4.0	5.4	7.5	9.1	12	16	21	25	29	33	38.2	48
Standard Tee—Full Size Branch*	2.3	3.2	4.4	6.2	7.5	10	13	17	20	24	27	31	39
Standard Tee—Through Run	0.7	1.0	1.4	1.9	2.3	3.2	4.0	5.2	6.2	7.3	8.4	9.7	12
Sudden Enlargement from d to D**													
d/D = ¼	1.2	1.7	2.2	3.2	3.8	5.2	6.5	8.6	10	12	14	16	20
d/D = ½	0.7	1.0	1.4	1.9	2.3	3.2	4.0	5.2	6.2	7.3	8.4	9.7	12
d/D = ¾	0.3	0.4	0.5	0.7	0.8	1.1	1.4	1.8	2.1	2.5	2.9	3.3	4.1
Sudden Contraction from D to d**													
d/D = ¼	0.5	0.8	1.1	1.5	1.8	2.4	3.1	4.0	4.8	5.6	6.5	7.5	9.4
d/D = ½	0.4	0.6	0.8	1.1	1.4	1.9	2.3	3.1	3.7	4.3	5.0	5.7	7.2
d/D = ¾	0.3	0.4	0.5	0.7	0.8	1.1	1.4	1.8	2.1	2.5	2.9	3.3	4.1
Ordinary Pipe Entrance with Upstream End of Pipe Flush with Inside Face of Reservoir Wall	0.6	0.9	1.2	1.7	2.1	2.8	3.6	4.7	5.6	6.5	7.5	8.7	11
Entrance with Pipe Projecting into Reservoir and Extending beyond Face of Reservoir Wall (Borda Entrance)	1.1	1.5	2.0	2.8	3.4	4.7	5.9	7.7	9.3	11	13	14	18

* Pressure drop through side outlet, or from side outlet through run.

** Equivalent feet of the smaller diameter pipe, "d."

TABLE 96

FRICTION LOSSES IN WATER PIPES *

Col. A = Standard Weight Steel Col. B = Extra Heavy Steel

Pipe size	½ in.		¾ in.		1 in.		1¼ in.		1½ in.		2 in.		2½ in.		3 in.	
Velocity factor	1.06	1.37	0.60	0.74	0.371	0.446	0.215	0.250	0.158	0.182	0.096	0.109	0.067	0.076	0.043	0.052
GPM	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
1	2.10	3.96														
2	7.57	14.30	1.93	3.21	0.593	0.930										
3	16.00	30.20	4.08	6.79	1.26	1.97										
4	27.30	51.40	6.94	11.60	2.14	3.28										
5			10.50	17.50	3.24	5.07	0.853	1.24								
6			14.70	24.50	4.54	7.10	1.20	1.74								
7			19.60	32.60	5.50	9.00	1.59	2.31								
8			25.00	41.70	7.73	12.10	2.04	2.96								
10					11.70	18.30	3.08	4.47	1.45	2.05						
15					24.00	38.00	6.53	9.51	3.09	4.35						
20					42.10	65.80	11.10	16.10	5.24	7.39						
25					37.80	63.00	16.80	24.90	7.92	11.20	2.35	3.20				
30							23.50	34.10	11.10	15.70	3.29	4.49	1.39	1.86		
40							40.00	58.10	18.90	26.70	5.60	7.64	2.36	3.17		
50									28.50	40.30	8.46	11.50	3.56	4.79	1.24	1.63
60	8.56	1.11									11.90	16.20	4.99	6.72	1.74	2.28
70	1.14	1.47									15.80	21.50	6.64	8.94	2.31	3.04
90	1.81	2.35									25.10	34.30	10.60	13.90	3.67	4.83
120	3.08	4.00	1.67	2.14									18.00	24.20	6.26	8.23
150	4.67	6.05	2.53	3.24	0.842	1.06									9.48	12.50

* In feet head loss per 100 feet of run.

TABLE 96 (*continued*)

Pipe size	3½ in.		4 in.		5 in.		6 in.		8 in.		10 in.		12 in.	
GPM	A	B	A	B	A	B	A	B	A	B	A	B	A	B
180	6.53	8.46	3.53	4.52	1.18	1.48	0.481	0.618						
220			5.12	6.56	1.70	2.15	0.697	0.895	0.183	0.229				
260			6.97	8.93	2.32	2.92	0.950	1.22	0.25	0.312				
320			10.20	13.10	3.41	4.29	1.39	1.79	0.368	0.459				
450					6.40	8.47	2.62	3.36	0.689	0.860	0.228	0.26		
650							5.17	6.64	1 36	1.70	0.45	0.514		
700							5.93	7.62	1.56	1.95	0.516	0.589	0.214	0.238
900									2.48	3.10	0.831	0.938	0.341	0.378
1000									3.02	3.77	0.998	1.14	0.415	0.460
1200									4.23	5.28	1.40	1.60	0.581	0.644
1800									8.95	11.20	2.96	3.38	1.23	1.36
2000											3.60	4.11	1.50	1.60
Velocity factor	.0325	.0361	.0252	.0279	.016	.0176	.0111	.0123	.00641	.00703	.00407	.00430	.00284	.00296

NOTES: C = 100 (constant allowing for surface roughness)
Velocity — ft per sec = gpm × velocity factor
Friction loss increases as much as 20% at 32° F
Friction loss decreases as much as 20% at 212° F
0.4331 × ft = lb per sq in. (Based on water at 39.2° F)
Tables cover an average range of 3 lb head loss or velocity of 3 **feet per second.**
Williams and Hazen's formula for water at about 60° F
Arranged by H. Herkimer—Reprints by Permission

Other useful formulas are:

For motor-driven pumps

Kilowatts per 1000 gal per hour of liquid

$$= \frac{0.00315\ (s)\ (h_{td})}{\text{(pump efficiency) (motor efficiency)}}$$

For direct-acting steam pumps

Duty in terms of foot-pounds per 1000 lb of steam

$$= \frac{\text{theoretical horsepower (1980) 1,000,000}}{\text{total steam per hour in pounds}}$$

Sometimes theoretical horsepower for figuring duty is calculated using static head instead of total dynamic head. In this case duty is in terms of foot-pounds of useful work done per 1000 lb of steam. As given above the formula is in terms of foot-pounds of actual work done per 1000 lb of steam.

181. Fluids in Motion—General Equations

Bernouilli's theorem states that at any point in a tube through which a liquid is flowing, the sum of the pressure energy, potential energy and kinetic energy is a constant.

$$P + hdg + \tfrac{1}{2}\, dv^2 = \text{a constant}$$

where

P is pressure
h is height above a reference plane
d is density of the liquid
v is velocity of flow

The theorem further states that when *fluid friction* or *skin friction* is taken into account there is a *loss of head* or *friction head* between any two selected points, such as the two ends of a pipe or air duct. Then, in a steady flow, the sum of the velocity, pressure and potential head at any point A, is equal to the sum of the corresponding upstream points less the frictional resistance H_f between points A and B.

Applying Bernouilli's theorem for friction between points m and n

$$\frac{v^2_m}{2g} + \frac{P_m}{w} = \frac{v^2_n}{2g} + \frac{P_n}{w} - H_f$$

where

$$H_f = \text{frictional resistance}$$
$$v^2_m = v^2_n \text{ for same pipe}$$
$$H_f = \frac{P_n - P_m}{w}$$

182. Design Factors

In the development of air flow formulas and tables it is customary to reduce head H_v in feet to h_v in inches of water and velocities to feet per minute for ventilating calculations.

$$V = 1096.5 \sqrt{\frac{h_v}{d}} \tag{1}$$

where V = velocity, feet per minute
h_v = velocity head or pressure, inches of water
d = weight of air, pounds per cubic foot

For dry air (70° F and 29.921 in. Hg barometric pressure d = 0.075 lb per cubic foot.

Substituting this value in equation (1)

$$V = 1096.5 \sqrt{\frac{h_v}{0.075}} = 4005 \sqrt{h_v}$$

183. Friction in Air Ducts

The *loss of head* or *friction head* between any two selected points in an air duct may be expressed

$$H_L = 4f \frac{Lv^2}{D2g}$$

TABLE 97

VELOCITIES OF DRY AIR IN FEET PER MINUTE (AT VARIOUS TEMPERATURES) CORRESPONDING TO VARIOUS VELOCITY PRESSURES IN INCHES OF WATER

Barometric Pressure 30 inches

Pressure		Temperature ° F					
Inches	Ounces	50°	70°	100°	150°	300°	550°
0.1	0.0577	1242	1266	1300	1358	1516	1830
0.2	0.1154	1757	1791	1841	1921	2145	2590
0.25	0.1443	1965	2003	2059	2149	2399	2895
0.3	0.1730	2151	2193	2254	2352	2626	2175
0.4	0.2308	2485	2533	2603	2717	3033	2660
0.5	0.2884	2778	2832	2911	3038	3391	4095
0.6	0.3460	3043	3102	3188	3327	3715	4490
0.7	0.4037	3287	3351	3445	3595	4013	4850
0.75	0.4326	3402	3468	3565	3720	4153	5020
0.8	0.4614	3524	3582	3682	3843	4290	5185
0.9	0.5190	3728	3800	3906	4076	4550	5500
1.0	0.5768	3929	4005	4117	4296	4796	5795
1.25	0.7209	4393	4478	4602	4804	5362	6470
1.50	0.8650	4812	4905	5042	5262	5874	7100
1.75	1.0092	5197	5298	5446	5683	6344	7655
2.00	1.1535	5556	5664	5822	6076	6783	8195
2.25	1.2975	5892	6007	6174	6443	7193	8690
2.50	1.4418	6211	6322	6508	6792	7582	9150
2.75	1.5860	6514	6641	6827	7124	7952	9600
3.00	1.7300	6807	6937	7130	7440	8307	10000
4.00	2.3070	7857	8010	8233	8592	9581	11580
5.00	2.8840	8772	8943	9192	9593	10710	12900
6.00	3.4600	9623	9810	10083	10523	11750	14180

where

H_L = loss of head, feet
f = coefficient of friction
D = diameter of duct, feet
L = length of duct, feet

TABLE 98

LOW VELOCITY PRESSURES AT 70° F

Velocity Pressure Inches of Water	Velocity of Air Feet per Minute
0.001	123
0.002	176
0.003	219
0.004	252
0.006	309
0.008	358
0.010	400
0.012	433
0.014	476
0.016	506
0.018	536
0.020	566
0.022	594
0.024	619
0.026	644
0.028	668
0.030	692
0.035	747
0.040	799
0.050	896
0.060	1000

The value of f is independent of the pressure, varies inversely with the square of the velocity and directly with the amount and roughness of rubbing surface and density. For air and other gases f was formerly taken as 0.005. A modification of this formula is

$$h_L = f\frac{L}{D}h_v = \frac{L}{CD}\left(\frac{V}{4005}\right)^2$$

where h_L = loss of head in inches of water

$h_v = \left(\frac{V}{4005}\right)^2$ = velocity head inches of water

V = velocity of air in feet per minute
D = diameter of pipe in feet
f = coefficient of friction
$C = \frac{1}{f}$ = length of pipe in diameters for one head loss

For all practical purposes C equals 50 for average duct construction. The old friction chart was based on a loss of velocity head (at a velocity of 2000 fpm) in a length equal to 50 diameters of a 24 in. duct. Recent work in this field of research by the A.S.H.V.E. Laboratory has resulted in a correction of the old formula upon which the following duct friction tables are based.

The tables are based on air at 70° F, 29.91 in. Hg, and air density of 0.075 lb per cubic foot. Since the tables apply for these conditions only it is desirable to apply a correction for other conditions.

This correction is

$$H_A = H_s \times S$$

where

H_A = friction loss, inches of water at actual conditions
S = ratio of density of air at actual conditions to density of air at standard conditions.

For all practical calculations the correction factor is not considered necessary.

TABLE 99

PRESSURE REQUIRED, INCHES WATER GAGE, TO OVERCOME DUCT FRICTION LOSSES PER 100 LINEAR FEET OF SMOOTH INTERIOR INCLUDING 40 JOINTS—GOOD CONSTRUCTION—

Air density 0.075 lb per cubic foot.

Round Duct Diameter Inches	Cross-Sectional Area, Sq Ft	Velocity in Feet Per Minute 300	400	500	600	700	800	900	1000
3	0.049	0.077	0.140	0.200	0.280	0.360	0.460	0.600	0.750
4	0.087	0.055	0.080	0.150	0.190	0.250	0.320	0.400	0.500
5	0.14	0.040	0.070	0.100	0.149	0.190	0.250	0.300	0.375
6	0.19	0.033	0.055	0.080	0.110	0.150	0.200	0.250	0.300
7	0.27	0.027	0.045	0.065	0.095	0.130	0.160	0.200	0.250
8	0.35	0.023	0.037	0.056	0.080	0.110	0.140	0.175	0.200
9	0.44	0.020	0.030	0.048	0.070	0.090	0.120	0.150	0.175
10	0.54	0.0175	0.028	0.042	0.060	0.080	0.100	0.120	0.160
12	0.78	0.014	0.022	0.035	0.048	0.065	0.080	0.100	0.130
14	1.07	0.012	0.018	0.029	0.040	0.054	0.070	0.085	0.110
16	1.39	0.011	0.016	0.024	0.035	0.045	0.059	0.081	0.090
18	1.77	0.010	0.013	0.022	0.030	0.040	0.050	0.071	0.075
20	2.18	0.009	0.012	0.018	0.026	0.035	0.045	0.055	0.070
22	2.63	0.008	0.010	0.016	0.023	0.031	0.040	0.050	0.060
24	3.14	0.007	0.009	0.015	0.022	0.027	0.035	0.045	0.055
26	3.68			0.014	0 019	0.025	0.033	0.040	0.050
28	4.27			0.013	0.017	0.023	0.030	0.036	0.045
30	4.90			0.012	0.016	0.021	0.026	0.034	0.040
32	5.58			0.011	0.015	0.019	0.025	0.031	0.038
34	6.30			0.010	0.014	0.018	0.022	0.030	0.035
36	7.06				0.013	0.017	0.021	0.028	0.033
38	7.88				0.012	0.015	0.020	0.025	0.031
40	8.73				0.011	0.014	0.019	0.024	0.030
45	11.04				0.010	0.013	0.016	0.021	0.025
50	13.63					0.011	0.015	0.018	0.022
60	19.64					0.009	0.012	0.015	0.018
70	26.73						0.010	0.012	0.015
80	34.90							0.010	0.012
90	44.17								
100	54.54								

Increase pressure 10% to 20% for poor duct construction. For quiet operation use values less than 0.07 in. shown below solid line.

Table 100

PRESSURE REQUIRED IN INCHES OF WATER GAGE TO OVERCOME DUCT FRICTION LOSSES PER 100 LINEAR FT OF SMOOTH INTERIOR INCLUDING 40 JOINTS—GOOD CONSTRUCTION—

Based on air density of 0.075 lb per cubic foot

Duct Diameter Inches	Velocity in Feet Per Minute 1200	1400	1600	1800	2000	2200	2400	2800	5000	9000
3	1.00	1.30	1.70	2.20	2.60	3.10	3.60	5.00		
4	0.70	0.92	1.20	1.50	1.80	2.20	2.50	3.50	10.5	
5	0.52	0.70	0.90	1.20	1.30	1.60	1.90	2.60	8.0	
6	0.42	0.55	0.70	0.90	1.10	1.30	1.50	2.10	6.5	
7	0.38	0.45	0.60	0.75	0.90	1.10	1.30	1.80	5.2	
8	0.28	0.39	0.50	0.63	0.75	0.90	1.10	1.50	4.5	
9	0.25	0.33	0.42	0.54	0.66	0.80	0.95	1.30	3.8	
10	0.22	0.30	0.38	0.48	0.59	0.70	0.82	1.10	3.5	10.00
12	0.17	0.23	0.30	0.38	0.46	0.55	0.65	0.90	2.75	8.50
14	0.15	0.20	0.25	0.32	0.39	0.46	0.55	0.74	2.30	7.00
16	0.12	0.156	0.22	0.27	0.33	0.40	0.46	0.62	2.00	6.00
18	0.11	0.14	0.19	0.23	0.28	0.35	0.40	0.55	1.70	5.30
20	0.095	0.125	0.16	0.21	0.25	0.30	0.36	0.49	1.50	4.80
22	0.085	0.110	0.15	0.18	0.23	0.27	0.34	0.43	1.30	4.00
24	0.075	0.100	0.13	0.16	0.20	0.24	0.28	0.39	1.10	3.60
26	0.070	0.092	0.12	0.15	0.18	0.22	0.26	0.35	1.00	3.30
28	0.065	0.082	0.11	0.13	0.165	0.20	0.24	0.32	0.98	3.00
30	0.058	0.078	0.10	0.12	0.155	0.18	0.22	0.29	0.90	2.75
32	0.052	0.071	0.091	0.115	0.140	0.17	0.20	0.27	0.85	2.00
34	0.050	0.066	0.085	0.110	0.130	0.16	0.18	0.25	0.79	2.50
36	0.046	0.061	0.080	0.100	0.120	0.15	0.17	0.23	0.71	2.40
38	0.044	0.058	0.075	0.095	0.115	0.14	0.16	0.22	0.69	2.20
40	0.041	0.056	0.070	0.090	0.100	0.13	0.15	0.21	0.65	2.00
45	0.036	0.047	0.060	0.077	0.095	0.11	0.13	0.18	0.55	1.75
50	0.032	0.041	0.052	0.069	0.081	0.10	0.12	0.16	0.50	1.50
55	0.028	0.037	0.048	0.060	0.075	0.09	0.105	0.14	0.45	1.30
60	0.025	0.034	0.042	0.055	0.067	0.08	0.095	0.125	0.40	1.20
70	0.021	0.028	0.035	0.045	0.055	0.065	0.079	0.105	0.33	1.10
80	0.018	0.024	0.031	0.039	0.047	0.058	0.068	0.090	0.28	0.90
90										
100										

Increase pressure 10 to 20% for poor duct construction. For silent operation use values less than 0.07 in. shown below solid lines, not exceeding 1400 FPM.

TABLE 101

FRICTION LOSS THROUGH BLAST HEATERS

(Air friction in inches of water at 68° F)

Rows of Tubes deep	Net Face Area Velocity Feet per Minute 200	500	700	1000	1200
1	0.011	0.042	0.071	0.126	0.178
2	0.021	0.091	0.161	0.33	0.53
3	0.031	0.132	0.232	0.47	0.71
4	0.041	0.181	0.322	0.67	1.05
6	0.062	0.271	0.481	1.00	1.57

TABLE 102

FRICTION LOSS THROUGH SPRAY CHAMBER AND ELIMINATORS OF A HUMIDIFIER

(Spray in direction of air)

Air Velocity Feet per Minute	Water Head Loss Inches Pressure
390	0.15
500	0.23
600	0.35

If spray water is opposite to direction of air the pressure loss is increased 1/16 inch.

TABLE 103

APPROXIMATE FAN BRAKE HORSEPOWER

Requirements for passing 10,000 cfm of air through humidifiers at various velocities and static pressures. Mechanical efficiency of fans 78%

	30° ELIMINATORS SPACED ON 1-1/8 IN. CENTERS		45° ELIMINATORS SPACED ON 2-1/4 IN. CENTERS	
VELOCITY FPM	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

184. Dynamic Losses

The pressure drop in air distributing systems is due to dynamic losses as well as friction losses. The friction losses are those caused by friction against the interior surface of the duct. The dynamic losses are those caused by a change in the direction or in the velocity of the air flow. Dynamic losses occur principally at elbows, reducers, entrance and exit, i.e., wherever a velocity change occurs. The entrance loss is the difference between the total pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from

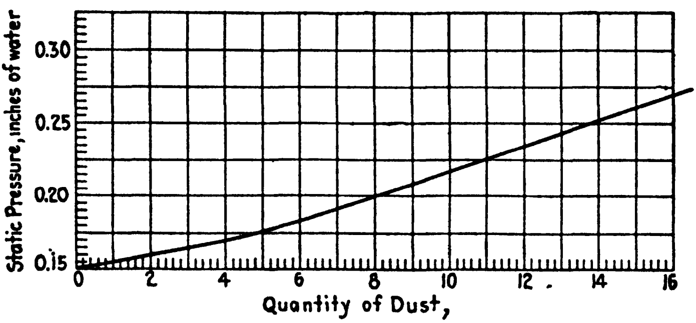
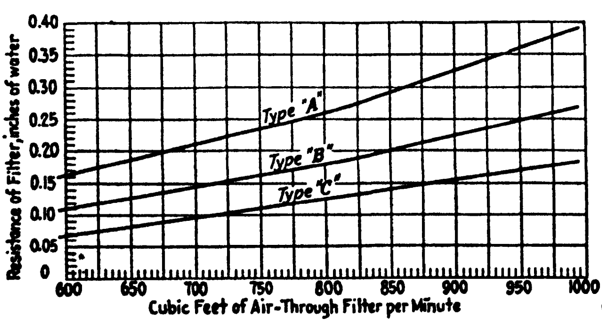


CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION



RESISTANCE TO AIR-FLOW OF A TYPICAL UNIT AIR FILTER

FIGURE 12B

Resistance to Air-Flow

0.1 to 0.5 times the velocity head. The pressure loss in elbows must also be allowed for in design. It is customary to express dynamic losses in terms of the percentage of the velocity head, which is expressed in inches of water gage pressure. See Figures 13 and 13A.

185. Fan Total Pressure

Practical experience has determined that a minimum allowance of 0.06 in. loss per 100 ft of duct, will result in quiet operation in living rooms, libraries, churches, theaters, etc., where noise is disturbing. In factories, or where noise is less annoying, a friction loss of 0.20 in. is allowable. It should be emphasized that, in addition to the friction loss in ducts as calculated from the tables following, consideration must be given to the losses in elbows, coolers, heaters, etc., that may increase the total pressure required to twice the duct loss. See Table 103 for brake horsepower with dehumidifiers.

186. Aspect Ratio of Ducts

The sides of all ducts and 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.

187. Friction Loss in Duct Fittings

The dynamic and friction losses in elbows is customarily expressed as equal to the number of diameters of round pipe or the number of widths of rectangular pipe equivalent to length of duct. Long radius elbows offer much less resistance than short radius elbows. Good results are obtained when the radius to the center of the elbow is 1.5 times the diameter of the duct. No advantage is gained by a center line radius of more than 2 diameters. One head loss is taken at 50 diameters.

See Figures 13 and 13A.

PRESSURE LOSS IN DUCT FITTINGS IN PERCENT VELOCITY PRESSURE

V = Ft. per minute P in. water velocity pressure $= \left(\frac{V}{4005}\right)^2$ f_v = in. pressure loss in terms of P

V	123	219	252	506	619	1000	1500	2000	3000	4000	5664	6937
P	0.001	0.003	0.004	0.016	0.024	0.0625	0.141	0.25	0.563	1.00	2.00	3.00

P = 0.0625 inches water (silent operation)
P = 0.25 inches for industrial applications

$f_v \times 50$ = Pressure loss in equivalent duct diameters (smooth)

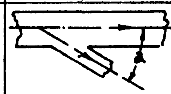
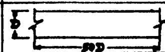
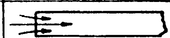
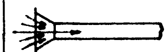
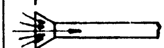
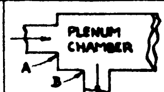
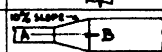
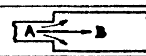
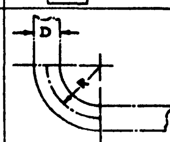
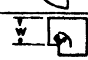
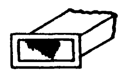
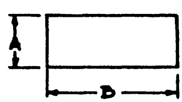
DESCRIPTION		f_vP	DIAGRAM
Branch Duct from Main. Loss in branch in percent velocity pressure in main	Angle a = 15° Angle a = 30° Angle a = 45° Angle a = 90°	0.08 P 0.18 P 0.45 P 1.00 P	
1.00 velocity head loss per 50 diameters duct	Straight Duct	1.00 P	
EXAMPLE Plain opening loss at 2000 FPM $f_v P = 0.87 \times 0.25 = 0.2175$ in. or equivalent to $0.87 \times 50 = 43.50$ duct diameters in equivalent length	Plain opening	0.87 P	
	Coned opening in wall	0.22 P	
	Coned opening	0.15 P	
Enlargement of Duct	Loss at A Loss at B	1.00 P 0.50 P	
	Neglect - Static pressure regain	0	
Entrance to Plenum Area B = 2A	Loss in A Gain in B (neglect)	0.25 P 0.50 P	
Orifice	70% free area	1.00 P	
Square Elbow (no radius)	Round Duct Square Duct	1.00 P 1.25 P	
Elbow (square duct) Radius of center line in percent of pipe diameters R/D with following duct	50% 75% 100% 125% 150% 200%	1.05 P 0.40 P 0.22 P 0.18 P 0.15 P 0.12 P	
Broad-way Elbow (square inside)	With turn vanes Without turn vanes	0.45 P 0.85 P	
Square Mitre Elbows with turn vanes	$\frac{R}{W} = 0.4$	0.90 P	
	$\frac{R1}{W} = 0.3$ $\frac{R2}{W} = 0.5$	0.75 P	
	$\frac{R1}{W} = 0.2$ $\frac{R2}{W} = 0.4$ $\frac{R3}{W} = 0.6$	0.65 P	

FIGURE 13

PRESSURE LOSS IN DUCT FITTINGS IN PERCENT VELOCITY PRESSURE (CONTINUED)

DESCRIPTION		$f_v P$	DIAGRAM
Standard Elbows with Various Radii	$\frac{R}{W} = 0.4$	0.74 P	
	$\frac{R1}{W} = 0.2 \quad \frac{R2}{W} = 0.4$	0.67 P	
	$\frac{R1}{W} = 0.6$	0.55 P	
Transition Reverse		0.22 P	
Transition Increaser or Reducer	Exceeding 2"-Less than 2" same as straight duct	0.16 P	
Reverse Elbow	(Right, Left or Center)	1.00 P	
Grilles (Deflection angle 45°) Friction loss less for smaller angle	80% Free Area 70% " " 60% " " 50% " "	1.45 P 2.15 P 3.20 P 5.00 P	
Fresh Air Louver	Based on Free Area measured normal to louver blades	0.75 P	

Friction of round ducts equals approx. square ducts. For rectangular ducts increase resistance factors for various aspect ratios by using the following duct proportion coefficients. $D = \sqrt{\text{area}}$

A x B	COEFFICIENT	DIAGRAM
1 x 1	1.00	
1 x 2	1.10	
1 x 3	1.20	
1 x 4	1.45	
1 x 6	1.55	

Approximate Equipment Resistance Losses in Inches Water Pressure (Check with manufacturers data)

EQUIPMENT	INCHES WATER
1. Wire grating	0.10
2. Outdoor air intake (1000 FPM)	0.085
3. Preheating Coils	0.10
4. Air Washer	0.25
5. Cooling Coils (Four coils 500 FPM)	0.25
6. Reheating Coils	0.15
7. Expansion at fan	0.085
8. Filters	0.25
9. Dampers	0.10

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FIGURE 13A

188. Circular Equivalents of Rectangular Ducts

The friction chart is based on circular ducts. To find the rectangular duct refer to tables 104 and 105 based on the equation below

$$d = 1.265 \sqrt[5]{\frac{(ab)^3}{a + b}}$$

where

a = one side of rectangular pipe, in feet or inches
b = other side of rectangular pipe, in feet or inches
d = equivalent diameter of round pipe for equal friction per foot of length to carry the same capacity, in feet or inches.

189. Measurements of Air Volume

To check accurately the performance of air-conditioning equipment, it is necessary to determine the velocity and volume of air or the gases involved. Velocity may be determined with an anemometer, a Pitot tube, an orifice and manometer or electrical resistance instruments. The volume is calculated by the application of the general equation

$$Q = AV$$

where Q = volume in cubic feet per minute
A = area of duct or orifice in square feet
V = velocity of air or gas in feet per minute.

190. The Anemometer

This instrument is a miniature wind mill. The air impinges against the vanes and causes rotation of the wheel. The instrument is calibrated to the rate of speed at which the wheel will rotate when facing the wind at various velocities. The number of revolutions (as shown on the dial of the counter mechanism) per unit of time, therefore, corresponds to certain known wind velocities. The dial, if graduated to a known time interval, may

Table 104

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Side Rectangular Duct	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
8	6.9	7.6	8.2	8.8																
9	7.3	9.0	8.7	9.3	9.9															
10	7.7	8.4	9.2	9.8	10.4	11.0														
11	8.0	8.8	9.6	10.2	10.9	11.5	12.1													
12	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2												
13	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3											
14	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4										
15	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5									
16	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	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	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	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	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0				
22	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2		
24	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	25.9	26.4
26	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	26.9	27.5
28	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	27.9	28.5
30	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	28.8	29.5
32	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	29.8	30.5
34	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	30.7	31.3
36	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	31.5	32.2
38	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	32.3	33.1
40	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	33.1	33.9
42	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	33.8	34.5
44	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	34.5	35.3
46	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	35.4	36.2
48	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	36.1	37.0
50	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	36.8	37.6
52	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	37.4	38.3
54	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.0	38.9
56	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	38.7	39.6
58	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	39.4	40.3
60	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.0	40.9
62	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	40.6	41.6
64	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	41.2	42.2
66	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	41.8	42.8

TABLE 104 (*continued*)

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Side Rectangular Duct	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	42	44	46	48
26	28.1	28.6																		
28	29.1	29.7	30.2	30.8																
30	30.1	30.7	31.3	31.9	32.5	33.0														
32	31.1	31.7	32.3	32.9	33.5	34.1	34.6	35.2												
34	32.0	32.7	33.3	33.9	34.5	35.1	35.7	36.3	36.8	37.4										
36	32.9	33.7	34.3	34.9	35.5	36.1	36.7	37.3	37.9	38.5	39.1	39.6								
38	33.8	34.6	35.3	35.9	36.5	37.1	37.7	38.4	39.0	39.5	40.1	40.7	41.2	41.8						
40	34.6	35.3	36.0	36.7	37.4	38.0	38.6	39.3	39.9	40.5	41.1	41.7	42.3	42.9	43.3	44.0				
42	35.2	36.0	36.8	37.6	38.3	39.0	39.6	40.3	40.9	41.5	42.1	42.7	43.4	44.0	44.5	45.1	46.2			
44	36.1	36.9	37.7	38.5	39.2	39.9	40.5	41.2	41.8	42.5	43.1	43.7	44.3	44.9	45.5	46.1	47.2	48.4		
46	37.0	37.8	38.5	39.3	40.0	40.8	41.5	42.2	42.9	43.5	44.2	44.8	45.4	46.0	46.6	47.2	48.4	49.5	50.6	
48	37.8	38.5	39.2	40.0	40.8	41.5	42.3	43.0	43.7	44.4	45.0	45.6	46.3	46.9	47.5	48.1	49.3	50.5	51.6	52.8
50	38.4	39.2	40.0	40.8	41.5	42.3	43.0	43.8	44.5	45.2	45.9	46.5	47.1	47.9	48.5	49.1	50.4	51.6	52.9	54.0
52	39.1	40.0	40.8	41.6	42.3	43.1	43.9	44.7	45.4	46.1	46.8	47.5	48.2	48.9	49.5	50.1	51.3	52.5	53.8	55.0
54	39.8	40.7	41.5	42.4	43.2	44.0	44.7	45.5	46.4	47.0	47.6	48.4	49.2	49.9	50.5	51.1	52.3	53.5	54.8	56.0
56	40.4	41.3	42.1	43.0	43.8	44.6	45.4	46.2	46.9	47.7	48.5	49.1	49.9	50.6	51.3	52.0	53.3	54.6	55.9	57.0
58	41.2	42.1	42.9	43.8	44.5	45.4	46.2	47.0	47.8	48.5	49.4	50.0	50.8	51.5	52.2	52.9	54.2	55.5	56.8	58.0
60	41.8	42.7	43.6	44.5	45.4	46.1	46.9	47.8	48.5	49.3	50.1	50.9	51.6	52.3	53.0	53.8	55.0	56.4	57.7	58.9
62	42.5	43.4	44.3	45.1	46.0	46.8	47.6	48.4	49.3	50.0	50.9	51.7	52.4	53.0	53.9	54.5	55.9	57.2	58.5	59.7
64	43.1	44.0	44.9	45.8	46.7	47.5	48.4	49.2	50.0	50.9	51.7	52.4	53.2	53.9	54.7	55.4	56.8	58.1	59.4	60.6
66	43.7	44.7	45.6	46.5	47.3	48.2	49.1	50.0	50.7	51.6	52.4	53.1	53.9	54.7	55.5	56.2	57.6	59.1	60.4	61.6
68	44.4	45.3	46.3	47.2	48.0	48.9	49.7	50.7	51.4	52.2	53.1	53.8	54.6	55.5	56.2	56.9	58.4	59.9	61.3	62.6
70	45.0	46.0	46.9	47.8	48.7	49.5	50.4	51.3	52.0	52.9	53.7	54.5	55.4	56.2	57.0	57.7	59.1	60.6	62.1	63.5
72	45.5	46.5	47.5	48.4	49.3	50.1	51.0	51.9	52.8	53.7	54.6	55.4	56.2	57.0	57.8	58.7	60.0	61.3	63.0	64.5

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Table 105

CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION

Side Rectangular Duct	8	8.5	9	9.5	10	10.5	11	11.5	12	12.5	13	13.5	14	14.5	15	15.5	16
3	5.2	5.4	5.5	5.7	5.8	5.9	6.0	6.2	6.3	6.4	6.5	6.6	6.7	6.8	6.9	7.0	7.1
3.5	5.7	5.9	6.0	6.2	6.3	6.5	6.6	6.7	6.9	7.0	7.1	7.3	7.4	7.5	7.6	7.7	7.8
4	6.1	6.3	6.5	6.7	6.8	7.0	7.1	7.2	7.4	7.5	7.7	7.8	7.9	8.1	8.2	8.3	8.4
4.5	6.5	6.7	6.9	7.1	7.2	7.4	7.6	7.7	7.9	8.0	8.2	8.4	8.5	8.6	8.7	8.9	9.0
5	6.9	7.1	7.3	7.5	7.7	7.8	8.0	8.2	8.3	8.5	8.7	8.8	8.9	9.1	9.2	9.4	9.5
5.5	7.3	7.5	7.7	7.8	8.1	8.3	8.5	8.6	8.8	9.0	9.2	9.4	9.5	9.6	9.8	9.9	10.1

Side Rectangular Duct	50	54	60	66	72	78	84	88
50	55.0							
52	56.1							
54	57.2	59.4						
56	58.3	60.5						
58	59.3	61.6						
60	60.3	62.7	66.0					
62	61.3	63.7	67.1					
64	62.2	64.7	68.2					
66	63.2	65.7	69.3	72.6				
68	64.1	66.6	70.3	73.7				
70	65.0	67.6	71.3	74.8				
72	65.9	68.5	72.3	75.9	79.2			
74	66.8	69.4	73.3	76.9	80.3			
76	67.6	70.3	74.2	77.9	81.4			
78	68.4	71.2	75.2	78.9	82.5	85.8		
80	69.2	72.1	76.1	79.9	83.6	86.9		
82	70.1	73.0	77.1	80.9	84.6	88.0		
84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

read velocity directly. The instrument must be handled carefully and frequently calibrated. Errors amounting to 15% are not uncommon. Anemometers are useful for comparison, and for such purposes as equalizing the flow from air registers or grilles, need not be exactly true to standard.

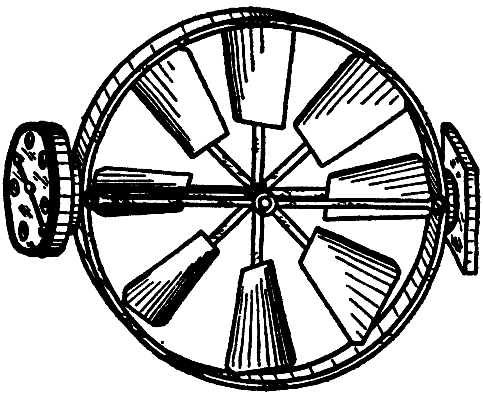


FIGURE 14
Anemometer

191. The Pitot Tube

The Pitot tube may be used to determine the velocity of air, but is more often used to determine the quantity of air passing through a duct. A layout of the equipment used is shown in Figure 15. The open Pitot tube "B" is set square against the direction of the air through the duct and so is exposed to a pressure equal to the velocity head plus the static head $h_v + {}_s$. Tube "A" is exposed only to the static head h_s. By connecting the tubes "A" and "B" to a U-tube or manometer, as shown in "C", the difference in level of the liquid used (usually water) indicates the difference of the two pressures which is equal to the velocity head or h_v in inches of water, in the following formula:

Q = Quantity, cubic feet per minute

v = Velocity of air, feet per second
V = Velocity of air, feet per minute
h_v = Velocity head, inches of water
H_v = Velocity head, feet of air
d = Diameter of duct, feet
A = Area of duct, square feet
g = Acceleration due to gravity = 32.2

$v = \sqrt{2gH_v}$
$H_v = h_v \times 69$ (approximately)
$V = 60 \times v$

$Q = AV$
$A = 0.7854d^2$

Example:
h_v = 0.5 inch water
d = 1.25 feet

$H_v = 0.5 \times 69$
$= 34.5$
$V = 60\sqrt{2 \times 32.2 \times 34.5}$
$= 60\sqrt{2230}$
$= 60 \times 47$
$= 2820$ ft per minute

$A = 0.7854 \times 1.25 \times 1.25$
$= 1.225$ sq ft
$Q = 2820 \times 1.225$
$= 3450$ cu ft per minute

This formula gives the theoretical quantity of air. To obtain the exact quantity, this result must be multiplied by a factor which takes into consideration the diameter of the duct and the roughness of its surface. Where the exact results are desired, several readings should be taken at points in equal areas of the duct and an average reading calculated.

192. Fan and Blower Tests

The Pitot tube method of measuring air has been incorporated in the following test code for rating fans and blowers:

TABLE 106

STANDARD CODE FOR TESTING CENTRIFUGAL FANS AND BLOWERS

The following nomenclature is used:
(1) Standard Air is air weighing 0.07488 pound per cubic foot. This weight corresponds to air having a barometric pressure

TABLE 106 (*continued*)

of 29.92 inches of mercury, a dry-bulb temperature of 68° F and 50% humidity.

(2) Static Pressure (SP) is measured at right angles to the direction of flow.

(3) Total Head or Total Pressure (TP) is measured by an impact tube.

(4) Velocity Head or Velocity Pressure (VP) is the difference between the total pressure and the static pressure.

(5) Capacity (CFM) is the cubic feet of air per minute handled by the fan.

(6) Horsepower Output (AHP) of a fan, or air horsepower, is expressed by the formula:

$$\text{(I)} \quad AHP = \frac{CFM \times TP}{6356}$$

where

CFM = capacity in cubic feet per minute
TP = total pressure in inches of water

(7) Horsepower Input (HP) is the horsepower required to drive the fan.

(8) Mechanical Efficiency (ME) of a fan is the ratio of horsepower output to horsepower input.

(9) Static Efficiency (SE) is the mechanical efficiency multiplied by the ratio of static to total pressure.

(10) Fan Performance is a statement of the capacity, pressure or pressures, speed, and horsepower input.

(11) Fan Characteristic is a graphical presentation of fan performances throughout the full range from free delivery to no delivery at constant speed.

(12) Fan Discharge or Outlet is the place provided for receiving a duct through which air leaves the fan.

(13) Fan Inlet is the place provided for receiving a duct through which air enters the fan.

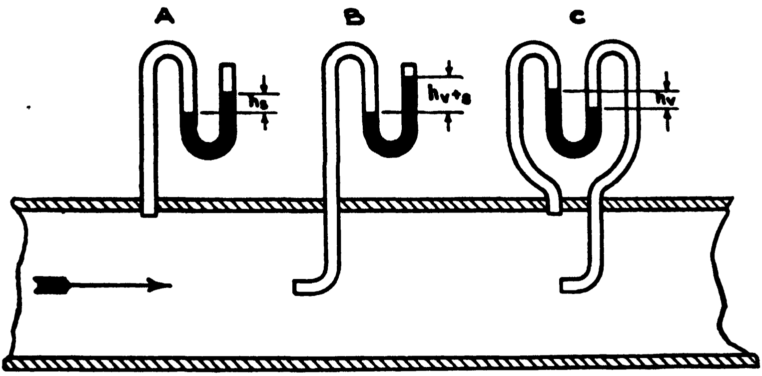


FIGURE 15
Illustrating Principle of Pitot Tube

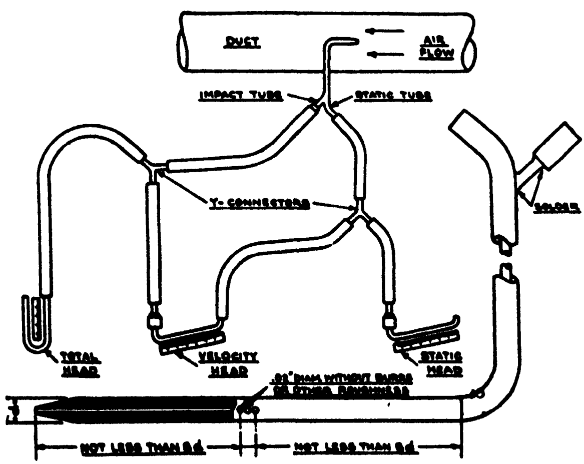


FIGURE 16
Pitot Tube and Connections

Chapter XI

ELEMENTS OF HEALTH AND COMFORT

The intangible matters of habit and suggestion are two influences in air conditioning that cannot be ignored. Some persons regard the unnecessary endurance of low temperatures as a virtue. Our present knowledge of air conditioning does not justify this view, as extreme conditions of temperature involve a strain upon the heat-regulating system of the human body. A fresh-air fiend may suffer in a room with windows closed even if the air is odorless, clean and at the proper temperature and humidity. Old and fallacious beliefs were that human discomfort was a consequence of excess of carbon dioxide in the air of an enclosure. Due to acclimatization or human adaptation to changing air conditions, it is practically impossible to set up invariable standards that will guarantee comfort for every occupant of an air-conditioned room. All that can be done is to prepare standards of recommended practice based not only upon laboratory tests on selected subjects but also on the opinion of paying customers, clients or patrons.

193. Ventilation Standards

The original meaning of the word *ventilation* was merely air change; to ventilate meant to displace the air within a building by outside air. Now ventilation has come to mean more than mere displacement of air and involves the maintenance of conditions essential to the comfort and health of occupants which is usually termed *air conditioning*. This topic is devoted to a discussion of the ventilation of public buildings in the original sense of the word.

Ventilating systems are classified as *natural* and *mechanical.* The objects of ventilation are

(1) To supply oxygen for respiration
(2) To remove odors
(3) To remove toxic substances
(4) To remove body heat and body moisture.

There are two standards upon which quantities may be based.

(1) Body heat removal
(2) Elimination of body odors

The most logical basis is that of body-heat removal. Each person at rest gives off approximately 5 Btu sensible heat per minute at 70° F. This is sufficient to raise 277 cu ft of air one degree per minute. If a 10 degree rise is allowed, then 30 cu ft of air per minute per person would be required. For air-conditioning problems a more exact determination should be made. Body odors can be removed at as low as 7½ cfm per person.

Natural ventilation is usually permissible when the following requirements are fulfilled:

1. Openings in windows and skylights equal to 5% of floor area.
2. At least 50 sq ft of floor area per person.
3. At least 500 cu ft of air space per person.

In all other cases positive supply should be provided. Air supply for chemical laboratories and toilets should be entirely separate from any other supply or exhaust system and should remove at least 2 cu ft of air per minute for each square foot of floor area. The air supply or exhaust for kitchens should also be separate, allowing one air change per minute.

While this topic particularly refers to non air-conditioned spaces, it may be well to note here that in metropolitan night clubs and restaurants that are air-conditioned it has been proved that 15 cfm of outside air per non-smoker and 40 cfm per each smoker is the minimum allowance.

Local municipal ventilation codes in every case should be examined before specifying requirements.

TABLE 107

VENTILATION REQUIREMENTS

Application	cfm Per Person Preferred*	Minimum*
Apartment‡	15	10
Banking space	10	7½
Barber Shop	10	7½
Beauty Parlor	10	7½
Broker's Board Room	30	20
Cocktail Bar	20	15
Department Store	7½	5
Directors' Room	30	30
Funeral Parlor	10	5
Hospital Room‡	15	10
Hotel Room	15	10
Office, General	15	10
Office, Private‡	30	15
Restaurant	15	12
Retail Shop	10	7½
Theater	7½	5
NOTE: For General Application		
Each person, not smoking	7½	5
Each person, smoking	40	25

‡ Figure at least one person for each 50 ft of floor area, but do not pyramid such loads for multiple rooms beyond the maximum simultaneous peak.

* Whether the preferred or minimum values are used, the outside air ducts shall be sized to admit at least 50% more than the preferred values.

Courtesy of A.S.H.V.E.

194. Respiration

An adult breathes ¼ to ⅓ cu ft of air per minute in quiet breathing. A normal inspiration consists of 24½ cubic inches called *tidal air*. After expiration the lungs still contain about 40 cu in. residual air. Expired air is at body temperature (98.8° F), is saturated with water vapor, contains 4% more carbon dioxide and 5% less oxygen than the inspired air. In addition, expired air may contain small traces of hydrogen and ammonia. It is entirely free of floating matter (dust). In large manufacturing centers 23 oz of dust are deposited in the respiratory passages over a period of a year. Nature removes this dust by mucous discharge. In 24 hours an adult expires 1¾ pounds of carbon dioxide, about 5% of the net weight of air inspired in 24 hours.

195. Body Heat

The following table (from Stewart's *Physiology*) gives an analysis of the heat loss in 24 hours from the body of an average adult at 70° F air temperature.

TABLE 108

ANALYSIS OF BODY HEAT LOSS (STEWART)

Source	Per Cent
Evaporation	15
Skin Radiation	30
Conduction and Convection	35
Evaporation	15
Heating Expired Air	5
	100

TABLE 109

HEAT OF PERSONS—BTU/HOUR

	ACTIVITY CLASS A Per Person Sitting or Moving Slowly			ACTIVITY CLASS B Per Person Working or Dancing		
Design Room Temperature	Sensible Heat Btu per Hour	Latent Heat Btu per Hour	Total Heat Btu per Hour	Sensible Heat Btu per Hour	Latent Heat Btu per Hour	Total Heat Btu per Hour
84	180	220	400	150	510	660
82	200	200	400	180	480	660
80	220	180	400	210	450	660
78	240	160	400	240	420	660
76	256	144	400	270	390	660
74	272	128	400	300	360	660
72	286	114	400	325	335	660
70	300	100	400	350	310	660

Data from A.S.H.V.E. Research ***Courtesy of A.S.H.V.E.***

196. Effective Temperature

Sensations of warmth or cold depend not only on the temperature of the surrounding air but also upon the relative humidity. Dry air at a relatively high temperature may feel cooler than damp air at a considerably lower temperature. Combinations of humidity, temperature and air motion which induce the same feeling of warmth by selected subjects are called thermo-equivalent conditions. A series of tests at the A.S.H.V.E. Laboratory established equivalent conditions as they occur in general air-conditioning practice. Effective temperature is an empirical scale of temperatures established by a jury of the selected subjects mentioned above. It was established, for example, that 75° F dry-bulb and a 50% relative humidity is of equal comfort to 80° F dry-bulb and 15% relative humidity and that both of these conditions are equivalent to 70° F effective temperature; as shown on the comfort chart, Figure 17.

197. Body Temperature

The body has a normal temperature of 98.6° F, as the resultant of heat energy received, mainly through food, and heat lost from the skin and lungs to the surroundings. Coordination of the different heat-regulating mechanisms varies with air conditions, internal heat production (metabolism) decreasing in warmer environments and increasing in colder ones. Measurements show, for example, that for men at rest, if the effective temperatures rises from 60 to 100° F, the temperature of the face rises about 3° F, while the pulse rate increases 40 (over a period of an hour). Measurements of bodily temperatures and loss by perspiration have also been made for men working and at rest, as well as the decrease in work that can be accomplished in a rising temperature (the ratio being about four to one for the case just cited).

The effect of temperatures fixed below normal appears to be undesirable, while various indices for large numbers of people,

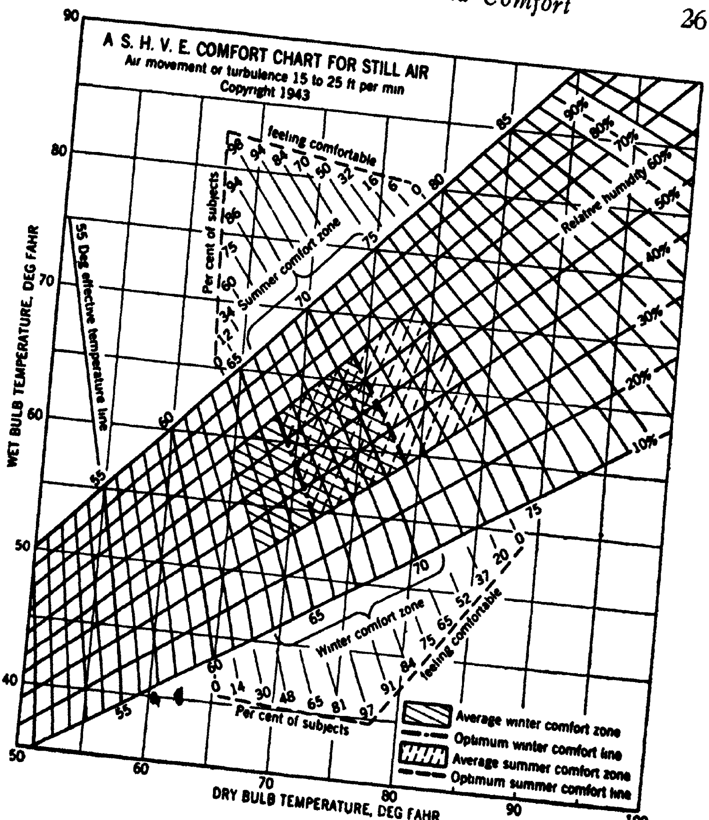


FIGURE 17

A.S.H.V.E. Comfort Chart for Still Air

Note—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours. The optimum summer comfort line shown pertains to Pittsburgh and to other cities in the northern portion of the United States and southern Canada, and at elevations not in excess of 1,000 feet above sea level. An increase of one degree ET should be made approximately per 5 degree reduction in north latitude.

Courtesy of A.S.H.V.E.

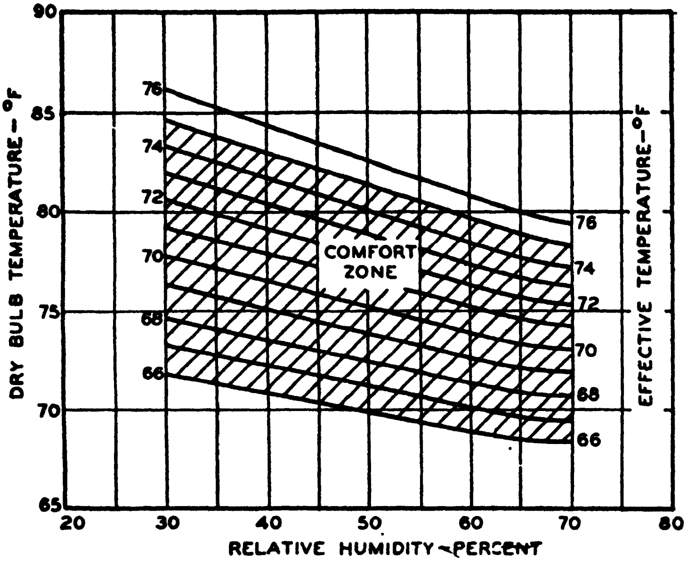


FIGURE 18
Comfort Zone Section of Comfort Chart
For air motion 15 to 25 ft per minute, using Basic Effective Temperature.

such as frequency of illness, accident rate, and production rate in industrial establishments, have an optimum in the neighborhood of temperatures commonly regarded as comfortable.

198. The Comfort Chart

Figure 18 is the comfort chart representing zones in which the majority of people feel comfortable in air moving at 15 to 25 ft per minute. The period of occupancy, drafts and individual tastes affect the determination of average conditions. It is necessary to determine whether occupants are at rest or in action. Many people do not object to an air movement up to 75 ft per minute, while others consider this much movement to be a draft. Drafts are more noticeable when the moving air is below room temperature than when it is above it. Air should be introduced with great care at satisfactory velocities and at the correct height above the floor. The lower the temperature of the

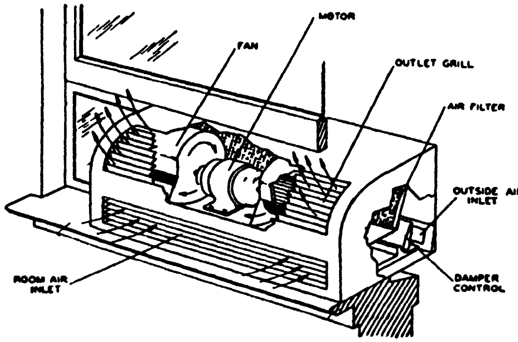


FIGURE 19
Typical Window Ventilator

air, the smaller the quantity required and the better must be the mixing with the room air before it reaches the occupied portions of the room. To avoid accumulation of body odors and tobacco smoke, not more than 75% of the air may be circulated and reconditioned. Air velocities over 100 ft per minute are found to be very objectionable and unpleasant even for 15 minute periods. The comfort chart is only a guide, since the jury best qualified to decide upon comfort conditions consists of the paying customers of the theater, restaurant or store.

199. Air Requirements

The quantity of fresh air required is based on health department regulations regarding a fixed quantity per occupant, a theater only half filled requiring half the amount of fresh air required when it is full.

The total quantity of air handled by the system is related to such factors as heat and moisture given off by the occupants, lights, electrical equipment, infiltration, sun effect, air changes required, exhaust for toilets, booths and lobby, and other factors affected by local conditions. This quantity is always maintained constant, regardless of the number of occupants, but the ratio of fresh air to recirculated air varies according to the number of occupants present.

TABLE 110

PREFERRED SCHEDULE OF INDOOR AIR CONDITIONS CORRESPONDING TO OUTDOOR SUMMER DRY-BULB TEMPERATURES

(Period of occupancy over 40 min; individuals engaged in sedentary or light muscular activity)

Outdoor Dry-Bulb Temperature, ° F	Indoor Air Conditions			
	Effective Temperature	Dry-Bulb, ° F	Wet-Bulb, ° F	Relative Humidity, %
100	75	83	66	40
	75	82	67	45
	75	81	68	51
	75	80	70	60
95	74	82	64	36
	74	81	66	44
	74	80	67	51
	74	79	68	57
	74	78	70	68
90	73	81	63	36
	73	80	64	41
	73	79	66	50
	73	78	67	56
85	72	80	61	32
	72	79	63	41
	72	78	64	46
	72	77	66	56
80	71	78	61	36
	71	77	63	45
	71	76	64	52
	71	75	66	61

TABLE 111

PHYSIOLOGICAL EFFECTS OF GASES

Gas	Parts Per 10,000 Rapidly Fatal	Parts Per 10,000, Maximum Allowable for Prolonged Exposure
Aniline	—	0.1
Ammonia	50–100	1.0
Arsine	2.5	—
Benzene	190	1.5–3.0
Carbon bisulfide	—	0.5
Carbon dioxide	800–1000	—
Carbon monoxide	40	1.0
Chlorine	10	0.01
Gasoline	243	—
Hydrochloric acid	10–20	0.1
Hydrofluoric acid	2	0.03
Hydrocyanic acid	30	0.2
Hydrogen sulfide	10–30	1.0
Lead vapor	—	5–6
Nitrobenzene	—	0.002
Nitrous fumes	2.5–7.5	0.33
Phosgene	0.25	0.01
Phosphine	20	—
Sulfur dioxide	4–5	0.1
Toluene and xylene	190	1.5–3.0

Tables 110 and 111 by courtesy of A.S.R.E.

200. Fresh and Recirculated Air (Central Systems)

With central systems a difference of more than 10 to 15° F, between the air introduced and the condition maintained in the room is conducive to drafts and discomfort. Therefore, if a temperature of 75 to 80° F is to be maintained, air should be introduced at 65 to 68° F dry-bulb. See Tables 180 and 181, Chapter XX.

Provided that smoking is not allowed, about 30 cu ft circulated per minute per occupant generally gives a ten degree difference. This air consists of about 22 cu ft per minute of recirculated air and 8 cu ft per minute of fresh air. In restaurants, a supply of fifteen or more cubic feet of fresh air per minute per occupant is preferred.

It is general practice to assume a difference of 20 degrees in the application of self-contained units up to ten tons capacity.

201. The Four Vital Factors

From the preceding discussion, it is clear that thermal environment cannot properly be adjusted to the requirements of human health and comfort without control of all four of the basic factors: air temperature (free from radiation effects), air movement, humidity and mean radiant temperature of surroundings.

TABLE 112

SUMMER WET-BULB TEMPERATURE IN ARTIFICIALLY COOLED ROOMS *

Summer Wet-Bulb Temperature Light Work	Summer Wet-Bulb Temperature At Rest	Air Velocity Feet Per Minute in Zone of Occupancy
60½	62	50 (good)
61½	63	100 (poor)
62½	64	150 (objectionable)

In winter the wet-bulb temperature may be 4 to 5 degrees lower.

* Dr. E. Vernon Hill.

202. Air Turbulence

It will be noted that the air velocity in feet per minute increases as the wet-bulb increases. In Table 112, air is assumed to flow in one direction from the grille; however, any velocity in the zone of occupancy above 150 ft per minute causes objectionable draft, especially if the person is sitting and the draft continually strikes only one portion of the body. A velocity of 500 ft per minute for a period of one minute with a change of direction, is not as objectionable as a velocity of 150 ft per minute in one direction. Therefore an oscillating fan effect, causing an air turbulence, is desirable.

Table 113

MINIMUM OUTDOOR AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS

(Provisional values subject to revision upon completion of work)

Type of Occupants	Air Space per Person, Cubic Feet	Outdoor Air Supply Cubic Feet per Minute per Person
Heating season with or without recirculation. Air not conditioned.		
Sedentary adults of average socio-economic status	100	25
Sedentary adults of average socio-economic status	200	16
Sedentary adults of average socio-economic status	300	12
Sedentary adults of average socio-economic status	500	7
Laborers	200	23
Grade school children of average class	100	29
Grade school children of average class	200	21
Grade school children of average class	300	17
Grade school children of average class	500	11
Grade school children of poor class	200	38
Grade school children of better class	200	18
Grade school children of best class	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary Adults	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person.		
Sedentary Adults	200	<4

Data from A.S.H.V.E. Guide.

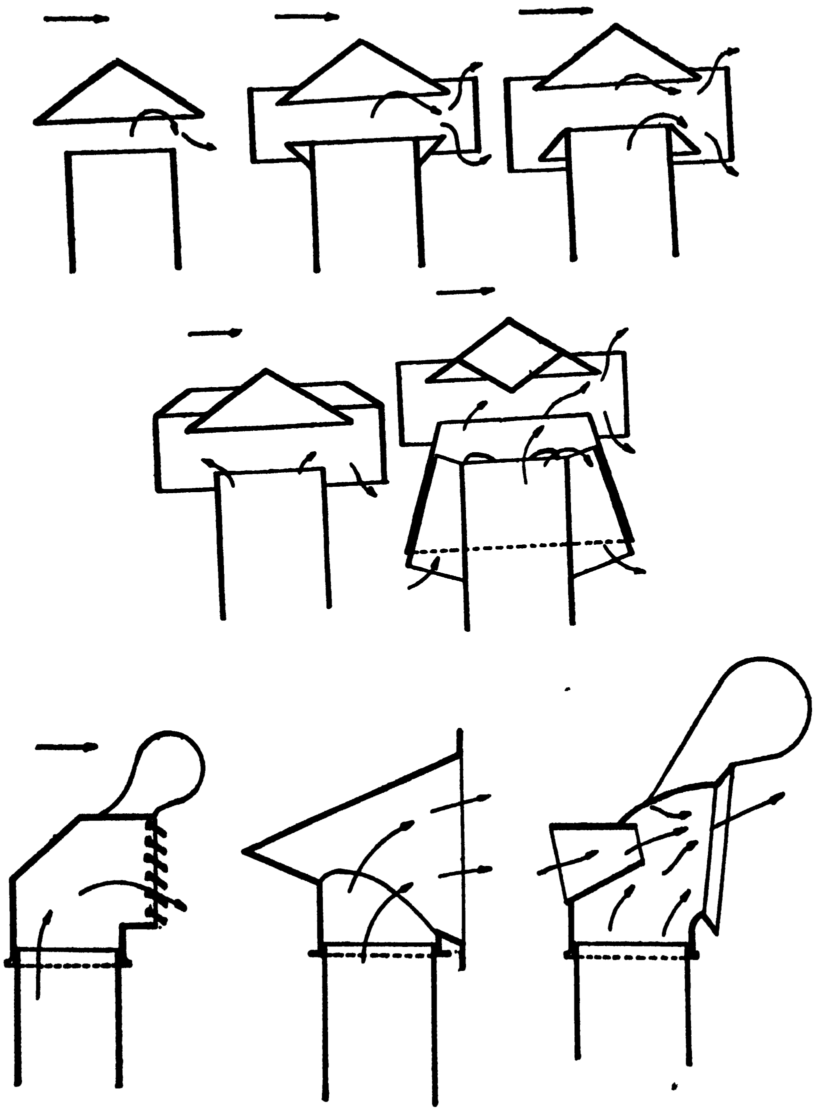


FIGURE 20
Typical Ventilators
Above, five common types of stationary ventilators; below, three typical oscillating ventilators.

The velocity of air striking a wall in front of a grille should not exceed 50 ft per minute downward, when measured at or below the five-foot level. Air may leave a grille at 1,000 ft per minute at 8 ft above floor level. The velocity in such a case would be reduced to about 200 ft per minute at a distance of 20 ft and perhaps 100 ft per minute at a distance of 30 ft, all above the six-foot level. However, when the air drops to the occupied zone, its velocity should average about 50 ft per minute up to within a few feet of the return grille.

203. Insulation Against Sound for Air-Conditioning Systems

Almost any air-conditioning system will be unsatisfactory if it is noisy in operation, regardless of how excellently it may operate in all other respects. Therefore noise must be specially considered. Noise may be of two kinds: air noise and machine noise. The air noise is the result of too high an air velocity and is similar to the roaring produced during periods of high winds. Machine noise is produced by moving parts of the fan, belt drive or motor and may be due to many different causes such as an unbalanced fan wheel, defective bearings, pulleys out of line, noisy operating motors, and other incorrect features. The designer should first select equipment which normally should produce but little noise and then he must insulate this noise as far as possible, so that it will not enter the occupied portion of the building and become a nuisance. If the ducts have been designed with velocities ranging from 400 fpm to 1,000 fpm, as previously recommended, and if the ductwork has been properly fabricated with easy radii on the turns and bends, no air noise will result, but it is possible that the passage of the air may set up a vibration of the sheet metal sides, especially on large ducts. Insulation of the duct will prevent this, therefore this is only another argument for duct insulation.

If the fan has been selected with tip speeds and outlet velocities which are not excessive, there should be no air noise from this source. On the other hand, it is impossible to operate a large fan wheel without any noise at all, and to prevent such small noise as may be developed, a canvas connection should be used between the fan outlet and the duct to prevent telephoning this noise along the ducts.

A certain amount of vibration also will result from the operation of the fan and motor and to prevent the transmission of this into the building, the fan and motor should be mounted on a wood frame which is lag-screwed to the fan and attached to the fan foundation by countersunk bolts. The fan foundation should be integral with the motor foundation and should weigh at least three times as much as the combined weight of the fan and motor, in order to have sufficient mass to absorb this vibration. This base is set on 2 in. of cork on the bottom and has 1 in. of cork all around the sides; it is held in position by a 4 in. concrete curb brought up all around the foundation so that the top is at least 6 in. above the bottom of the fan foundation and the top edges of the cork are finished off with about ½ in. of mastic. This makes an excellent type of foundation and should prevent trouble from both noise and vibration. If an air washer is used for cooling, the entire bottom of the pan should be set on cork 2 in. thick.

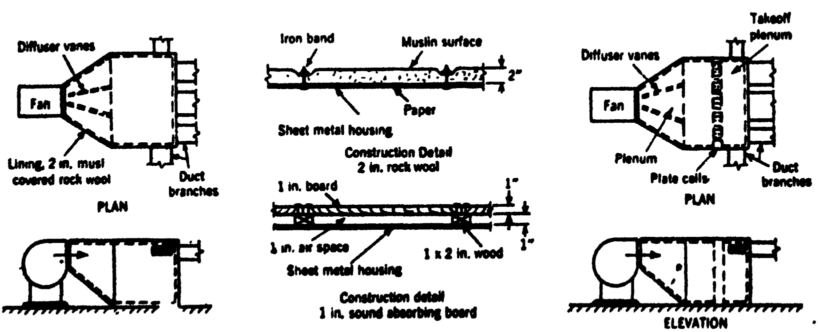


FIGURE 21
Absorption Plenums With and Without Sound Cells.
Courtesy of A.S.H.R.V.E.

Heating and cooling coils should be supported on concrete piers or structural steel supports, but no sound insulation is necessary in these cases. Filters do not require any sound insulating.

204. Sound-Proofing the Compressors

With mechanical refrigeration the only machines requiring sound insulating are the compressors, and, possibly, the circulating pump. In the case of the compressors the foundation may be constructed in the same manner as described for the fan, with the foundation weighing from three to six times the weight of the compressors, and with 4 in. of cork below and 2 in. of cork around the sides. The circulating pump may be set on 2 in. of cork with a wood frame on the top, to which the pump is attached. Wood frames usually are made of 4 in. x 6 in. hard pine for large machines and 3 in. x 5 in. hard pine for small machines such as the circulation pump. In some cases, sand is substituted in place of cork for the compressors and it will be found that, if the compressors are installed on a floor resting in the ground, this is perfectly satisfactory.

205. Correcting Old Systems

Elimination of noise from old systems, which were basically wrong in design, is a difficult job and usually expensive. When air noise is the trouble, the insertion of acoustical treatment in the duct outlets for a distance of about 10 ft back from the outlet often will remedy the trouble. The introduction of a short length of acoustical treatment near the fan also may be a simple solution. In this case, stiff wire mesh is bent into hollow tubes about 6 in. x 6 in. and wrapped with felt; these are then packed into the duct as tightly together as possible and an enlargement made in the duct so that the net amount of opening through the tubes equals the full area of the duct; these tubes usually are made from 4 to 6 ft in length and will absorb all noise produced on the fan side of the acoustical treatment but will not help with air noise produced beyond the acoustical treatment. If there is no canvas joint between the fan outlet and the duct, one should be inserted.

Where the fan speed is too high, it is a very difficult matter to correct noise, as a reduction in speed will be accompanied by a proportional reduction in the amount of air delivered. In cases where the air supply is in excess of the minimum requirements, some reduction in fan speed could be made; otherwise the only solution would be to install a larger fan so that a lower speed could be used to obtain the same quantity of air. Where the air supply is inadequate more air may be obtained by increasing the speed of the fan, but with the expenditure of more motor horsepower; this may necessitate the changing of the motor, probably with the production of a greater amount of noise which may require acoustical treatment.

Where the noise is of a mechanical nature, the fan, drive and motor should be inspected to see that they are properly balanced and in line and that the bearings are in good condition. A burnt-out coil in the motor will produce a motor noise which is difficult to diagnose without the services of a motor expert. Worn belts often result in noise in the drive. Improper foundations may

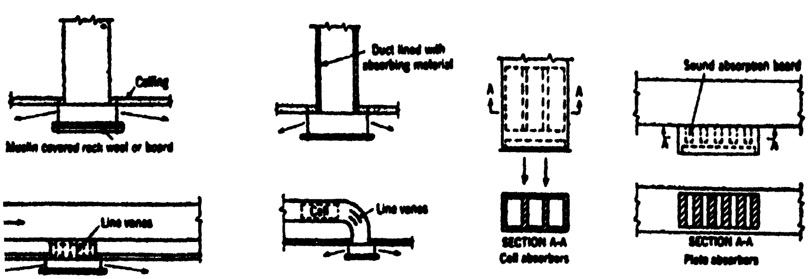


FIGURE 22
Outlet Cells for Pan Outlets or Grilles
Courtesy of A.S.H.V.E.

transmit small noises to the building in a manner resulting in complaints.

206. Acoustical Treatment Inside of Ducts

Certain acoustical treatments involve the lining of the inside of the duct with felt or some similar material, to absorb the noise as the air passes through the duct; such treatments have the following objections: In the first place, they occupy a certain area on the inside of the duct and actually increase the air velocity; second, it is a difficult treatment to apply properly when the ducts get so small that a man cannot work inside of the duct; third, it is practically impossible to inspect such work after it is complete, so as to know if it is done properly or not, and last, no one is in a position to guarantee that any form of cement will have, ten years hence, sufficient holding power to prevent the insulation on the top of the inside of the duct from falling down and partially or completely blocking the air flow. The combination of all of these factors makes the use of this type of insulation very questionable unless thorough inspection can be provided at the time of its installation and unless a mechanical means of support is provided to insure that the material will stay in place as long as the duct is in existence.

207. Unit of Noise Measurement

In the United States and England the unit of noise measurement is the decibel (db); in Germany this unit is called the Phon. The most satisfactory method of measuring noise is by means of a sound-level meter which usually consists of a microphone, a high gain audio-amplifier, and a rectifying milliammeter which will read directly in decibels.

The following table is compiled from information which represents the best opinion on satisfactory noise levels:

TABLE 114

TYPICAL NOISE LEVELS

Rooms	Noise Level in Decibels to be Anticipated Minimum	Representative	Maximum
Sound Film Studios	10	14	20
Radio Broadcasting Studios	10	14	20
Planetarium	15	20	25
Residence, Apartments, etc.	33	40	48
Theaters, Legitimate	25	30	35
Theaters, Motion Picture	30	35	40
Auditoriums, Concert Halls, etc.	25	30	40
Churches	25	30	35
Executive Offices, Acoustically Treated Private Offices	30	38	45
Private Offices, Acoustically Untreated	35	43	50
General Offices	50	60	70
Hospitals	25	40	55
Class Rooms	30	35	45
Libraries, Museums, Art Galleries	30	40	45
Public Buildings, Court Houses, Post Offices, etc.	45	55	60
Small Stores	40	50	60
Upper Floors Department Stores	40	50	55
Stores, General, Including Main Floor Dept. Stores	50	60	70
Hotel Dining Rooms	40	50	60
Restaurants and Cafeterias	50	60	70
Banking Rooms	50	55	60
Factories	65	77	90
Office Machine Rooms	60	70	80
Vehicles			
Railroad Coach	60[a]	70	80
Pullman Car	55[a]	65	75
Automobile	50	65	80
Vehicular Tunnel	75	85	95
Airplane	80	85	100

[a] For train standing in station a level of about 45 db is the maximum which can ordinarily be tolerated.

208. Conclusions

Sound and noise must be guarded against in every feature of the air-conditioning design and effective means must be taken to eliminate or prevent sound or noise from reaching the occupied portions of the building. The overlooking of this requirement has brought disaster on more than one air-conditioning system. The noise level on the discharge of a first class unit type air conditioner is about 20 decibels with the spray off and about 24 decibels with the spray on.

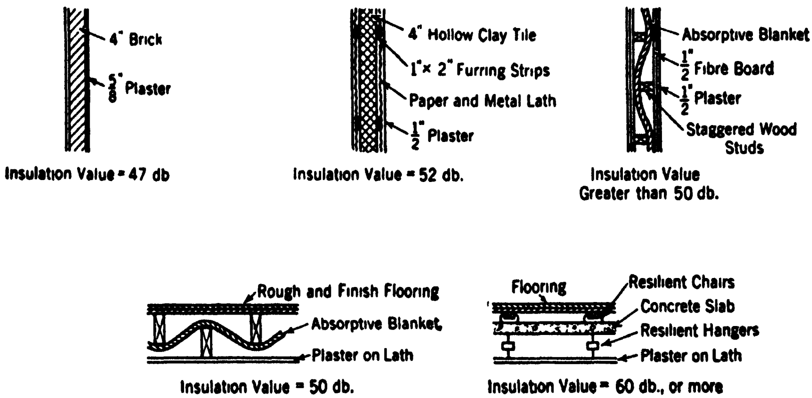


FIGURE 23
Three Wall Sections and Two Floor and Ceiling Sections Which are Suitable for the Insulation of Equipment Rooms [a]

[a] *Acoustical Problems in the Heating and Ventilating of Buildings,* V. O. Knudsen *(A.S H.V.E. Transactions,* **38,** *1932).*

209. Air Cleaning Devices

In cities and around industrial regions, the air contains a considerable quantity of soot and other solid matter, commonly termed dust. The removal of these impurities is highly desirable for efficient conduct of manufacturing processes and for general cleanliness.

Observations have shown that practically all atmospheric impurities are less than 5 microns in size. (One micron equals 0.001 mm 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.

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 surburban 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

210. Air Cleaner Requirements

An air cleaner should fulfill the following requirements:

1. Efficiency in removal of dust, dirt, pollens and bacteria.
2. Efficiency over a range of air velocities.
3. Low frictional resistance to air flow.
4. Large dust-holding capacity without excessive increase in resistance.
5. Ease of cleaning.
6. Ability to discharge exit air free from entrained moisture or charging liquids.

The A.S.H.V.E. has developed a code and classification, which explains how such devices are rated.

211. Classification 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 also be classified as follows:

1. According to principle of air cleaning.
   a. Viscous air filters.
      (1) Unit type.
      (2) Automatic type.
   b. Dry air filters.
   c. Air washers.
   d. Electrical precipitators.

2. According to application.
   a. For central fan systems of ventilation and air conditioning. Filters of the automatic or semi-automatic type, as well as the non-automatic viscous unit or dry 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 or viscous type, adapted for placement 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 or automatic 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.
g. For stack gases. Settling chambers, dynamic or electrical precipitators.
h. For exhaust systems. All types.

Air cleaners may be classified further as follows:

1. For general air conditioning. 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.
2. For removal of dusts, smokes and fumes from stack gases. Prevention of atmospheric pollution from this source is of ever increasing importance, sometimes forced legally and frequently used in order to obtain increased efficiency.
3. For removal and collection of industrial dusts from the point of their production through exhaust systems.

212. Air Washers

The primary purpose of air conditioning is to cleanse, cool, and dehumidify or humidify the air. The air washer assists in performing all these functions. However, certain fine dirt particles pass through the conventional washer. New types have been developed for application where the air to be cleaned is extremely dirty. See Capillary Air Conditioner, Figure 56, Chapter XV.

213. Stack Gases

The common foreign matter in stack gases includes fly-ash and unburned carbon particles 100 microns and larger in diameter. It is usually economical to collect the coarser particles in separators, either centrifugal or gravitational.

214. Industrial Filters

Industrial filters are similar to those used for general ventilation. Bag filters up to 2.5 ft in diameter and 30 ft long, hung vertically, are fed through a header; they allow gas to pass out through the sides of the bag and retain dust particles on the inner surface. Depending on the nature of the cloth, retention of fines can be very high if gas velocity is low, about 0.5 to 3 cu ft per square foot per minute.

Readily removable filters, built in small sections, in which the filter media can be replaced, are used to advantage where deterioration is rapid.

The use of several independent units in parallel is important for cleaning each unit separately.

215. Electronics in Air Filtration

Electronic air filtration is a method of removing entrained solids from air or gases by means of electrical attraction. It is based on the principle that an electrically charged dust particle will be attracted to a charged electrode of opposite charge and repelled by one of like charge.

The efficiency of an electronic filter is from 85 to 90% as compared with 20% of the best filtering material. The electronic filter is guaranteed to collect 65 to 70% of the fine dust and smoke that normally passes through felt, paper or other materials.

Tobacco smoke is the most difficult to remove from air and is made up of individual particles approximately 0.00001 inch in diameter.

216. Centrifugal Separators

The force causing settling can be increased many times by giving the air a whirling motion. In centrifugal and cyclone

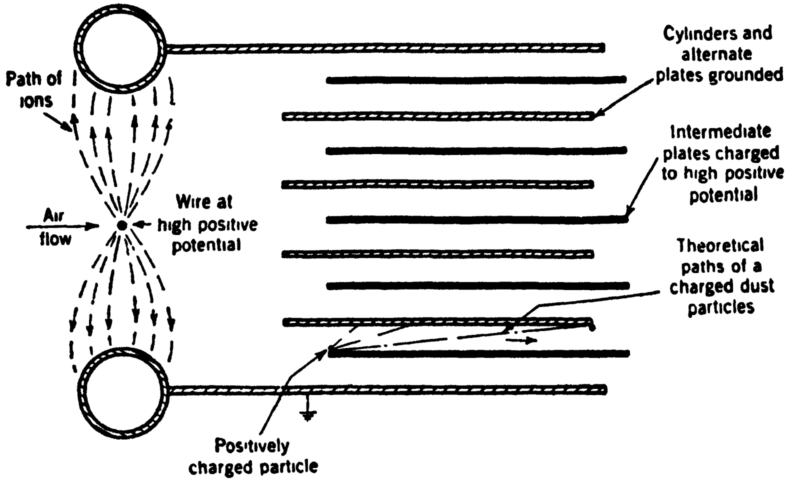


FIGURE 24
Diagrammatic Cross-Section of Electrostatic Precipitator
Courtesy of A.S.H.V.E.

separators air is introduced tangentially into a vertical cylinder and passes out from the center of the top.

217. Exhaust Systems

Quick removal of dust particles caused by such operations as grinding, screening and mixing is accomplished with exhaust systems. Mere discharge to the atmosphere outside the building without collection is frequently of little effect, for incoming air redistributes the objectionable dust.

218. Air Scrubbers

Air scrubbers provide for the removal of coarser particles. The liquid medium should wet the particles and the wetting is a surface-tension phenomenon, specific for each solid-liquid pair. Water effectively wets particles similar to silica; oil wets particles similar to carbon.

Intimate contact between the scrubbing liquid and the dust particles is essential. Fine sprays, baffles and packed chambers are used. The collected material is removed as a thick sludge. Corrosion is serious especially in the case of gases at high temperatures which contain solid constituents.

CHAPTER XII

FANS, DUCTS AND AIR DISTRIBUTION

The satisfactory distribution of air is largely responsible for the success of an air-conditioning system. The problem consists of distributing air as a cooling, heating, drying, moistening or ventilating medium in a specified space within accepted limits of air motion, temperature variation, temperature fluctuation, direction, humidity and noise. Certain limits are determined by experience and research and have generally been accepted by the industry. Variations from accepted standard limits may result in discomfort to occupants of the air-conditioned space. Complaints of discomfort usually arise from draftiness or stuffiness. A draft may be defined as an air current which, because of its temperature, humidity or motion, removes more heat from a human body surface than is generated by metabolic processes. Although stuffiness may be attributed to odors, the complaint of stuffiness usually results from a personal feeling of excess warmth. Outside of localized sensation, such as that caused by a single draft, draftiness and stuffiness may be considered functions of effective temperature. Effective temperature takes in the factors of air temperature, motion and humidity as explained in Chapter XI. Draftiness therefore may be associated with too low an effective temperature, and stuffiness with too high an effective temperature. Therefore, satisfactory conditions for comfort are attained by applying the accepted standard factors of temperature variation, air motion and noise.

219. Definitions *

1. *Supply Opening or Outlet:* Any opening through which air is delivered into a space which is being heated, or cooled, or humidified, or dehumidified or ventilated.

2. *Exhaust Opening:* Any opening through which air is removed from a space which is being heated, or cooled, or humidified, or dehumidified or ventilated.

3. *Outside Air Opening:* Any opening used as an entry for air from outdoors.

4. *Grille:* A covering for any opening and through which air passes.

5. *Damper:* A device used to vary the volume of air passing through a confined cross-section by varying the cross-sectional area.

6. *Multiple Louver Damper:* A damper having a number of adjustable blades.

7. *Single Louver Damper:* A damper having one adjustable blade.

8. *Face:* A grille with provision for attaching a damper.

9. *Register:* A face with a damper attached.

10. *Flange:* The portion (either integral or separate) of a grille, face, or register extending into the duct opening for the purpose of mounting.

11. *Frame:* The portion (either integral or separate) of a grille, face, or register extending around the duct opening for the purpose of mounting.

12. *Margin:* The margin of a grille, face, or register is one-half the difference between the duct dimension and overall dimension measured either horizontally or vertically.

13. *Fret:* The member separating the openings of a grille, face, or register.

14. *Free Area:* The total minimum area of the openings in the grille, face, or register through which air can pass.

* *Courtesy of A.S.H.V.E.*

15. *Core Area:* The total plane area of the portion of a grille, face, or register bounded by a line tangent to the outer openings through which air can pass.

16. *Mean Area:* The total of the core and free areas divided by two.

17. *Duct Area:* The area of a cross-section of the duct based on the inside dimensions at the point where the grille, face or register is mounted.

18. *Percentage Free Area:* The ratio of the free area to the core area expressed in percentage.

19. *Aspect Ratio:* The ratio of length of the core of a grille, face or register to the width.

20. *Throw:* The distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which air motion reduces to 50 fpm.

21. *Envelope:* The outer boundary of an air stream.

22. *Drop:* The vertical distance the lower edge of the air stream drops between the time it leaves the outlet and reaches the end of its throw (h in ft).

23. *Rise:* The converse of drop.

24. *Induction:* The entrainment of room air by the air ejected from the outlet.

25. *Primary Air:* The air leaving an outlet (Q_1 in cfm).

26. *Secondary Air:* The room air picked up by the primary air through induction (Q_2 in cfm).

27. *Total Air:* The mixture of primary and secondary air (Q_3 in cfm).

28. *Induction Ratio:* The total air divided by the primary air equals r, *or* Q_3/Q_1.

29. *Outlet Velocity:* The average air velocity emerging from the outlet (V_1 in fpm) measured at the plane of the opening.

30. *Terminal Velocity:* The average air stream velocity at the end of the throw (V_T in fpm).

31. *Horizontal Spread:* The divergence of the air stream in the horizontal plane after it leaves the outlet (Degrees).

32. *Vertical Spread:* The divergence in the vertical plane (Degrees).

33. *Temperature Differential:* Temperature difference between primary and room air ($t_r - t_{as}$).

34. *Vane Ratio:* The ratio of depth of vane to shortest opening width between two adjacent grille bars.

220. Outlet Performance (A.S.H.V.E. Standards)

The factors of outlet performance, throw or blow, drop, capacity, noise, dirt and room air motion place considerable limitations on the design of a satisfactory distribution system.

1. *Blow.* The blow of wall or ceiling outlets should be selected to cover three-quarters of the distance toward an exposed wall or window as shown in *A* of Figure 27. Overblowing is considerably more serious than underblowing, as an overblow will create objectionable down drafts from any surface it strikes; although underblowing in the case of heated air may be serious in that the warm air may rise too rapidly and thus cause stratification in the occupied zone. In spaces with beamed ceilings, the outlets should be located below the bottom of the lowest beam level, and preferably low enough so that an upward or arched blow may be employed. The blow should be arched sufficiently to miss the beams and, at the same time, in such a manner as to prevent the primary or induced air stream from striking furniture and obstacles producing objectionable drafts. If an outlet is adjusted downward to avoid a beam, cold air may enter the zone of occupancy long before the desired induction has taken place, thus causing serious complaints.

2. *Drop.* The outlets should be located so that the air stream at the termination of the blow is not less than 5 or 6 ft above the floor level. As illustrated in *B* of Figure 27 the maximum permissible blow for a given ceiling height may be obtained by locating the outlet low on the wall, arching the blow, and sweeping the air across the flat ceiling. The air, as it traverses the

room, will adhere to the ceiling. The objection to this method is the possible streaking of the ceiling with dirt.

TABLE 115

RECOMMENDED RETURN GRILLE FACE VELOCITIES

Grille Location	Velocity Over Gross Area Feet per Minute
Above occupied zone	800 up
Within occupied zone, not near seats	600–800
Within occupied zone, near seats	400–600
Door or wall louvers	500–700
Undercutting of doors (through undercut area)	600

3. *Capacity.* Manufacturers' rating sheets may be consulted for selection of the proper number of outlets for a given air quantity. Due to their high induction ratio, ceiling outlets will in general handle more air per outlet than either comparable sidewall or floor outlets.

4. *Noise.* The noise of an outlet is primarily the function of the outlet velocity and size, and secondarily of the outlet construction. The maximum acceptable noise level in a space may completely dictate the permissible outlet velocities that may be employed. (See Chapter XI for discussion of permissible room noise levels and noise generated by outlets.)

5. *Room Air Motion.* The factors leading to high air motion are excessive velocity, high air volume per square foot of outlet wall area, overblow striking beams causing a spilling of the air into the zone of occupancy, and heating in severe climates by means of ceiling outlets which are directed downward.

6. *Dirt.* Although primary air may be carefully filtered, dirt from the conditioned space may be deposited on walls or ceiling wherever there is considerable secondary air motion. With ceiling outlets, dirt streaking may be minimized by carefully

streamlining the discharge of the outlets. With wall outlets, dirt streaking may be minimized by preventing the air from impinging on any ceiling or room surface. Floor outlets may be dirt collectors and objectionable for that reason.

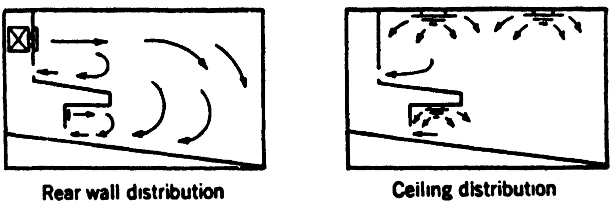


FIGURE 25

Air Distribution Methods for Theaters, Churches and Auditoriums
Courtesy of A.S.H.V.E.

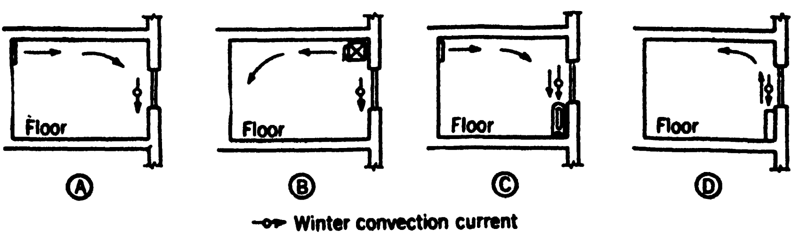
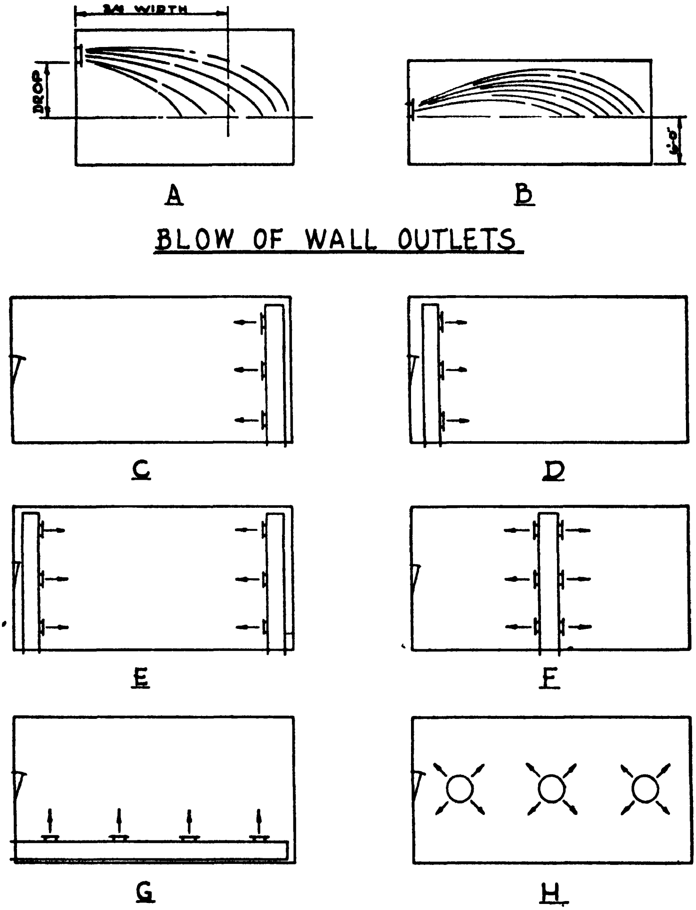


FIGURE 26

Distribution Methods for Small Rooms

A. Satisfactory for cooling. Unsatisfactory for heating in severe climates where the outside temperature is consistently below 40° F, and single glass and uninsulated walls are prevalent.
B. Performance approximately that of A when small diffusers are used in bottom of the duct.
C. Satisfactory for cooling. Satisfactory for heating if direct radiation is properly controlled.
D. Satisfactory for both cooling and heating. The air should be discharged slightly away from the wall, and for low velocities should be fanned out parallel to the wall.

AIR DISTRIBUTION DIAGRAMS

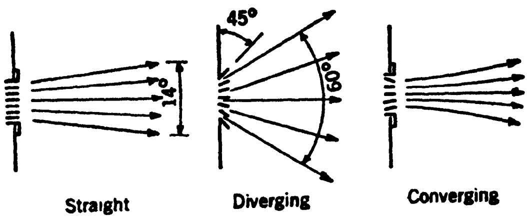


SMALL STORE COOLING DISTRIBUTION

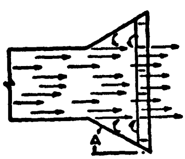
C. REAR WALL: Possibility of excess air motion & drafts due to high outlet velocity.
D. FRONT WALL: Same as "C" and may cause excessive infiltration thru door.
E. FRONT & REAR WALLS: Room air motion moderate. Blow of outlets must not impinge or down drafts will be created in center of store.
F. CENTER: Good results. Room air motion moderate. No impingement of air streams.
G. ONE SIDE: Good results. Blow toward exposed wall. Room air motion moderate.
H. CEILING OUTLETS: Good results. Low room air motion.

HH

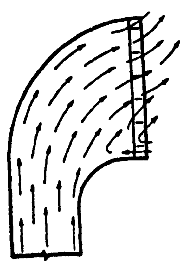
FIGURE 27



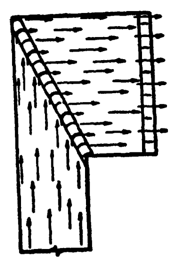
SPREAD OF AIR STREAM WITH VARIOUS VANES



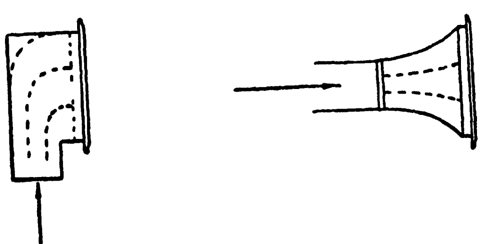
EFFECTS OF EXPANDING DUCT



UNEQUAL FACE VELOCITIES



EFFECT OF TURNING MEMBER



DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

FIGURE 28

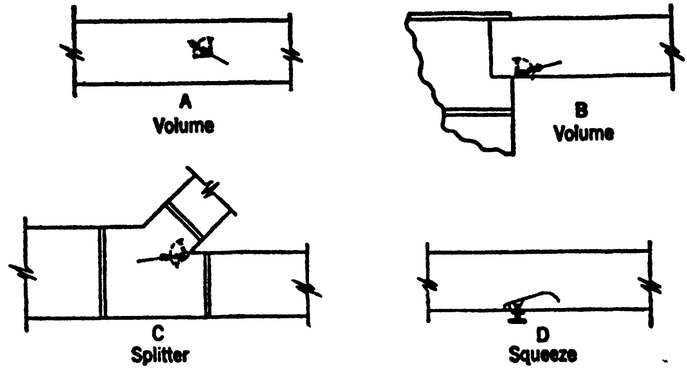


FIGURE 29
Three Types of Dampers Commonly Used for Trunk and Individual Duct Systems

221. Duct and Grille Sizing

The distribution of air from a central air-conditioning apparatus to the various outlets in the room, or building, is one of the most important functions of an air-conditioning system. Air, like everything else, has inertia and in order to circulate the air power must be employed. The circulation is accomplished by the use of a fan, or fans, and the fans in most cases are driven by electric motors through V belts. The characteristics of fans are discussed in a later chapter and in this discussion only the ducts and grilles will be considered. The size of an air duct is determined by two considerations:

a. The quantity of air to be carried by the duct
b. The velocity at which the air is to flow

for it is evident that a given duct will deliver twice as much air if the air is flowing through the duct at the rate of 1,000 fpm as it will if the air is flowing through the same duct at only 500 fpm.

There are two general methods in use for determining the size (or area) required for ducts carrying air, one of these methods being known as the velocity method and the other as the pressure drop method. The velocity method is a much simpler method and will be taken up first.

222. Velocity Method

In the velocity method the duct system is laid out in diagram and all outlets, with the amounts of air to be delivered, in cfm are indicated. These quantities of air are added up from the farthest outlet back toward the fan to obtain the amount of air which must flow through each individual section of the piping, and the quantities are marked on each portion of the piping system. This part of the procedure is the same, no matter which scheme of determining the pipe sizes is to be used.

223. Selecting the Velocities

The velocities to be used are arbitrarily selected so as to favor the far ends of the system where air delivery is likely to fail unless some favoritism is shown. For the present purposes and in most air-conditioning systems, the velocities will not be allowed to exceed 1,000 fpm at the fan, nor will they be allowed to fall below 400 fpm at the farthest outlet. The reasons for selecting these two limitations are to guard against noise on the high velocity end and to keep control of the air flow on the low velocity end. Velocities of 1,000 fpm or lower never produce any air flow noise; from 1,000 fpm to 1,500 fpm noise might be produced under certain conditions and over 1,500 fpm noise is almost sure to result. Hence, by assuming a maximum velocity of 1,000 fpm we simply are assured that the duct system will not be noisy.

If the duct is to be sized on velocities ranging between 400 fpm and 1,000 fpm, then there must be a velocity increase from the farthest outlet to the fan of

$$1{,}000 \text{ fpm} - 400 \text{ fpm or } 600 \text{ fpm}$$

and, in order to keep this consistent in form and quantity, the

600 fpm increase which is to take place is equally distributed along the main line of the duct; the *main line* of the duct is usually assumed to be the run which has the *greatest number of outlets to feed,* counting from the fan to the last outlet, and is not necessarily the *longest* duct nor the one carrying the *greatest amount* of air.

The number of portions of the duct, in which the quantity of air is different, are then counted up and are usually found to be one less than the number of outlets and branches on the pipe. Thus, a plain duct serving five outlets would have four velocity step-ups to reach the required 1,000 fpm at the fan. After determining the number of velocity step-ups necessary, the 600 fpm and 1,000 fpm, then there must be a velocity increase by the number of step-ups to determine the individual magnitude step-up. In the duct with five outlets and four step-ups the velocity step-up for each portion would amount to

$$\frac{600 \text{ fpm}}{4} \text{ or } 150 \text{ fpm}$$

and this would give the following:

Farthest outlet pipe	400 fpm (assumed for design)
Last two outlets	550 fpm (150 fpm added)
Last three outlets	700 fpm (150 fpm added)
Last four outlets	850 fpm (150 fpm added)
All five outlets	1000 fpm (150 fpm added)

As the fan supplies the pipe feeding all of the outlets and branches, this shows that at the fan the velocity in the pipe would be 1,000 fpm as desired in the design. All of the velocities for the system now have been determined.

224. Areas of the Duct Pipes

If, for example, each of these outlets were to deliver 1,000 cfm the pipe supplying the various outlets would have to supply air in the following amounts:

Farthest outlet	1,000 cfm
Last two outlets	2,000 cfm
Last three outlets	3,000 cfm
Last four outlets	4,000 cfm
All outlets	5,000 cfm

The next step is to set up the quantities of air, divide these quantities by the velocities previously determined for that portion of the duct and obtain the area of the duct in square feet. This is done as follows:

Duct Supplying Outlets	Quantity of Air cfm	Velocity fpm	Area sq ft
Last outlet	1,000	400	2.5
Last 2 outlets	2,000	550	3.6
Last 3 outlets	3,000	700	4.3
Last 4 outlets	4,000	850	4.7
All outlets	5,000	1,000	5.0

An inspection of these results shows that, while only 5 sq ft of duct area would be used to carry the entire 5,000 cfm at the fan outlet, by the time the air has reached the end of the system the pipe sizing has so increased that 2.5 sq ft of duct area is being used to convey 1,000 cfm. Moreover, this increase, or favoring of the far outlets, is accomplished in a consistent and graduated manner as the distance from the supply end of the duct is increased.

225. Friction in Velocity Method of Sizing

In the velocity method of duct sizing no attention is paid to the friction in the duct, as it has been found by experience that with the velocities recommended the friction will be small, probably amounting to ¼ in. to ⅜ in. static pressure for the duct system alone. This usually forms such a small portion of the total static load on the fan that great exactness is not an essential so long as a reasonable and probable amount of static pressure loss is assigned for duct use.

226. Sizing of Grilles and Outlets

If the velocity method of determining duct sizes is followed, it is evident that the velocity in the last pipe (as well as the velocity to each outlet through the branch pipe) will be 400 fpm. At the termination of the pipe a grille, register or open end may be used for the purpose of admitting air from the pipe into the room. The velocity through the grille is very variable, ranging from 50 fpm, in cases where cold drafts are to be feared, up to 1,000 fpm in cases where there are high ceilings with deflectors, such as are encountered in the main ceilings over auditoriums and theaters. The velocity at the grille outlet is one that the designer must settle for himself, taking into consideration the height, size and volume of air to be used and the danger of cold drafts which may be present. Ordinarily, it may be assumed to be about 300 fpm for registers of reasonable height above the breathing line, for preliminary design, at a ten degree diffusion temperature.

In order to determine the area of the grille, register or open end on the duct, using a velocity of 300 fpm, divide the amount of air to be supplied by the velocity and the result will be the area, in square feet, of grille, register or open end required. Thus, one of the outlets in the previous example of duct design to handle 1,000 cfm would, at 300 fpm velocity, require an area of

$$1{,}000 \text{ cfm} / 300 \text{ fpm or } 3\tfrac{1}{3} \text{ sq ft}$$

Some engineers like to make an allowance for the mesh in the register or grille (when one is used), but this is entirely un-

TABLE 116

APPROXIMATE SUPPLY AND RETURN AIR VELOCITIES

Type of Opening	Average Velocity Through Free Area of Grille, Feet Per Minute Heating	Cooling
Supply—Low in wall, discharging horizontally	300–400	50
Supply—In floor, discharging horizontally	300–400	500–800
Supply—High in wall, discharging horizontally	600–1500*	300–1500*
Return	300–500	300–500

* Note: Higher velocities are sometimes permissible with some types of supply grilles, and when a long throw of the air is required.

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necessary as the air flows together within an inch or two of the mesh and attains the same velocity as if the opening were a clear opening.

227. Registers and Grilles

Registers and grilles cap the inlets or outlets which provide air distribution. To obtain proper air distribution, compliance with the following simplified rules, plus a knowledge of the requirements and pecularities of the job in hand will eliminate much trouble. These simple rules may be applied in either cooling or mechanical heating.

1. The air should not enter the breathing level with a velocity in excess of 50 ft per minute. Higher velocities will cause drafts.

2. When two or more grilles are located on one wall, the air

streams should not cross before air has traveled at least two-thirds of its throw.

3. Air should be projected in such a manner as to blanket the room thoroughly, and strike the opposite wall with a velocity not in excess of 50 ft per minute. In unusually large spaces, satisfactory results may be obtained by projecting two or more streams of air toward each other so that they will meet with a velocity not in excess of 25 ft per minute.

4. The largest possible amount of induced air movement should be obtained, so that the incoming air will be thoroughly mixed with the room air, resulting in an even temperature distribution.

The following data are based upon experimental work conducted by a well known grille manufacturer:

a. The throw from a straight flow grille varies with the square root of the daylight area of the grille and with the face velocity within ordinary grille proportions.

b. The aspect ratio of a grille has no appreciable effect on the distance of air throw.

c. Fanning out the air stream shortens the throw, the extent depending on the degree of deflation.

d. For a given throw, the drop of an air stream below room temperature, varies inversely as the face velocity and directly as the temperature differential.

Table 117 may be used as a guide in estimating discharge velocities for bar-type or lattice-type grille without deflectors, and for plain nozzles. This table is approximate, and intended only for use in the general preliminary layout. The performance of numerous special types of grilles may differ appreciably from this table.

228. Sizing Ducts by the Pressure Drop Method

Owing to the fact that the friction of air flowing through a pipe or duct varies with the square of the velocity—that is to

TABLE 117

APPROXIMATE VELOCITIES FOR VARIOUS DIFFUSION TEMPERATURE DIFFERENCES AND LENGTHS OF THROW

	Diffusion Temperature Difference, ° F			
	15	20	25	30
Throw from Point of Supply, Feet	Approximate Velocity Over Face Area, Feet Per Minute			
15	400	525	650	775
25	500	625	750	875
35	600	725	850	975
50	750	875	1000	1125
75	1000	1125	1250	1375
100	1250	1375	1500	1625

Note: Above velocities are above the zone of occupancy.

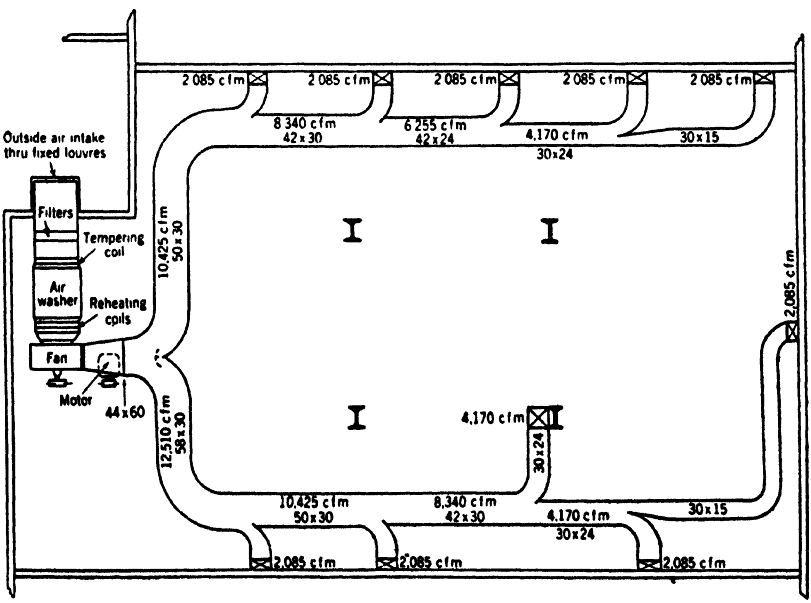


FIGURE 30

Typical Layout of Air Distributing System.

Courtesy of A.S.H.V.E.

say, if the velocity is doubled the friction is increased four times—it is very important in high velocity work to know to what extent the pipe friction has increased. This usually is approximated by determining in advance just what pressure drop is desired and then to size the pipe so that this total pressure drop will occur at the end of the line with proportional pressure drops from the fan towards the far end of the system.

The size of a duct for a specific pressure drop is given in the friction table in Chapter X. After obtaining the approximate size for a round duct, this size is converted to apply to a rectangular duct by referring to dimensions in Tables 104 and 105 in Chapter X. A design loss of 0.06 in. per 100 ft of duct has proved satisfactory for silent operation.

229. Use of Round and Square Ducts

In air-conditioning work for public and private buildings it is customary to use square or rectangular ducts, while in industrial work (where long runs without branches or turns are the rule) the use of round ducts is frequent practice. The square, or rectangular, duct can carry more air through a given space than a round duct and it is easier to construct rectangular branch pieces and elbows than round ones. For long runs it is cheaper to construct round ducts since they require little reinforcing to keep them in shape and the sections may be put together by slipping the end of one inside the end of the adjoining section and holding them in place with a few rivets. When telescoped joints are used, the edge of the interior pipe should *not be turned against the air flow.*

230. Rules for Duct Work

In order to reduce the friction in the flow of air through a duct system certain rules in regard to the construction of the ducts are in use and a duct system which conforms with these rules is termed a well-constructed duct system, while a system which violates any or all of the rules is termed a poorly, or very poorly, constructed duct system. The most important of these rules is that the throat radius of all elbows shall not be less than the diameter, or side, of the duct parallel with the radius. Larger radii do not show any marked reduction in friction, but smaller radii do increase the friction materially. This also applies to all turns where offsets are made and where branches are taken off a main line. See Figures 13 and 13A, Chapter X.

A second rule is that the interior surface of the duct system should be smooth and have no projecting seams or other obstructions to the air flow. A third requirement is, that where branches are connected to a duct, the area of the branch should be proportional to the amount of air expected to flow out of the branch; thus, if a branch were to take off 10% of the total air flowing in the main line at the point where the branch is located, then the area of the main branch times 10% represents the area of the branch where the air is divided. In rectangular ducts this would mean that the distance of the splitter for the branch would be 10% of the width of the duct away from the near side. It will be seen that such an arrangement naturally will tend to divert the proper quantity of air into the branch.

All duct outlets as well as all main branches should be controlled by volume dampers. The purpose of the volume damper is to add a slight resistance to any outlet or branch in which the actual resistance proves to be a trifle less than expected in the design, and thus to equalize the flow so that the air will be delivered in the proper quantities at all of the outlets. Theoretically, it is possible (with a carefully designed duct system based on the pressure drop method) to produce a system which would deliver the desired amount of air at each outlet; practically, this is poor practice, as any additional turns which may prove necessary in order to clear an unexpected obstruction, or for any other cause, will upset the resistance which should be obtained in that duct and will require the addition of volume dampers in all of the other ducts so as to introduce a similar and equal addition of resistance throughout the system. If this is not done, the system will not be balanced and unequal air flow will result. Since volume dampers probably will be required in any event, it is wise to install them in the beginning so that they may be used in adjusting and balancing the system.

231. Balancing up a Duct System

This term means the adjusting of the air flow in a duct system so that the desired amounts of air will be delivered from each and every outlet. It seldom is possible to do this with any great accuracy, and a system 95% correct may be regarded as a well balanced system. The method generally used is to take a preliminary set of readings at the outlets with all volume dampers wide open. These readings will indicate where too much air is being delivered and all outlets with excess air then should have the volume dampers partly closed and a second set of readings made. This will indicate that, as the excess air is cut off from the outlets delivering too much, the outlets which are low will have more air forced into them and will approach their correct amounts. It is also likely that the second set of readings will show that certain outlets which were almost correct on the first reading have had more air forced into them and now are reading slightly high. A third adjustment of dampers on all high outlets, cutting these down to a reasonable and desired amount, should then cause the correct, (or approximately the correct) quantity of air to be delivered from each outlet. As previously stated, it will probably be found that it is impossible to secure exactly the correct quantities from outlets without going over the system repeatedly and even then correctness within 95% is more or less unlikely.

TABLE 118

RECOMMENDED AND MAXIMUM DUCT VELOCITIES

Designation	Recommended Velocities, Feet Per Minute: Residences	Recommended: Schools, Theaters, Public Buildings	Recommended: Industrial Buildings	Maximum Velocities, Feet Per Minute: Residences	Maximum: Schools, Theaters, Public Buildings	Maximum: Industrial Buildings
Outside Air Intakes [a]	700	800	1000	800	900	1200
Filters [a]	250	300	350	300	350	350
Heating Coils [a]	450	500	600	500	600	700
Air Washers	500	500	500	500	500	500
Suction Connections	700	800	1000	900	1000	1400
Fan Outlets	1000–1600	1300–2000	1600–2400	1700	1500–2200	1700–2800
Main Ducts	700–900	1000–1300	1200–1800	800 1000	1100–1400	1300–2000
Branch Ducts	600	600-900	800–1000	700	800–1000	1000–1200
Branch Risers	500	600–700	800	650	800–900	1000

[a] These velocities are for total face area, not the net free area.

Courtesy of A.S.H.V.E.

TABLE 119

WEIGHTS OF SHEET METAL USED FOR DUCT CONSTRUCTION

U. S. Std. Gage	Black Sheets: Approximate Thickness, Inches: Steel	Black: Thickness: Iron	Black: Weight Per Square Foot: Ounces	Black: Weight: Pounds	Galvanized Sheets [b]: Approximate Thickness, Inches: Steel	Galvanized: Thickness: Iron	Galvanized: Weight Per Square Foot: Ounces	Galvanized: Weight: Pounds
30	0.0123	0.0125	8	0.500	0.0163	0.0165	10.5	0.656
28	0.0153	0.0156	10	0.625	0.0193	0.0196	12.5	0.781
26	0.0184	0.0188	12	0.750	0.0224	0.0228	14.5	0.906
24	0.0245	0.0250	16	1.000	0.0285	0.0290	18.5	1.156
22	0.0306	0.0313	20	1.250	0.0346	0.0353	22.5	1.406
20	0.0368	0.0375	24	1.500	0.0408	0.0415	26.5	1.656
18	0.0490	0.0500	32	2.000	0.0530	0.0540	34.5	2.156
16	0.0613	0.0625	40	2.500	0.0653	0.0665	42.5	2.656
14	0.0766	0.0781	50	3.125	0.0806	0.0821	52.5	3.281
12	0.1072	0.1094	70	4.375	0.1112	0.1134	72.5	4.531
11	0.1225	0.1250	80	5.000	0.1265	0.1290	82.5	5.156
10	0.1379	0.1406	90	5.625	0.1419	0.1446	92.5	5.781

[b] Galvanized sheets are gaged before galvanizing and are therefore approximately 0.004 in. thicker.

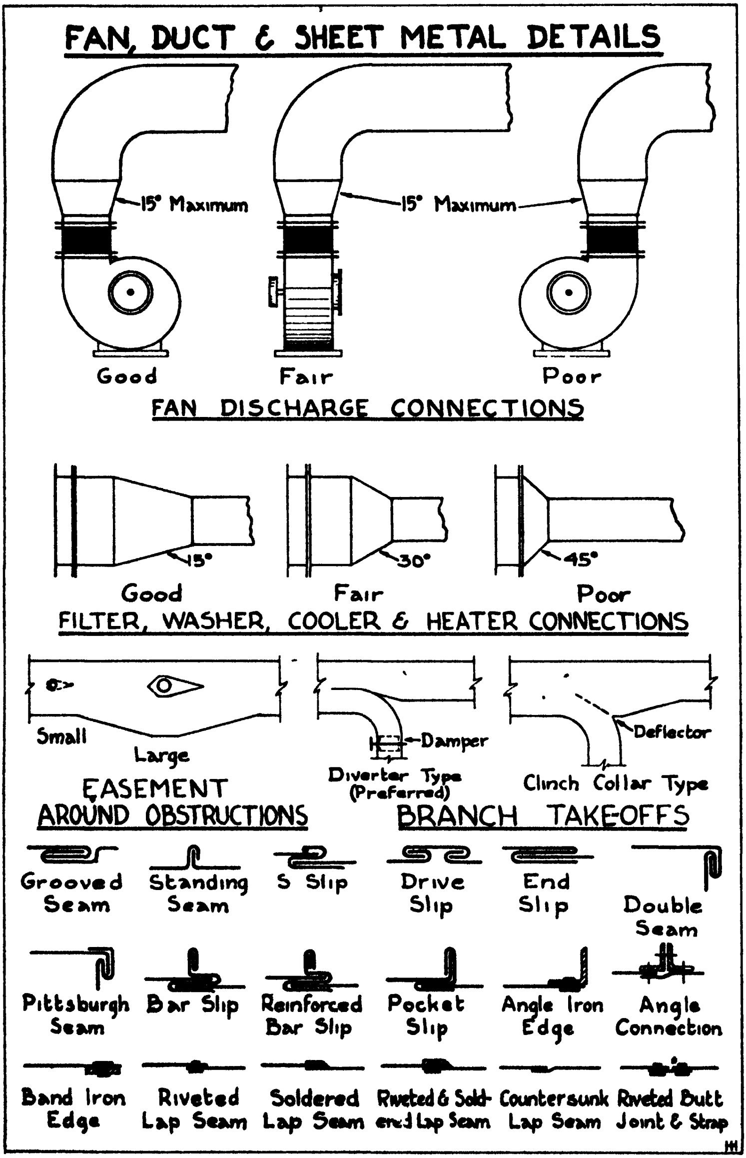

FAN, DUCT & SHEET METAL DETAILS
15° Maximum
15° Maximum
Good
Fair
Poor
FAN DISCHARGE CONNECTIONS
15°
30°
45°
Good
Fair
Poor
FILTER, WASHER, COOLER & HEATER CONNECTIONS
Small
Large
EASEMENT
AROUND OBSTRUCTIONS
Damper
Diverter Type
(Preferred)
Deflector
Clinch Collar Type
BRANCH TAKE-OFFS
Grooved Seam
Standing Seam
S Slip
Drive Slip
End Slip
Double Seam
Pittsburgh Seam
Bar Slip
Reinforced Bar Slip
Pocket Slip
Angle Iron Edge
Angle Connection
Band Iron Edge
Riveted Lap Seam
Soldered Lap Seam
Riveted & Soldered Lap Seam
Countersunk Lap Seam
Riveted Butt Joint & Strap
HH

FIGURE 31

TABLE 120

WEIGHTS AND THICKNESSES OF STANDARD COPPER SHEETS[c]

Rolled to Weight

Weight per Square Foot		Thickness, Inches		Nearest Gage Number		
Ounces	Pounds	Decimal Equivalent	Nearest Fraction	B. & S.	Stubs	U. S. Standard
10	0.625	0.0135	1/64	27	29	29
12	0.750	0.0162	1/64	26	27	28
14	0.875	0.0189	1/64	25	26	26
16	1.000	0.0216	1/32	23	24	25
18	1.125	0.0243	1/32	22	23	24
20	1.250	0.0270	1/32	21	22	23
24	1.500	0.0324	1/32	20	21	22
28	1.750	0.0378	1/32	19	20	20
32	2.000	0.0432	3/64	17	19	19
36	2.250	0.0486	3/64	16	18	18
40	2.500	0.0540	3/64	15	17	17
44	2.750	0.0594	1/16	15	17	17
48	3.000	0.0648	1/16	14	16	16
56	3.500	0.0756	5/64	13	15	14
64	4.000	0.0864	5/64	11	14	13

[c] Variations from these weights must be expected in practice.

Courtesy of A.S.H.V.E.

232. Fans

In air-conditioning practice, fans are required to produce air flow, except where positive displacement is required, in which case air compressors or rotary blowers are used. The type and style used depends upon noise limitations, the volume of air required and the static pressure required for the given application.

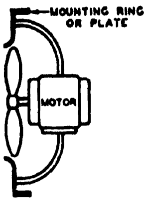
Fans are classified according to the direction of air flow as (1) *axial flow* or *propeller* type if the flow is parallel with the axis, and (2) *radial flow* or *centrifugal* type if the flow is parallel with the radius of rotation.

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.

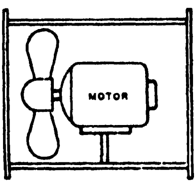
A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements 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.

Figure 32 shows the names and definitions of types of fans recently distributed by the *National Association of Fan Manufacturers.*



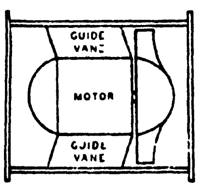
Propeller Fan

A propeller fan consists of a propeller or disc type wheel within a mounting ring or plate and including driving mechanism supports either for belt drive or direct connection.



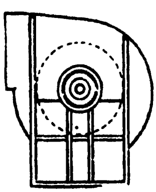
Tubeaxial Fan

A tubeaxial fan consists of a propeller or disc type wheel within a cylinder and including driving mechanism supports either for belt drive or direct connection.



Vaneaxial Fan

A vaneaxial fan consists of a disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports either for belt drive or direct connection.



Centrifugal Fan

A centrifugal fan consists of a fan rotor or wheel within a scroll type of housing and including driving mechanism supports either for belt drive or direct connection.

FIGURE 32

Names and Definitions of Types of Fans.

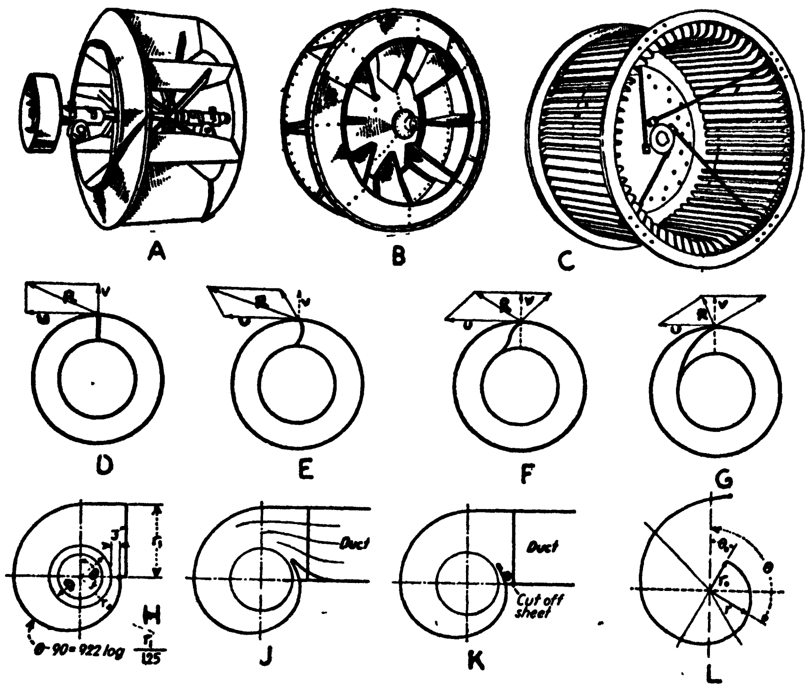


FIGURE 33

Fans

A. Pressure blower
B. Conical plate centrifugal fan
C. Squirrel-cage type centrifugal fan
D. Velocity diagram of straight blade tip where U = tip speed; V = radial velocity of air; R = true air velocity relative to fan casing
E. Velocity diagram of forward curved blade
F. Velocity diagram of semi-backward curved blade
G. Velocity diagram of full backward curved blade
H., J., K. Scroll designs of various fan casings
L. Typical logarithmic spiral

233. Centrifugal Fans

Centrifugal fans were first produced in the steel plate or "paddle wheel" type and are still used today in some instances for forced draft as this type of fan possesses certain desirable features in connection with forced draft use even if its features for ventilation work have been surpassed by improved designs. Later, the multi-blade type of wheel was developed and gave more or less satisfactory service in ventilating work for many years. Following this, the non-overloading type of wheel was designed and today this is the most satisfactory fan on the market for ventilating and air-conditioning work.

Centrifugal fans may have blades curved forward or curved backward in the direction of air flow. Centrifugal fans with forward curved blades differ in performance from those with backward curved blades. For a given air volume, forward curved blades must be run at lower speed or have smaller diameters. If similar fans are run at the same speed, the output of the forward-curved one will be considerably greater than

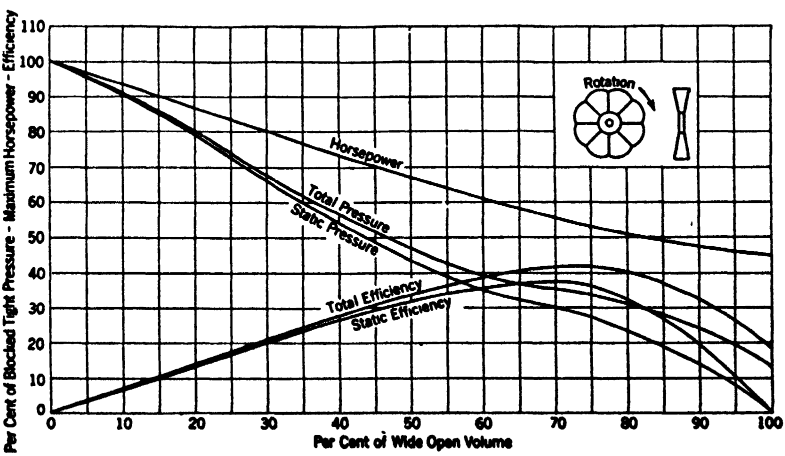


FIGURE 34
Operating Characteristics of an Axial Flow Fan
Courtesy of A.S.H.V.E.

that of the fan with the backward-curved blade. There is also a difference in typical characteristics as shown in Figures 34 to 36.

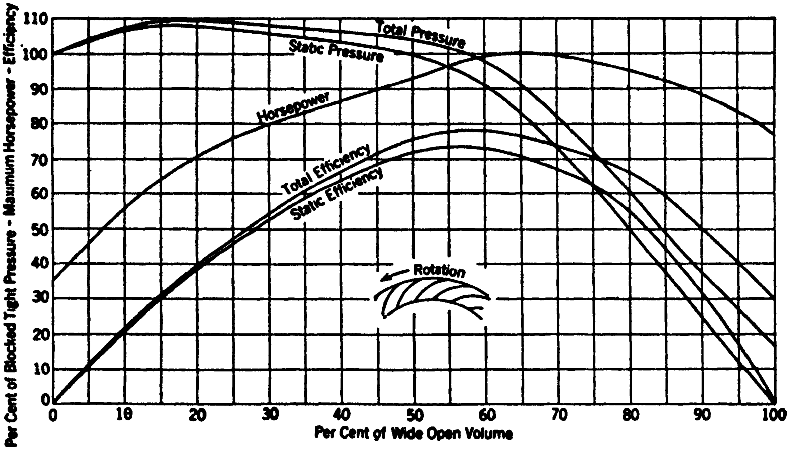


FIGURE 35

Operating Characteristics of a Fan with Blades Curved Backward

Courtesy of A.S.H.V.E.

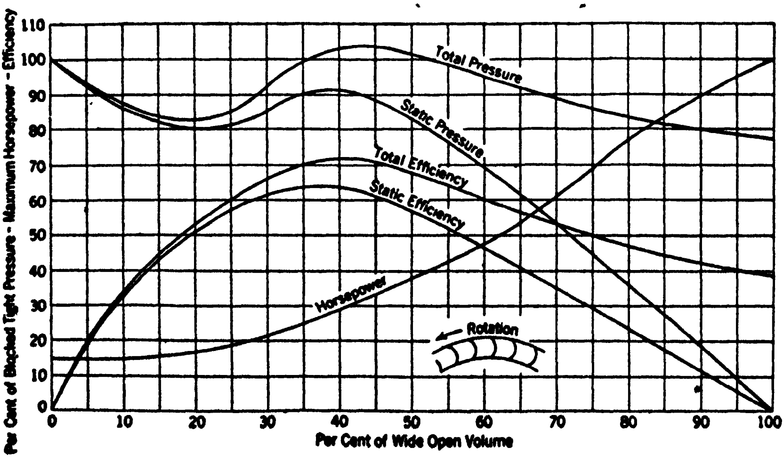


FIGURE 36

Operating Characteristics of a Fan with Blades Curved Forward

Courtesy of A.S.H.V.E.

Fan characteristic curves, representative of propeller fans in general, are shown in Figure 34. The curves show that the capacity of the fan drops quickly with an increase of static pressure, and that the horsepower increases because of the reduced efficiency with increased pressure. Hence, propeller fans are limited, for the most part, to the movement of relatively large volumes of air at comparatively low static pressures.

Characteristic curves for a forward curved centrifugal fan are shown in Figure 36. This type of fan is widely used, has a large capacity for the space occupied, and is quiet in operation. The point of maximum efficiency is very close to the point of maximum static pressure. Forward curved fans should be operated close to this point. At lower pressures operation may be accompanied by objectionable noise. Beyond the point of maximum efficiency (to the left of the peak of the curve of static efficiency), forward curved fans may be unstable in operation and subject to wide variations in air flow with comparatively small changes in static pressure. This may be accompanied by excessive noise caused by fluttering, or sudden changes in air flow. The power curve rises continuously from a minimum at shut-off to a maximum at free delivery, showing that from the standpoint of the fan alone the driving motor might be subject to overloading or underloading in case it was subject to fluctuating static pressure.

Figure 35 shows characteristic curves for a backward curved centrifugal fan. Its usual high speed operation often makes desirable direct connected motor drive, even with large size fans, and its non-overloading power curve has advantages. The steep pressure curves to the right of the point of maximum efficiency give a relatively flat regulation curve, so that considerable changes in static pressure do not greatly affect the air flow. The maximum power requirement generally occurs at the point of maximum efficiency. If a motor is selected for this point, the possibility of its being overloaded is eliminated, even though the static pressure of the system is considerably smaller than expected.

There are other variations in centrifugal fans between the full forward and full backward curved types, but they all have characteristics varying between those of the fans mentioned above.

234. Multi-blade and Non-overloading Fans

A study of the characteristics of the multi-blade fan as compared to the non-overloading type shows that the greatest efficiency in the multi-blade fan usually occurs at about 40% of its free air delivery, but the greatest horsepower is required, not at this 40% free air delivery, but at 100% of free air delivery, or when the fan is discharging with an open end. Inspection of the non-overloading type of fan shows that its point of greatest efficiency occurs at about 65% of free air delivery and that the greatest horsepower requirement to drive the fan occurs at 65% of the free air delivery and not at 100% as was the previous case. Because of these characteristics, it is necessary to install a multi-blade fan with a capacity close to 40% of its maximum delivery, while a non-overloading fan may be used at 65% of its maximum delivery. It is desirable, of course, to use a fan at or near its point of maximum efficency because, at this point, the fan delivers more air with less power consumption than under any other conditions.

The most important consideration, however, is the fact that when the multi-blade fan is used the motor which drives it should have from 1.5 to 2 times the horsepower actually required when the fan is operating under the ordinary duct load so as to protect the motor from burning out, should a door or bypass be left open. The static pressure on the fan is reduced and the air quantity increased, resulting in a much higher motor load. This results in the motor (under ordinary operating conditions) running with only about 50% of full load with the drop in motor efficiency which usually occurs under half instead of full load.

With the non-overloading fan, no condition will increase the load on the motor and so the motor which drives it may have only the horsepower which the fan requires as a maximum, and therefore, this will be a relatively smaller motor than that for a multi-blade fan with a corresponding increase in the motor efficiency. The reason that this type of fan is called a *non-overloading* type is that it will not overload the motor even if doors are left open.

235. Brake Horsepower

A knowledge of the terminology of fans is necessary if one is to understand their operation. One of the most common terms in this field is *horsepower,* i.e., the power required to operate the fan. One horsepower is equivalent to 33,000 foot pounds.

The term is used loosely to describe different kinds of power, e.g., mechanical horsepower, electrical horsepower and brake horsepower.

Brake horsepower is actual mechanical horsepower as developed at the pulley or coupling of a machine and measured by a brake. It represents in a *power-developing* machine the actual power produced after all losses have been subtracted, which is another way of saying available power; in a *power-using* machine it means the actual power which must be applied to the machine to drive the machine and includes all losses in the machine.

Thus, a generator producing 10 horsepower with 90% generator over-all efficiency would require 11 horsepower (brake) to drive it but would produce only 10 horsepower (brake) at the pulley; the other horsepower unit represents the losses in the generator. A fan requiring 10 horsepower (brake) to operate and operating at 70% efficiency would actually deliver to the air only 7 horsepower as the other 3 horsepower are lost in the fan, it may be seen that the brake horsepower for a fan includes all of the fan losses.

236. Static, Velocity and Total Pressure

In moving air a certain quantity of power is required to set the air in motion at its maximum velocity and the pressure necessary to do this is called the *velocity pressure.* The power required to raise the pressure of the air as well as to give it velocity is termed *static pressure*, and the sum of these two pressures represents the total work on the air and is called the *total pressure.* Air pressures in fan and duct work are very low, and they are customarily measured in inches of water; 1 in. static pressure means a pressure sufficient to hold up or maintain a column of water 1 in. high.

237. Calculating the Power for Fans

The horsepower for fans may be calculated by computing the *theoretical horsepower* or the horsepower assuming the fan 100% mechanically efficient; the actual horsepower, however, will be the theoretical horsepower divided by the fan efficiency as some power is always lost in the fan itself. The formula for theoretical horsepower is:

$$\frac{cfm \times tp}{6356} = \text{theoretical hp}$$

when

cfm is the number of cubic feet of air handled per minute
tp is the total pressure of the air
6356 is a constant

The static pressure for a fan usually is known and in such cases, if the velocity pressure is not known, it may be obtained by the formula given below:

$$\left(\frac{\text{Fan Outlet Velocity}}{4{,}000}\right)^2 \quad \text{equals the velocity pressure}$$

The theoretical power computed on the basis of a fan efficiency of 100% then may be changed to brake horsepower by dividing by the actual fan efficiency; or

$$\frac{\text{theoretical hp}}{\text{efficiency}} \text{ equals brake hp}$$

238. Selecting Fans for a Given Installation

A single width and single inlet (s.w. –s.i.) fan is a fan of the ordinary width and construction with one inlet on one side and the drive pulley on the other side. In cases where a larger fan is required, a double width and double inlet (d.w. –d.i.) fan may be used, such a fan being really two single width fans set together with the center partition removed. The double width fan *must have two inlets* in order to supply sufficient air to the suction of the fan, but single width fans customarily have but one inlet. Single width fans may be obtained with two inlets if necessary, but are unusual in practice. The capacity of a double width fan is twice that of the same size fan in single width, and a single width fan with two inlets has a capacity about 40% greater than that of the ordinary single width fan with one inlet. It is very easy to make a tight duct connection to a single width fan with one inlet as the pulley comes on the opposite side but with double inlet fans it usually is best to set them in a chamber from which they take their suction, because of the difficulty of getting a tight connection to the fan suction on the pulley side.

In selecting a fan for an air-conditioning system great care must be exercised to obtain a fan with sufficient capacity, adequate static pressure, practically noiseless operation and minimum horsepower requirements.

All fan manufacturers' catalogs give air quantities for each size of fan when operating against different static pressures and such deliveries require certain speeds (rpm) and different outlet velocities. As a usual thing, the capacity required in the system is increased 10% in picking out the fan to allow for duct leakage and to be sure that sufficient air is going to be supplied. The static pressure may be calculated from the duct system designed by the pressure-drop method or may be assumed on the basis of experience in the instance of a duct system designed by the velocity method. It has been found, for instance, that in small duct systems, e.g., up to 100 ft maximum run length, the static pressure for all losses outside of the grilles, louvers, dampers, heaters, cooling coils, air washer and filters may be assumed to be approximately ⅜ in. The losses for the heaters, cooling coils, air washer, filters, etc., must be added to this to secure the total static pressure for the fan and these losses for different conditions of service are given in the manufacturers' catalogs. Usually they approximate the following:

Air Filters	0.20 to 0.25 in.
Air Washer	0.25 to 0.33 in.
Air Heaters	0.25 in.
Cooling Coils	0.25 to 0.40 in.

All losses on moderate sized duct systems, e.g., 100 to 300 ft long, outside of the equipment, etc., may be assumed to approximate ½ in. and for long duct systems, not over 600 ft, ⅝ to ¾ in. After computing all static pressure losses 0.10 in. should be added for safety.

Thus a moderate-sized duct system using an air washer and air heaters should have a fan developing a static pressure of

Duct System	0.50
Air Washer	0.28
Air Heaters	0.25
Contingencies	0.10
Total static pressure	1.13 in. or 1⅛ in.

After determining the volume of air for the fan rating and the static pressure against which the fan must operate, a fan may be picked out of the catalog and tested for tip speed and outlet velocity. If the tip speed is too high the blades will *chop* the air in revolving and air noise will be produced in the fan. If the outlet velocity is too high, air noise is likely to be produced at the fan outlet. All fans make *some* noise, but they should not make an unusual amount of noise or noise so great as to annoy the occupants of a building. While there is some variation between different makes of the non-overloading fans the following table gives approximate outlet velocities and tip speeds of the wheels for various static pressures which are recommended for quiet operation.

TABLE 121

AIR VELOCITIES AND TIP SPEEDS USED WITH MULTIBLADE FANS

	Forward Curved Fans		Backward Curved Fans	
Pressure (Static) Inches H_2O	Outlet Velocity, Feet Per Minute	Tip Speed, Feet Per Minute	Outlet Velocity, Feet Per Minute	Tip Speed, Feet Per Minute
0.250	1000–1100	1500–1700	1000–1100	3300–3700
0.375	1000–1100	1700–1900	1000–1100	3700–4100
0.500	1000–1200	1900–2100	1100–1300	4100–4500
0.625	1100–1300	2100–2400	1100–1300	4500–4700
0.750	1200–1400	2400–2700	1100–1300	4700–4900
0.875	1300–1600	2700–2900	1100–1200	4700–4900
1.00	1400–1800	2900–3100	1000–1100	4900–5100
1.25	1700–1900	3100–3400	1000–1100	5000–5200
1.50	1800–2100	3400–3800	1000–1100	5200–5600
1.75	1900–2300	3800–4200	900–1000	5600–6000
2.00	2200–2600	4200–4500		
2.25	2400–2800	4500–4700		
2.50	2600–3000	4700–4900		

The following precautions should be observed when applying fans to air-conditioning systems. The fan should be selected to deliver the required cfm at a slightly greater static pressure than the calculated value of the system, in order to allow for small errors in calculation and to assure the delivery of the required amount of air.

A fan picked out on the single width and single inlet basis will have a given capacity; a double width and double inlet fan will deliver twice as much but with double the horsepower required to drive it. The brake power quoted in the different catalogs will indicate which fan delivers the air with the lowest power requirement. This, of course, is the most economical fan to operate as it will require least current for the motor.

239. Fan Foundations

While the fan selected may be (and should be) so designed as to avoid the possibility of air noise and the blades *chopping* the air, it is also necessary to be sure that the setting is such that no *mechanical noise* will be produced. The fan should be perfectly balanced both statically and dynamically so that it will not produce a vibration of the machine when in operation; what vibration is unavoidable must be absorbed in a foundation of suitable weight insulated from the building construction by a non-transmitting material so that the vibration will not shake the floor or vibrate the building or portion of the building in which the fan is housed.

As a general thing the foundation weight should be at least three times the weight of the fan; this usually means 12 in. of concrete under large fans and perhaps 6 in. of concrete under smaller fans. The fan is attached to this concrete foundation by lag screws extending into a 4 in. x 6 in. wood frame and the wood frame in turn is attached to the concrete by foundation bolts having countersunk heads in the wood frame. To separate the foundation from the building construction 4 in. of cork is laid below the slab of concrete and 2 in. of cork all around the four sides, finishing off with plastic at the top and with the floor concrete brought up against the cork so as to form a 4 in. thick curb at least 6 in. high. The fan and motor should be mounted on a single foundation so as to keep the two in line and the motor should have slide rails to permit tightening of the belt when it has stretched.

240. Canvas Connection on the Fan

Still another item in noise prevention is the matter of using canvas connections on all duct connections to the fan to prevent the telephoning of noise from the fan through the sheet metal duct. Such connections are about 6 in. long and are bolted at each end to galvanized strips which are attached to the fan outlet and to the termination of the duct. Canvas should be applied to the discharge and the suction openings of the fan when tight duct connections are used.

241. Motor Speeds

The fan motor speed should bear some relation to the fan speed and should not be more than four times that of the fan, if possible. Motor speeds are influenced by the current characteristics, but if the motor speed becomes too high in relation to the fan speed, an unusually large pulley will be required on the fan together with longer belts and other undesirable features.

242. Fan Formulas

The volume of air delivered by a fan varies as speed.

The pressure of air varies as square of speed.

The power consumed varies as cube of speed.

The approximate total head, or dynamic pressure produced by a squirrel cage fan is expressed as

$$V = 2{,}640\sqrt{H}, \text{ inches water gage.} \qquad (1)$$

or

$$H = \frac{V^2}{7{,}000{,}000}$$

V = tip velocity, feet per minute.
H = dynamic head, inches.

Variations in design cause coefficient changes of 20%. For medium velocities, in American practice, designs generally allow static pressure = 0.85 dynamic pressure and velocity pressure equal to 0.15 dynamic pressure.

$$\text{Air Horsepower} = \frac{cfm \times \text{total pressure in inches of water}}{6356} \quad (2)$$

$$\text{Static efficiency} = \frac{cfm \times \text{static pressure in inches of water}}{6356 \times \text{Horsepower input}} \quad (3)$$

Mechanical or Total efficiency

$$= \frac{cfm \times \text{total pressure in inches of water}}{6356 \times \text{Horsepower input}} \quad (4)$$

For known conditions let

V = cubic feet air per minute
SP = static pressure
hp = power
D = diameter of fan wheel
rpm = revolutions per minute

For unknown conditions let

V_x = cubic feet air per minute
SP_x = static pressure
hp_x = horse power
D_x = diameter of fan wheel
rpm_x = revolutions pèr minute

then for centrifugal fans of different sizes but of same proportions,

$$\frac{V_x}{V} = \frac{rpm_x}{rpm} \quad (5)$$

$$\frac{SP_x}{SP} = \left(\frac{rpm_x}{rpm}\right)^2 \quad (6)$$

$$\frac{hp_x}{hp} = \left(\frac{rpm_x}{rpm}\right)^3 \quad (7)$$

$$\frac{V_x}{V} = \left(\frac{D_x}{D}\right)^3 \quad (8)$$

$$\frac{SP_x}{SP} = \left(\frac{D_x}{D}\right)^2 \quad (9)$$

$$\frac{hp_x}{hp} = \left(\frac{D_x}{D}\right)^5 \quad (10)$$

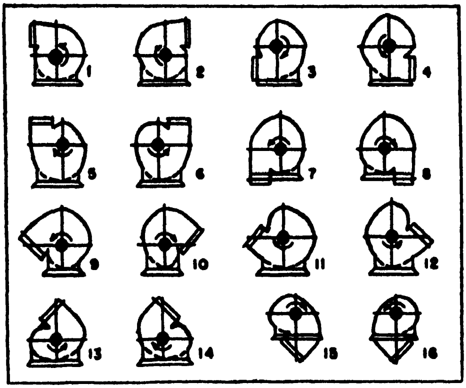


FIGURE 37

Standard Designation of Fans

1. Counter-Clockwise Top Horizontal. 2. Clockwise Top Horizontal. 3. Clockwise Bottom Horizontal. 4. Counter-Clockwise Bottom Horizontal.

5. Clockwise Up Blast. 6. Counter-Clockwise Up Blast. 7. Counter-Clockwise Down Blast. 8. Clockwise Down Blast.

9. Counter-Clockwise Top Angular Down. 10. Clockwise Top Angular Down. 11. Clockwise Bottom Angular Up. 12. Counter-Clockwise Bottom Angular Up.

13. Counter-Clockwise Top Angular Up. 14. Clockwise Top Angular Up. 15. Clockwise Bottom Angular Down. 16. Counter-Clockwise Bottom Angular Down.

CHAPTER XIII

HEATING LOAD

Human progress is very definitely related to the discovery of fire and the evolution of heating. Fire cannot burn without fuel, and Bulletin 139, United States National Museum, says: "There is more historical data on mineral lamp fluids than on any other class. Native naphtha was used from an early period in Persia. . . . In Japan and Burma outcrops of mineral oil were put to some use for burning."

In the Bible, Job, speaking of his lost blessings, mentions "rock that poured me out rivers of oil." We also find that ten thousand years ago Noah caulked the seams of the ark with pitch taken from native outcroppings of crude petroleum.

It is now known that the fire worshippers in 600 B.C. used petroleum in their rites, as we know that the followers of Zoroaster made trips to the eternal fires of Surakhani on the Apsheron Peninsula (in the Baku district on the Caspian Sea). From our knowledge of this region today, we can safely assume these fires were fed by crude oil seepages. Plutarch, in his *Life of Alexander,* describes how, in the district of Ecbatana (Persia) Alexander was particularly struck with "a gulf of fire, which streamed continually as from an inexhaustible source."

Marco Polo, at the end of the 13th Century, says of the Baku petroleum: "On the confines toward Georgine there is a fountain from which oil springs in great abundance, inasmuch as a hundred shiploads may be taken at one time. This oil is not good to use with food but is good to burn, and is also used to anoint camels that have the mange." Early records indicate that petroleum was first used in lamps. The Vestal Virgins were modern for their time. We are told they filled their lamps with petroleum and that the wicks were of a mineral substance, probably asbestos. Russia, where oil seepages were plentiful, was the first to make use of petroleum for heating and to refine it in a crude way. When the heavier refuse oil became a real problem, Werner, a mechanic, conceived the idea of burning it. His burner, consisting of a series of griddles over which the oil trickled and burned, was patented in 1861.

The ancient armorer of the sword used charcoal in the forge and when coal was found in the vicinity, its advantages were soon utilized. If the artisan of old could have written we would possibly have earlier records of the use of other fuels. The priests monopolized the art of writing and from their records we learn that oil was their favorite source of heat.

243. Maximum Heating Demand

The maximum probable heat demand must be accurately estimated for the design of a heating system, in order that the apparatus installed should be capable of attaining the temperatures required. The factors which determine the maximum heat demand, many of which are never in equilibrium, include the following:

1 Inside temperatures. 2 Stratification of air. 3 Type of heating system. 4 Ventilation requirements. 5 Period and nature of occupancy. 6 Temperature regulation.	*Inside Conditions*
7 Heat transmission of exposed parts of building. 8 Infiltration of air through cracks, crevices and open doors and windows. 9 Heat capacity of materials. 10 Rate of absorption of solar radiation by exposed materials.	*Building Construction*
11 Outside temperature. 12 Rain or snow. 13 Sunshine or cloudiness. 14 Wind velocity.	*Outside Conditions (The Weather)*

244. Calculating Heat Losses

The quantity of heat passing through the walls and glass (either into a room when the room is cooler than the outside temperature, or out of a room when the room is warmer than the outside temperature) must be calculated in order to cool or warm the room, as the case may be. This is because, in cooling a room, the heat must be removed as fast as it enters or the temperature in the room will rise, while in warming a room, the heat must be supplied as fast as the heat is lost or the room will grow cold. In other words, a *heat balance* must be maintained.

It is a law in connection with the flow of heat that *heat always flows from a region of higher temperature to one of lower temperature,* and that *the rate of flow is proportional to the temperature difference* within reasonable limits. That is to say, if a certain amount of heat flows from one side of a wall to the other when there is a difference of one degree between the two air temperatures, a difference of 10 degrees between the two air temperatures will cause the flow to be 10 times as great.

245. Conduction and Resistance to the Flow of Heat

All materials will allow a heat flow to some extent and to this extent conduct heat; similarly all materials resist the flow of heat to a greater or less extent and have a resistance to this flow. Materials of high resistance to heat flow are termed *insulation,* but there is no dividing line between insulation and materials which are not insulation—the division is relative rather than absolute. Metals are usually very good conductors of heat; so good, in fact, that it never has been possible to establish the difference in efficiency between a boiler having ¼ in. thick plate and a boiler having ½ in. plate. The quantity of heat which a material will transmit per square foot of area for one degree difference in temperature and for 1 in. in thickness is termed the *conductivity* and the amount of heat which will be transmitted through several materials of varying *conductivities* is termed the *conductance,* both of these being for the material itself and exclusive of the "surface" conductivity.

On the other hand, the *resistivity* or resistance of a material to heat flow is the reciprocal of the conductivity or 1/conductivity while the *resistance* of a combination of materials is the sum of the resistivities of the various parts. Surfaces resist the flow of heat into a wall and out of a wall; this is termed *surface resistance,* and is greater where the air in contact with the wall is still air than where the air in contact with the wall is moving air, e.g., when a wind is blowing on a wall. An air space will transmit a certain amount of heat and also resist the flow of heat, the same as a building material, this transmission for spaces of 1 to 6 in. usually being taken as 1.10 Btu per square foot per degree temperature difference per hour.

The following brief list gives conductivities for various materials so that if the thicknesses are known the heat transmission through any thickness of wall and made up of any materials may be calculated. See Tables 46, 47 and 48, Chapter V.

Material	*Conductivity*	*Thickness*
Brick	5.00	1.00 in.
Inside surface	1.65	still air
Outside surface	6.00	15 m.p.h.
Stone	10.00	1
Lath and plaster	2.32	as constructed

246. Calculating the Coefficient of Heat Transmission

The *coefficient of heat transmission* for a given wall may be defined as the quantity of heat, in Btu per hour, which will be transmitted through one square foot of wall area for a temperature difference of one degree between the air on the two sides of the wall. The *factor* for the wall is the coefficient multiplied by the actual temperature difference and is therefore the *quantity of heat in Btu per hour transmitted through the wall for a given temperature difference.* In order to obtain the factor it is necessary, of course, to first calculate the coefficient and then multiply the coefficient by the temperature difference to be maintained.

The coefficient for the heat transmission through any wall is the reciprocal of the sum of all the resistances. Thus a wall with a total resistance of 25 would have a coefficient of 1/25 or 0.04, while a wall with a total resistance of 10 would have a coefficient of 1/10 or 0.10. It also should be borne in mind that where the conductivity is given for 1 in. of thickness, the resistivity will be 1/conductivity for 1 in. of thickness, but for 2 in. of thickness it will be 1/conductivity x 2, and for 4 in. of thickness will be 1/conductivity x 4, and so on.

Suppose, for example, that it is desired to calculate the factor for a 70 degree difference on a wall made up of 4 in. stone facing, backed up with 8 in. of brick, furred with metal lath and plaster on the inside. From the previous table it is found that the conductivity for the various materials is as follows:

Stone	10.00 per in. of thickness
Brick	5.00 per in. of thickness
L. & P.	2.32 as constructed

and the resistivities will be:

Material	*For 1 In.*	*Thickness Used*	*Total Resistivity*
Stone	1/10.00	4 in.	4/10.00
Brick	1/5.00	8 in.	8/5.00
L. & P.	1/2.32	As constructed	1/2.32

But there is also an inside and outside surface which will amount to an added resistance of:

Outside Surface	1/6.00	As constructed	1/6.00
Inside Surface	1/1.65	As constructed	1/1.65

In order to add these resistances together it will best to convert them to decimals giving the following:

Stone	4/10.00	or	0.40
Brick	8/5.00	or	1.60
L. & P.	1/2.32	or	0.431
Outside Surface	1/6.00	or	0.167
Inside Surface	1/1.65	or	0.606
Total Resistance			3.204

and the coefficient will be 1/3.204 or 0.312, while the factor will be 0.312 x 70 degrees or 21.84 which would be taken as 22 Btu per hour per square foot of wall area.

247. Heat Factors for Walls and Glass

It is not necessary, however, to enter into such calculations of the coefficient except in cases of special walls where the coefficient cannot be found in tables for standard constructions which already have been worked out by a similar method. Such tables are found in more or less variety in many handbooks which should make unnecessary the calculation of the coefficient in some cases. See Tables 46, 47, 48, Chapter V.

Glass windows are usually assumed to have a coefficient of 1.13 Btu per square foot per degree difference per hour, and outside doors are assumed the same as glass as far as heat losses or gains are concerned. Double-glass coefficient is 0.45 and that of glass brick 6 in. thick, 0.60.

248. Exposure and Air Change

All coefficients are for south walls and certain percentages are added for walls facing east, west and north to allow for lack of sun, colder winds, and so on. Usually these percentages are for well protected city buildings and approximate 15% on the north and 10% on the east and west; if the building is not protected there should be an increase of 10% on each side including the south; if in a badly exposed position, such as on the top of a bluff, the increase should be 20% on each side, including the south.

Some leakage of outside air or *infiltration* occurs around window sash and to some extent, through the walls as well. Usually this is approximated as 1 air change for one wall exposed, 1.5 air changes for two walls exposed, and 2 air changes for three walls exposed. An exposed wall is a wall with the outside weather on one side. An air change is the air content, in cubic feet, of the room; thus one air change means that the quantity of air leaking into the room will equal the entire air content of the room during the period of one hour. The heat required to warm this outside air will be 1/55 of a Btu for every cubic foot of air so entering for one degree temperature difference between the room air temperature and the outside temperature. Thus a room containing 1,000 cu ft, and having one wall exposed would have one air change, or 1,000 cu ft of outside air leaking into the room per hour. If it is 0° F outside and the room is at 70° F, the air must be warmed to 70° F after it enters the room and this will require

$$1{,}000 \text{ cu ft} \times 1 \text{ A.C.} \times 1/55 \text{ Btu} \times 70 \text{ or } 1{,}272 \text{ Btu}$$

249. Procedure in Making a Heat Calculation

To make a heat calculation compute all of the wall area in each exposed wall, subtract the glass and door area to obtain the net wall area; then multiply the net wall area by the factor for the temperature difference desired and the glass and door area by the factor for glass, add the two together and add the exposure allowance if it is not a south wall. Repeat this with each exposed wall if there is more than one wall so exposed. If there is a cold ceiling and/or floor compute the area of each and multiply by the proper factor assuming *half the temperature difference* in each case. This is because the cold attic above or the cold cellar beneath will assume a temperature which usually is about halfway between the inside temperature and the outside temperature. If there are any cold rooms adjoining the room to be heated, calculate the area of cold partition resulting and multiply this by one-half the temperature difference between the inside and outside in the same manner as the floor and ceiling were treated. There is no exposure to be added on floors, ceilings and inside partitions. Then compute the infiltration based on the proper number of air changes as previously noted and determine the Btu required for this. Add all of the Btu together and the total will be the total Btu to be supplied to the room every hour in order to keep the room at the number of degrees above the outside temperature which was assumed in the determination of the factors.

250. Example in Calculation of Heat Losses

A room is 10 ft x 10 ft x 10 ft. It has a west wall exposed, one window 5 ft x 4 ft in the wall; no door, and a warm ceiling and floor. The room is to be maintained at 70° F when the outside temperature drops to 0° F. The wall is 8 in. hollow tile with stucco and plaster inside. What will be the hourly heat loss in Btu?

Gross Wall 10 ft x 10 ft or	100 sq ft	
Glass 5 ft x 4 ft or	20 sq ft x 80 Btu	1600 Btu
Net Wall	80 sq ft x 21 Btu	1680 Btu
Total without exposure		3280 Btu
Exposure for west side 10 per cent		328 Btu
Total for outside wall		3608 Btu
Ceiling, warm		None
Floor, warm		None
Infiltration, 1000 cu ft x 1 A.C. x 70 degrees x 1/55		1270 Btu
Total loss		4878 Btu

In calculating heat losses the use of a standard form is recommended. See Figure 51, at end of this chapter.

251. Auxiliary Heat Sources *

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly

* *Courtesy of A.S.H.V.E.*

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.

252. Condensation on Interior Walls *

Condensation in the interior surfaces [1] of buildings may cause irreparable damage to manufactured articles and machinery. It often results in short-circuiting of electric power, and causes disintegration of roof structures not properly protected.

The prevalence of moisture on a surface is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature. It can be eliminated by (1) raising the surface temperature with increased air velocities passing over the surface, or adding a sufficient thickness of insulation, and (2) by lowering the humidity; this is often not possible due to manufacturing processes.

The condensation of moisture within walls [2] is an important problem with many types of construction under adverse conditions. The temperatures of the various parts of a wall are

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* *Courtesy of A.S.H.V.E.*

[1] Preventing Condensation on Interior Building Surfaces, by Paul D. Close (A.S.H.V.E. Transactions, Vol. 36, 1930, p. 153).

[2] Condensation within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1938).

controlled by the type and amount of insulation used and the vapor densities in the corresponding sections are controlled by the type of vapor barriers installed. The transmission of heat and vapor through a wall should be considered together, and in most cases the proper combination of insulation and vapor barriers will eliminate the possibilities of condensation within walls. A consideration often overlooked in problems of condensation within walls is that a vapor barrier should be placed on the warm side and not on the cold side of a wall.

253. Rapid Estimating of Heating Problems

It is recommended that approximate and condensed estimates of heating capacities be made before calculating accurately, as previously described. A rough idea of the quantity of radiation or heat required enables the engineer to form a judgment as to size and cost of the job and offers a rough check on the calculations. Not only should a rough calculation be made before the job is estimated, but also a condensed estimate after the calculations are made. The following data, if used with discretion, will prove valuable in making condensed estimates of heating requirements.

254. Simple Heating Rules

The following rules for rapid estimating have been used by heating contractors for many years. They were created at a time when most residential walls were not insulated, and so may be considered to give results on the conservative side.

In the following formulas:

C = content of room in cubic feet;

G = exposed glass and door area in square feet;

P = perimeter of windows and exposed doors in feet;

W = net exposed wall surface in square feet, plus inside surfaces adjacent to unheated spaces as attics, basement, etc., as follows: ½ ceiling and floor area in square feet, ¼ wall area in square feet;

n = number of air changes per hour:

for halls	$n = 3$
1st floor rooms	$n = 2$
2nd floor rooms	$n = 1$
1st floor offices and stores	$n = 2$ to 3
2nd floor offices and stores	$n = 1\frac{1}{2}$ to 2
public halls	$n = \frac{3}{4}$ to 2
rooms with open fireplaces	$n = 4$ to 6

$t - t_0$ = temperature difference between inside temperature and outside temperature t_0

Carpenter's Rule: Heat loss in Btu per hour =
$(0.02\ C + G + .25W) \times (t - t_0)$

Mills Rule: Heat loss in Btu per hour =
$1.25\ (C + 10W + 100G) \times \left(\frac{t - t_0}{70}\right)$

Harding and Willard Rule: Heat loss in Btu per hour =
$(1.2P + .25W + G)\ (t - t_0)$

N. W. Rule: square feet of radiation required =
$\frac{1}{3}\ G + 1/10\ W + 1/200\ C$

C B T Rule: Heat loss in Btu per hour =
$72W + 22W + 1.4C$

To the heat losses figured as above, allowances for exposure should be made as shown in the following table.

Example: Heat loss × Exposure Factor = Total heat loss.

Exposure	Factor
North, Northeast and Northwest where winds are to be counted as an important factor	1.25
East or West Walls moderately exposed	1.15
South walls	1.00

255. How to Recognize Heating Systems Quickly

Heating systems may be classified as follows:

A Steam
B Hot Water
C Warm Air (Hot Blast)

D Combination of above, known as indirect or split systems.

In air conditioning we are more concerned with the combination of "C" and "D" than "A" and "B", although it is necessary to be able to recognize quickly various types of systems.

A steam system radiator is *always* equipped with a protruding automatic air relief vent valve. A hot water system *never* has an automatic vent valve. A piped warm air system has the tell-tale registers along the baseboard. In the cellar a warm air furnace is obvious. A steam boiler is always equipped with a pressure gage and a glass-tube water gage. A hot water boiler might be equipped with a thermometer and water pressure gage but *never* with a glass-tube water gage.

256. How to Estimate Radiation Ratings and Boiler Capacity by Condensed Method

On domestic and small heating plants where it is impossible to obtain the rating of the boiler from either the boiler manufacturer, heating contractor or owner, the boiler capacity and radiation requirements may be estimated roughly as follows:

1 sq ft of steam radiation will heat:

Dwellings—cold rooms	50 cu ft
ordinary rooms	60 to 70 cu ft
warm, sunny rooms	75 cu ft
Stores—wholesale	125 cu ft
retail	100 cu ft
Office Rooms	75 cu ft
Churches, Theaters	125 to 150 cu ft
Factories, etc.	200 cu ft

After determining the total number of square feet of actual radiation required to heat the building, 25% of this figure should be added for the risers and then 25% of this sum should be added to determine the proper size of the boiler for steam systems. For hot water systems 50% instead of 25% should be added.

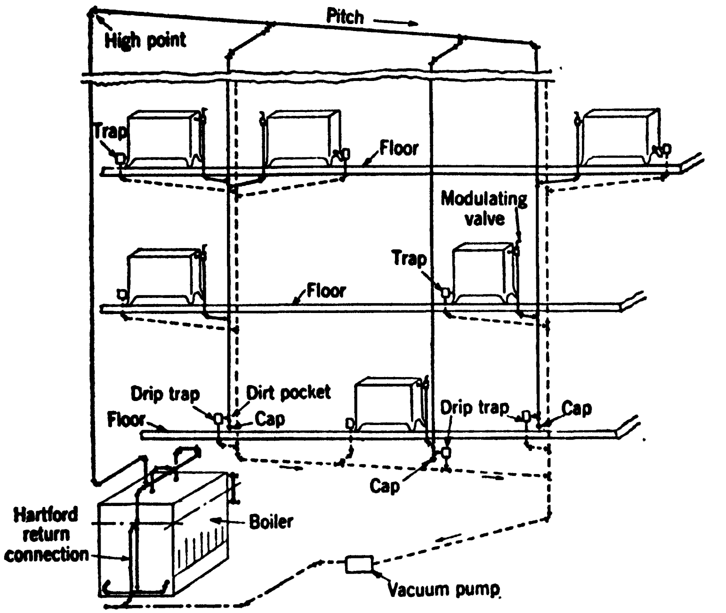


FIGURE 38

Typical Down-Feed Vacuum System

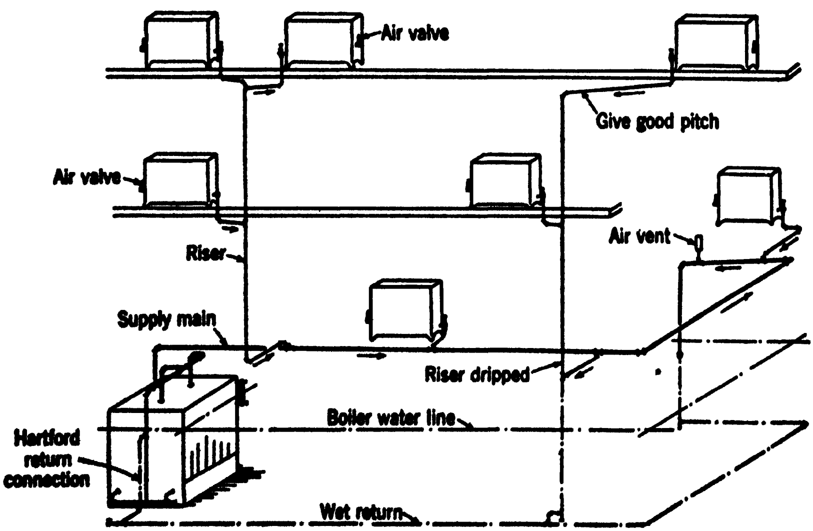
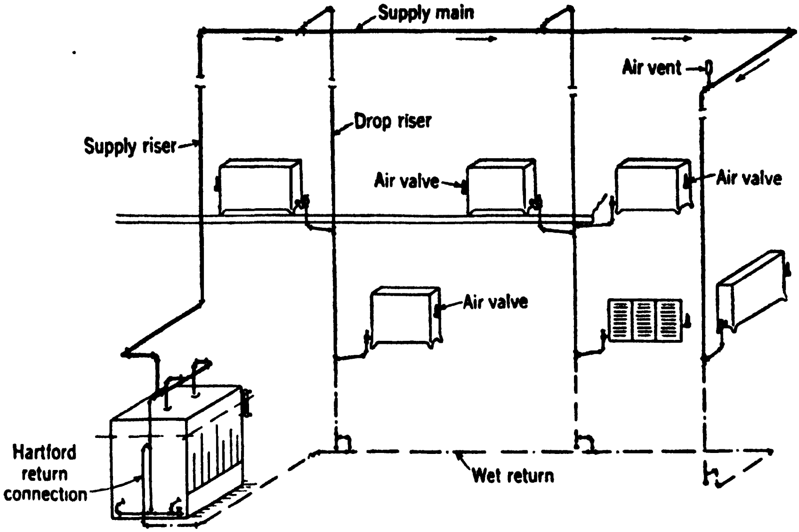
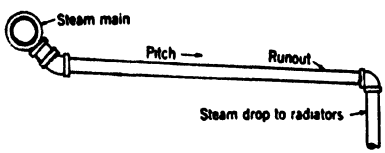


FIGURE 39

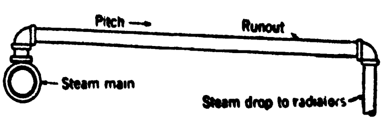
Typical Up-Feed Gravity One-Pipe Air-Vent System



TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM



STEAM RUNOUTS DRIPPING MAIN



STEAM RUNOUTS WITH MAIN DRIPPED AT END ONLY

FIGURE 40

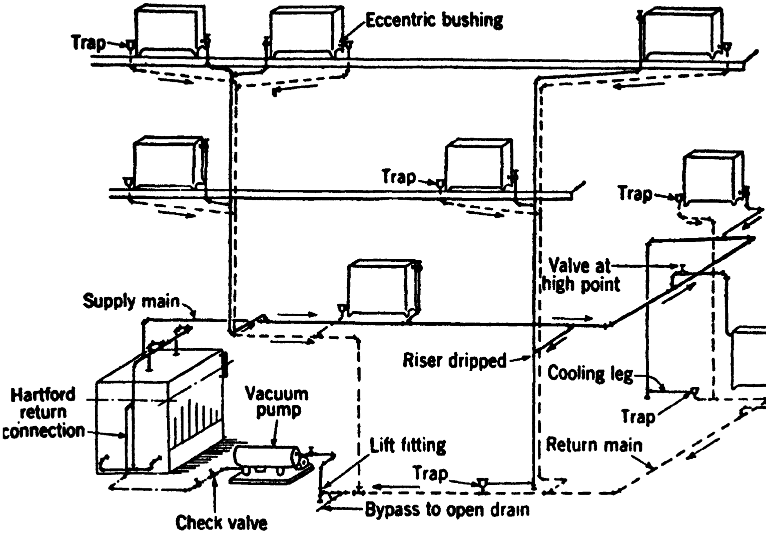


FIGURE 41
Typical Up-Feed Vacuum Pump System

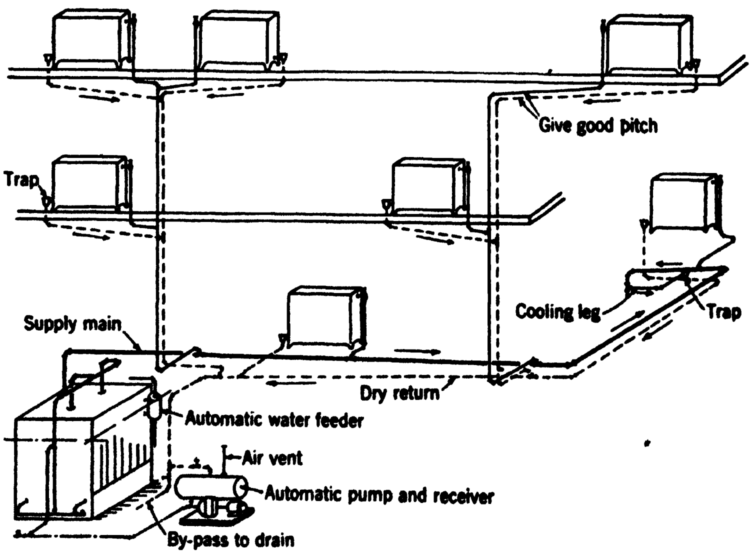


FIGURE 42
Typical Installation Using Condensation Pump

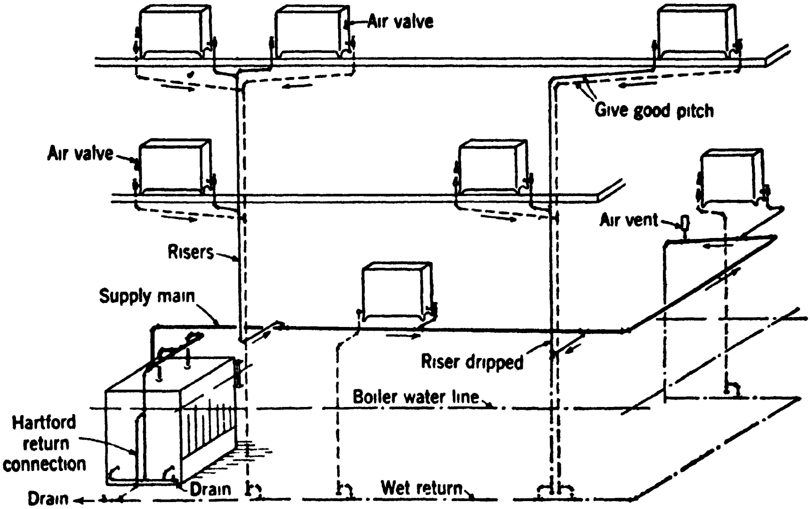


FIGURE 43

Typical Up-Feed Gravity Two-Pipe Air-Vent System

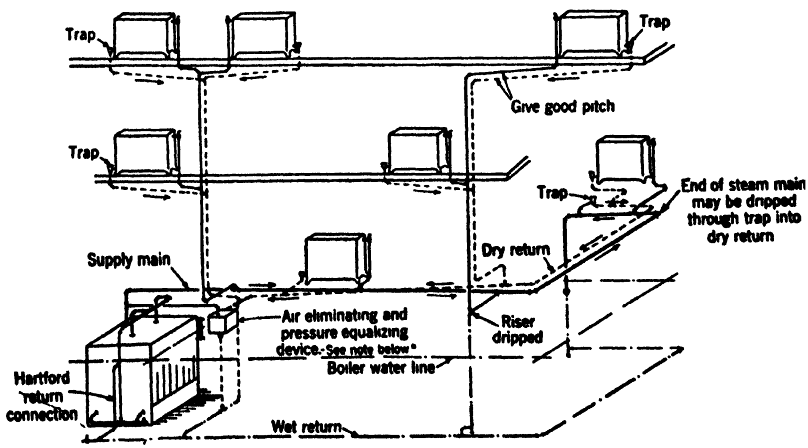


FIGURE 44

Typical Up-Feed System with Automatic Return Trap
Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

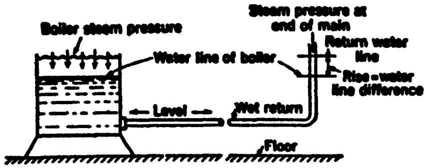


FIGURE 45

Difference in Steam Pressure on Water in Boiler and at End of Steam Main

257. Piping for Steam Heating Systems

Steam piping systems distribute steam not only at full design load but also at excess or partial loads. The functions of the piping system are the distribution of the steam, the return of the condensate and, in systems where no local air vents are provided, the removal of the air. The determination of pipe sizes depends on:

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 permissible for quiet and dependable operation of the system, considering the direction of the condensate flow.

3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.

258. Pipe Sizing for Blast Heating Coils

The size of the pipe mains and connections for indirect heating units is determined on the same basis as the size of radiator piping, but the equivalent direct radiation must be ascertained for each row of heating stacks, and for each unit in the stack.

$$EDR = \frac{Q \times 60 \times (t_a - t_b)}{55.2 \times 240} = \frac{Q \times (t_a - t_b)}{220.8}.$$

where

EDR = equivalent direct radiation, square feet

Q = volume of air, cubic feet per minute

t_a = the temperature of the air entering the row of heating units, in degrees F

t_b = the temperature of the air leaving unit, in degrees F
60 = number of minutes per hour
55.2 = quantity of air in cubic feet, heated by one Btu
240 = EDR, Btu per square foot

It is usually 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.

259. Pressure Drop Selection

In determining the pressure drop the following table is convenient for preliminary estimating purposes.

TABLE 122

System	e.l.r.* Drop in Pressure
Vacuum—over 500 ft	½ lb total
Vacuum—under 500 ft	¼ lb total
Vapor—over 400 ft	¼ lb total
Vapor—under 400 ft	⅛ lb total
One pipe gravity—under 250 ft	⅛ lb total
One pipe gravity—over 250 ft	¼ lb total

* Equivalent length of run.

The drop in pressure should not exceed one ounce per 100 ft e.l.r. nor should it exceed 1 lb total for the system. The condensate flows against the steam in all up-feed risers and with the steam in down-feed risers.

TABLE 123

SIZE OF SUPPLY AND RETURN MAINS (VACUUM SYSTEM)

Size Vapor Supply Main Inches	Drip from Supply Main Inches	Size Return Main Inches	Radiation on Mains Square Feet	Heat Release Btu
1	¾	¾	50	12000
1¼	¾	1	100	24000
1½	¾	1	200	48000
2	1	1	350	84000
2½	1	1¼	600	144000

TABLE 123 (*continued*)

Size Vapor Supply Main Inches	Drip from Supply Main Inches	Size Return Main Inches	Radiation on Mains Square Feet	Heat Release Btu
3	1	1½	1000	240000
3½	1¼	1½	1500	360000
4	1¼	2	2500	600000
4½	1¼	2	3500	840000
5	1½	2½	5000	1200000
6	1½	3	8000	1920000

Table 123 is based on a 1 ounce pressure loss in 100 feet of equivalent length of pipe, steam and water flowing in the same direction.

All horizontal supply connections to coil are to be one size larger than supply valve.

All horizontal connections to supply risers are to be one size larger than risers.

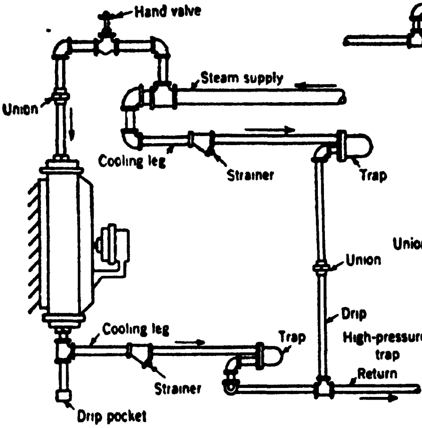
Pitch all horizontal piping ¼ inch in 10 feet in direction of flow and provide for all necessary expansion and contraction.

In laying out vapor systems of heating, equalize the load by making the circuits as short and direct as possible. Keep the main supply lines liberal in size up to the last connection. At the end of each supply circuit, run independent drips back to the boiler and connect them to boiler return header.

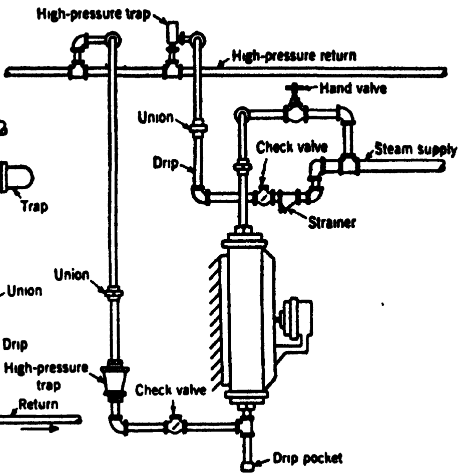
TABLE 124

SIZE OF SUPPLY RISERS, RETURN RISERS AND SUPPLY VALVES FOR COILS

Size Steam Riser Inches	Size Return Riser Inches	Square Feet Radiation On Riser	Size Supply Valve Inches	Square Feet Radiation On Supply Valve	Maximum Btu Per Valve
¾	½	40	½	1 to 30	7200
1	¾	80	¾	31 to 75	18000
1¼	1	140	1	76 to 125	30000
1½	1	250	1¼	126 to 200	48000
2	1	400	1½	201 to 300	72000
2½	1¼	600	2	301 to 400	96000
3	1¼	800	—	——	——
3½	1½	1,500	—	——	——



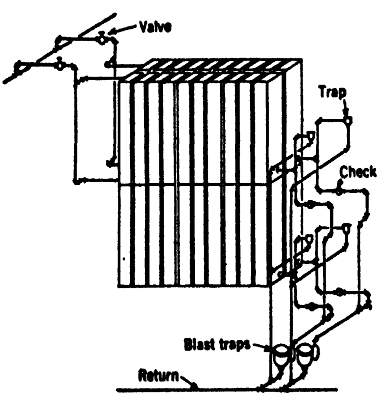
Unit Heater Connection for Vacuum or Vapor System Discharging Condensation into Dry Return



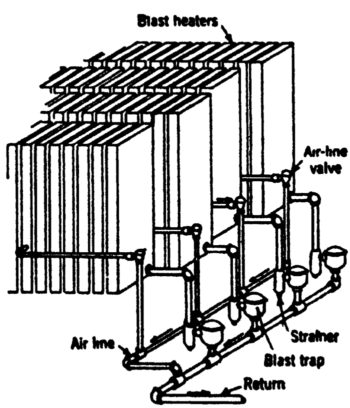
Method of Connecting Unit Heater to High Pressure Return

Figure 46A

Unit Heater Connections



Supply and Return Connections for Heating Units of Central Fan Systems



Typical Connections to Central Fan System Heating Units Exceeding 12 Sections

Figure 46B

Unit Heater Connections

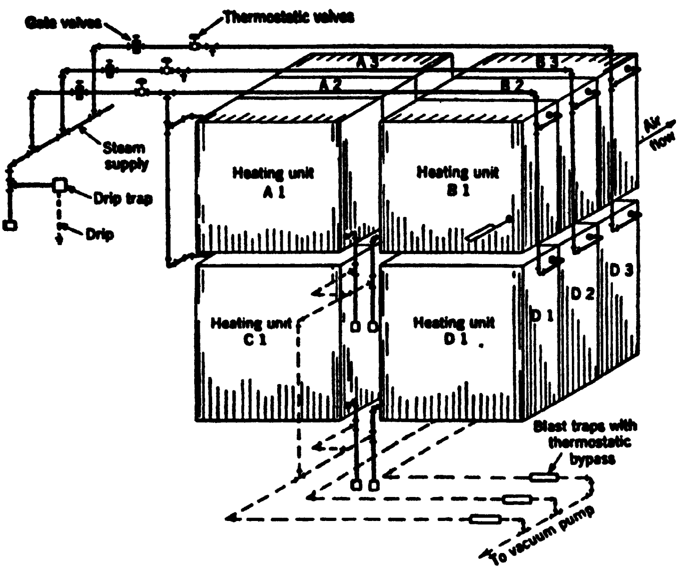
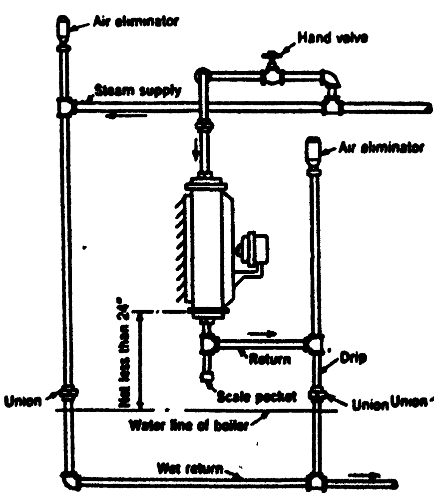
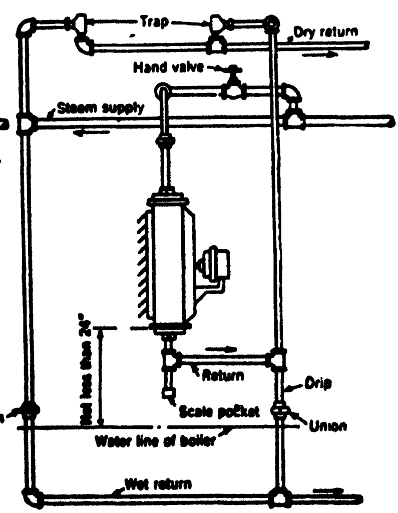


FIGURE 47A

Typical Piping for Atmospheric and Vacuum Systems with Thermostatic Control (Central Fan System)



UNIT HEATER CONNECTION TO ONE-PIPE GRAVITY STEAM SYSTEM



UNIT HEATER CONNECTION TO GRAVITY SYSTEM WITH WET AND DRY RETURNS

FIGURE 47B

Central Heating Units

260. Radiators and Convectors

Rooms and buildings may be heated with warm air radiators or convectors; in some cases panel heating or electric space heaters are used. Originally, stoves were employed and before stoves, fireplaces. Both stoves and fireplaces introduce separate fires for each unit, requiring separate fuel and maintenance, so that they have been generally discarded in favor of central heating systems where one fire is required and the fuel, dirt and other objectionable features can be confined to a single room where they do not affect the balance of the building. Radiators are of two general types: the radiator exposed to view in the room and the radiator concealed from view either behind grilles, covered up in the walls, or enclosed in a cabinet of furniture steel. Radiators which are concealed from view have to transfer all of their heat to the air flowing across or through them and set up convection air currents; therefore they are called *convectors.*

With the exposed radiator a large portion of its heat is given off in the shape of heat rays which are emitted similarly to light rays and proceed in a straight line until they impinge on some object which absorbs them. Sun rays act in a similar manner. No radiant heat is received from convectors owing to their heat rays being cut off by the concealment.

A radiator set back of a grille covering the entire face of the radiator does give off some radiant heat and, hence, usually is termed a *concealed* radiator, while a radiator set in a recess or niche in the wall, but without grilles, is usually termed a *recessed* radiator.

The ordinary steam radiator of today is of the cast iron *tubular* type, with small vertical tubes or bars, as distinguished from the older type (now no longer manufactured) in which the vertical portions were large and which were called *column* radiators. This type of radiator is still found in many buildings erected over ten years ago.

261. Heat from Radiators

At the present time tubular radiation is rated on the quantity of heat given off:

a. For steam at 1 lb gage pressure in the radiator, when the radiator is standing in a room at 70° F

b. For hot water at an average temperature of 175° F passing through the radiator, when the radiator is standing in a room at 70° F

Under these conditions the steam radiator will have a nominal rating of 240 Btu per square foot per hour, and the hot water radiator will have a nominal rating of 150 Btu per square foot per hour. This applies to everything except wall radiators and pipe coils.

It is usually assumed that the wall radiator delivers 300 Btu per square foot (of rated surface) per hour; pipe coils vary somewhat, some delivering up to 400 Btu per square foot (of actual pipe surface on the coil) per hour; however, where several pipes are used in the coil, they are usually considered to deliver 300 Btu per hour, as do wall radiators. In cases where the wall radiator or pipe coil is installed on the ceiling, 240 Btu per square foot is assumed because the radiator is placed in a position where the temperature surrounding the radiator or coil will be higher than if it were placed at a lower level.

Wall radiators usually come in sections containing 5 sq ft, 7 sq ft, or 9 sq ft and the number of sections times the area in square feet of each section, multiplied by the heat emission of 300 Btu, if on the side wall, or 240 Btu, if on the ceiling, will give the heat emitted by such radiators. In the case of pipe coils, the outside surface of the pipe must be calculated by determining the length of the pipe in the coil and multiplying this length by the surface area of lineal foot of pipe.

The outside surface of standard weight pipe is shown in the following:

Nominal Size of Pipe Diameter, Inches	External Surface Square Feet Per Lineal Foot
0.5	0.554
0.75	0.867
1.0	1.36
1.25	2.16
1.5	2.84
2.0	4.43

The length of pipe, multiplied by the number of square feet per lineal foot, and then by 300 Btu (if on the wall) or 240 Btu (if on the ceiling) will give the heat emission for the pipe coil.

262. Heat from Convectors

The quantity of heat emitted by the convector type of radiator is dependent on the height of the flue and the size of radiator installed. The same convector will give off a greater quantity of heat with a high flue than it will with a low flue because the higher flue produces a more rapid flow of air through the convector due to the chimney action. There is no standard type or design of convector, each manufacturer making his own model,

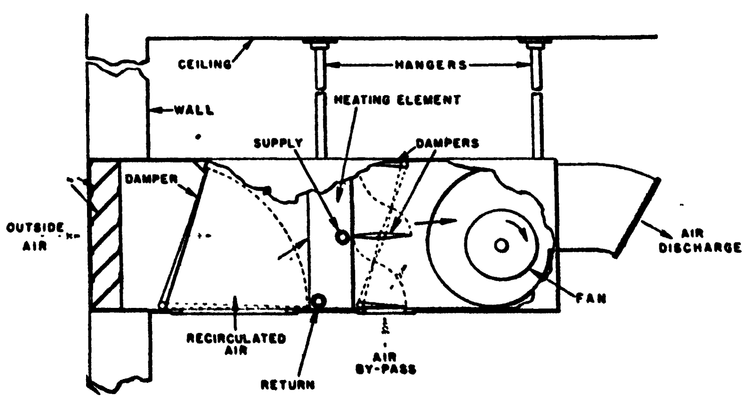


Figure 48

Suspended Type Unit Heater, Housed Type Fan and for this reason the quantity of heat emitted at the flue height to be used must be obtained from the catalogs of the particular make of convector to be installed.

263. Unit Heaters

Unit heaters consist of a convector-like surface, usually of copper, behind which a fan or fans is located and blows the air through the heating element at a high velocity. Due to the high velocity of the air passing through the heating element, these elements transmit a great deal of heat in comparison to their size, and the air discharging from the unit may be directed downward by adjustable louvers so as to throw the heated air toward the floor. Such units are frequently used in industrial work, gymnasiums, garages and so on. In the different catalogs, their heat delivery is generally quoted on an inlet temperature of air at 60 degrees and for various steam pressures.

264. Unit Ventilators

The unit ventilator is another type of heating unit which combines heating with a supply of outside air; where the unit heater ordinarily recirculates 100% of the air all of the time, the unit ventilator can take 100% of its supply from the outside, although some of them are equipped with recirculation dampers so that they can recirculate the air when the room is unoccupied. This type of unit is much used in schools where its capacity must be 30 cu ft of air per minute for each pupil in the class room.

265. Radiators with Hot-Water Circulation Systems

Recently the scheme of using a small centrifugal pump as an aid in circulating hot-water systems has come into vogue, and the tank-in-basement arrangement is used to keep water pressure on the radiators. Under these conditions water as hot as 200° F is recommended; this makes it possible to reduce the size of the radiators. The increase in heat emitted from hot water radiators under different average temperatures of water passing through the radiators is as follows:

Average Temperature of Water	Btu Per Square Foot Per Hour
175° F	150
180° F	160
190° F	178
200° F	200
205° F	216
215° F—same as steam at 1 lb	240

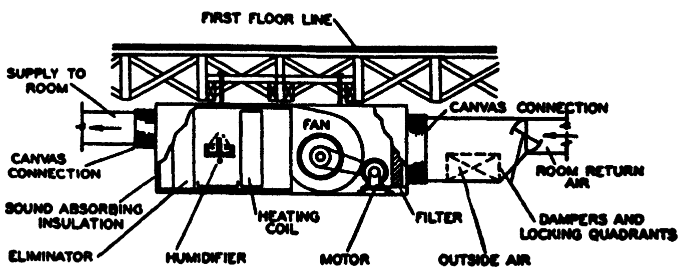


FIGURE 49

Typical Unit Humidifier of the Spray Type with Steam Coil to Preheat the Air for Residences

266. Radiators at Other Than Standard Conditions

While the rating on radiators is based on 70° F room temperature, the amount of heat given off from a radiator depends entirely, or almost entirely, on the temperature difference between the water or steam temperature in the radiator and the surrounding room temperature. Thus, under standard conditions the steam at one pound gage pressure in the radiator has a temperature of 215° F and the surrounding air in the room is at 70° F, giving a temperature difference of

$$215° \text{ F} - 70° \text{ F or } 145° \text{ F}$$

If the radiator under this condition gives off 240 Btu, for each degree difference between the steam temperature and the room temperature the number of Btu given off will be

240 Btu/145° F or 1.66 Btu per degree difference

In a hot water radiator with an average water temperature of 175° F and a room temperature of 70° F the average temperature difference is

175° F − 70° F or 105° F

so that if the radiator gives off 150 Btu under these conditions the number of Btu per degree of temperature difference amounts to

150 Btu/105° F or 1.43 Btu per degree.

This is slightly less than that of the steam radiator owing to the air flow through the radiator being somewhat reduced by the lesser temperature difference.

A steam radiator standing in a room at 60° F will give off more heat because of the greater temperature difference, the rate being increased in this case to

(215°F − 60° F) × 1.66 Btu per degree or 257 Btu

while if the same radiator had the room temperature go up to 80° F it would give off less heat or

(215° F − 80° F) × 1.66 Btu per degree or 234 Btu

and similar increases and reductions occur with hot water. It is evident that if the room temperature with a steam radiator ever went to 215° F, or if it ever rose to 175° F with a hot water radiator, the room in each case would have a temperature the same as that of the heating medium and all delivery of heat from the radiator would cease as heat cannot flow without a temperature difference.

267. Heating with Air

Heating with air has gained impetus with the advent of air-conditioning systems where air is used for cooling; with the addition of very little extra equipment heated air may be supplied for winter use, and year-round air conditioning maintained. When heat is supplied to a room or building by means of warm air, the heat loss from the building must be constantly replaced by the hot air in order to maintain a constant temperature condition or *heat balance.* If a room loses 20,000 Btu during a period of one hour, the warm air supplied must add 20,000 Btu to the room during the same period; if this is not done the temperature in the room will fall, while, if more than 20,000 Btu are added, the temperature will rise. The degree of rise or fall is determined by the increase or reduction of the heat supplied when compared to the heat actually needed.

Where there is more heat supplied than is actually required, the result is a rise in room temperature until the heat loss from the room due to its increased temperature will equal the larger quantity of heat supplied, while, in the case of a shortage of heat, the room temperature will fall until the heat loss from the room is reduced to that of the smaller quantity of heat; in each case a heat balance is finally obtained and the heat supplied must equal the heat lost.

268. Heat Supplied by Air

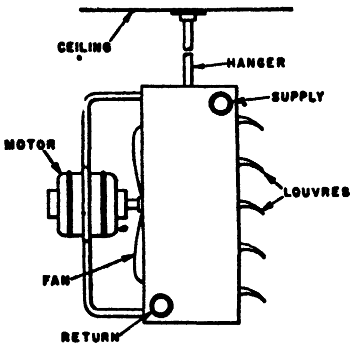
When heated air is delivered into a room it will not add heat to the room unless the temperature of the air is higher than the temperature of the room; when the temperature of the air is less than that of the room, cooling of the room results. The quantity of heat supplied will be governed by:

a. The *temperature* of the air supplied

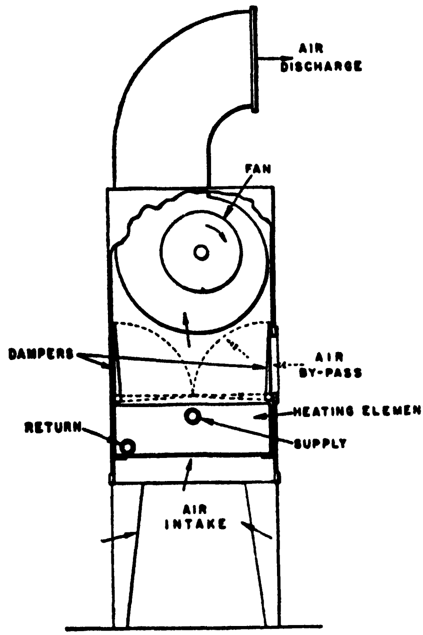
b. The *quantity* of air supplied

The temperature of the air supplied always must be considered in relation to the temperature of the room. For instance, 1,000 cfm at 100° F will not supply the same quantity of heat to a room at 80° F as it would to a room at 70° F. The heat delivered by each pound or cubic foot of air supplied, then, depends on the temperature difference between the supplied air and the room.

The previous statements will be understood more readily if one realizes that the heat supplied by the incoming air is only



SUSPENDED UNIT HEATER, PROPELLER TYPE FAN



FLOOR MOUNTED UNIT HEATER, HOUSED TYPE FAN

FIGURE 50

Unit Heaters

the heat released when the air cools down from its inlet temperature to the room temperature; thus, in the case of a room at 80° F an air supply at 100° F will only cool from 100° F (the inlet temperature) down to 80° F (the room temperature) while in the room at 70° F the same air would cool from 100° F (inlet temperature) down to 70° F (room temperature). This gives

For the room at 80° F, 100° F − 80° F or 20° F

For the room at 70° F, 100° F − 70° F or 30° F

and the quantity of heat supplied to the room at 80° F will be

$$\frac{20}{30} \text{ or } \frac{2}{3}$$

as much with the same amount of air at the same inlet temperature.

269. Deciding on the Supply Air Temperature

The temperature at which the supply air shall enter the room is one abitrarily chosen before the design is completed and it will be found that the higher the air temperature is, the smaller will be the quantity of air required. This is evident as the quantity of heat supplied is a function of the difference between the temperature of the air and that of the room, so that the greater this temperature difference becomes, the smaller will be the quantity of air needed. Against this is the fact that the higher the air temperature desired the larger the heater employed must be and the greater the heat loss from the air between the heater and the room. It is evident that, if it were attempted to heat air to 215° F with a steam heater having steam at 1 lb gage pressure, as the air approached 215° F the quantity of heat transmitted from the heater to the air would become smaller and smaller until at 215° F there would be no transmission of heat at all since all temperature head, or difference in temperature, between the air and the heater has been lost.

To avoid excessively large heaters and increased losses of heat from the air in transmission, a compromise figure for the air temperature is usually selected, often between 125° F and 180° F, with 140° F as a reasonable average. This temperature may be changed, if desired, in order to suit some other feature of the installation.

270. Calculating the Amount of Air Required

In order to calculate the quantity of air required to heat a given room it is necessary to know the heat loss from the room under the lowest outside temperature for which the system is to function. This does not necessarily mean the lowest outside temperature ever recorded, but usually is considered to be ten degrees above the lowest recorded outside temperature; in the vicinity of New York and other cities in the same latitude this is 0° F. The heat losses from the room are then calculated on the basis of maintaining 70° F in the room when the outside temperature is 0° F, or at a 70° difference. It is to be noted that, in calculating the heat loss for a room, no consideration is given to the matter of lights, body heat or sun effect as the conditions under which the room has to be heated may include none of these items, although they may exist to a greater or lesser extent at various times. It would be ridiculous to design a heating system which requires all of the occupants to be present before the building could be brought up to temperature, or which requires that the sun be shining or the electric lights turned on!

Assuming that the heat loss from the room has been calculated by some approved method, e.g., as previously explained, it is necessary to set the temperature at which the air will enter the room.

Example. Suppose that air at 125° F is used to heat a room where the heat-loss calculations show an hourly heat loss of 180,000 Btu per hour; how many cubic feet of air will be required per minute?

This problem may be solved in several ways, one of which is to determine the number of Btu delivered to the room by each cubic foot of air under the conditions stated and then dividing the total required by the quantity obtained from each cubic foot of air. This will give the number of cubic feet per hour and this quantity divided by sixty gives the number of cubic feet per minute. Using this method yields the following results:

Temperature difference between air and room: 125° F — 70° F = 55° F

1 Btu will raise 1 cu ft of air 55 degrees

1 cu ft of air dropping 55 degrees will give up

$$\frac{55}{55} \text{ or } 1 \text{ Btu}$$

To supply 180,000 Btu will require 180,000 Btu or 180,-000 cu ft per hour of air

$\frac{180{,}000 \text{ cu ft per hour}}{60 \text{ min per hour}}$ gives 3,000 cu ft per minute as the quantity required to heat the room.

This may also be expressed as a formula: ·

$$\frac{\text{Btu per hour required} \times 55}{td \times 60} = \text{cubic feet per minute required}$$

where

Btu required is the heat loss from the room per hour

55 is the number of cubic feet of air at 1 degree temperature difference

td is the temperature difference between the air and the room

60 is the number of minutes in one hour.

In cases where summer cooling has already determined the amount of air to be handled by the system, it is possible to arrange the air temperature for heating so that the same amount of air will be required for heating as for cooling; or, if four-speed motors are used on the fan, so that ¾ or ½ of the amount of air used in summer can be employed in winter for heating. This is done in the following manner:

Taking the previously noted case with 180,000 Btu winter room loss and assuming that the summer cooling shows that about 5,000 cfm must be circulated, at what temperature must the air be used in winter to require (1) the same quantity of air as in summer? (2) three-quarters as much air as in summer? (3) one-half as much air as in summer?

In (1) the quantity of air will be 5,000 cfm, the same as used in summer cooling; in (2) the quantity of air will be 3,750 cfm; in (3) the quantity of air will be 2,500 cfm. In each case, however, the air must supply 180,000 Btu per hour to the room in winter. This may be solved by transposing the formula above to read

$$\frac{\text{(Btu per hour required) 55}}{\text{(cubic feet per minute) 60}} = td$$

in which the Btu per hour required, multiplied by 55, represents the number of cubic feet of air required per hour at 1° temperature difference cfm × 60 represents the number of cubic feet supplied per hour; dividing the quantity above the line by the quantity below the line will yield the necessary temperature difference to make the lower quantity equal to the upper quantity.

Applying this to (1) yields

$$\frac{(180{,}000 \text{ Btu}) \; 55° \text{ F}}{(5{,}000 \; cfm) \; 60} = 33° \text{ F } td$$

and the entering air temperature will be

$$70° \text{ F} + 33° \text{ F} = 103° \text{ F}$$

For (2) the result becomes

$$\frac{(180{,}000 \text{ Btu}) \; 55° \text{ F}}{(3{,}750 \; cfm) \; 60} = 44° \text{ F } td$$

and the entering air temperature will be

$$70° \text{ F} + 44° \text{ F} = 114° \text{ F}$$

For (3) the result becomes

$$\frac{(180{,}000 \text{ Btu}) \; 55° \text{ F}}{(2{,}500 \; cfm) \; 60} = 66° \text{ F } td$$

and the entering air temperature will be

$$70° \text{ F} + 66° \text{ F} = 136° \text{ F}$$

271. Saving in Using Less Air at Higher Temperature

There is an operating cost saving in using less air at a higher temperature as long as the higher temperature does not become excessive, i.e., over 175° F. This is due to the fact that the motor, when operating the fan at ¾ or ½ speed, will require less current to run than when delivering the full quantity of air. There will be no saving in the heating as the same number of Btu is supplied to the room in each case. It is also interesting to note that, when air is taken from the outside, it produces no heating effect until its temperature is higher than that of the room which it enters because the heating effect is limited to the temperature difference between the hot air and the room to be heated. This means that when outside air is used without recirculation and air at 140° F is used for heating, half of the heat will go to warm the air from 0° F to 70° F and the other half will be used to warm the air from 70 to 140° F, yet the last half of the heat is the only part which actually goes to heat the room; the inefficiency of such an arrangement is evident. If 100% recirculation is employed, then all of the heat added to the air goes to heat the room as the air comes back into the recirculation chamber at 70° F and all the heat added is utilizable. If, as is the usual case, 75% recirculation is used, ¼ of the air will have to be heated from 0° F to 70° F in the coldest weather and will supply no heat to the room for this portion of its temperature rise.

On the other hand, the 25% of outside air will prevent inward leakage into the room, whereas a radiator system supplying no outside air would be subject to infiltration, and the heat requirements, or fuel used, would be approximately the same for either an air heating system or a radiator heating system. The electric current used to drive the fan would be an extra expense on the air system, however.

272. Warm Air Furnace Systems

Many warm air furnaces are now equipped with a propeller fan and return circulation. When humidifying equipment is added to such an installation, it may be called a winter air-conditioning system. Under gravity operation, the velocity of air in warm air furnace pipes is generally assumed to be 240 fpm to the first floor outlets, 360 fpm to the second floor outlets, and 480 fpm to the third floor outlets, if any. When a fan, of proper size and type, is added to the system these velocities probably can be doubled.

273. Drafts from Warm Air

While great care must be exercised in the design of cooling systems where cold air is introduced into the room, no such difficulty exists in the case of warm air; a warm air draft is far less disagreeable and certainly is not dangerous. The principal thing to be kept in mind is that the air should be distributed so as to reach the coldest spots in the room in order to prevent unequal heating; many systems employ registers in the window seats or sills to counteract the cold down-drafts from the glass. In the absence of any other consideration the inlet registers or grilles for warm air may be sized on a velocity of 250 fpm, but where summer air conditioning is supplied through the same grilles the size of the grille will have to be governed by the summer requirements of cool air and low velocity so as to prevent objectionable cold drafts.

274. Split Systems

Since it is necessary to keep buildings up to temperature or nearly so throughout the winter season and whether they are occupied at all times or not, a winter warm air system will require constant operation throughout the winter season. This results in a sizable bill for fan power which may be overcome by installing radiation to warm the building to 70° F, using the fan system to supply the necessary air for ventilating only, in which case the air is put into the room at or about 70° F. It will be seen that this air, at or nearly at the same temperature as the room, will supply no heat but does permit winter air conditioning by simply humidifying the air so as to maintain the humidity desired. It is customary in such installations to omit the matter of infiltration in calculating the radiator surface by assuming that the 25% outside air used with the conditioning system will maintain a positive pressure in the room.

Note: To select heaters in air conditioner the following calculations are necessary. Design will determine whether pre-heaters, re-heaters, booster heaters are required. The air conditioner frequently is used only to supply tempered air slightly above room temperatures because building heat losses are offset by direct radiation, lighting, body heat or appliances as in restaurants, theaters, etc. The following chart does not allow credit for heat gain from body heat, lighting and appliances. In theaters, restaurants, etc. this factor should be considered. This item, if omitted, may be considered a factor of safety.

HEATING AND HUMIDIFYING ESTIMATE

ESTIMATED BY______ DATE______ PLAN NO.______ SHEET NO.______

JOB NAME______ LOCATION______

ROOM ______ SIZE: ____ × ____ = SQ FT × ____ = ____ CU FT

ITEM No.	CONDITIONS	DRY-BULB	WET-BULB	DEW POINT	GRAINS PER POUND AIR	NOTES
1	Room					
2	Outside					
3	Diff.					

	ITEM	DIMENSIONS	AREA SQ FT	FACTOR *U*	TEMP. DIFF.	SENSIBLE HEAT BTU PER HOUR
	Transmission Loss	....	....	....	....	....
4	All Glass					
5	All Walls					
6	All Partitions					
7	Roof					
8	Floor					
	INFILTRATION HEAT LOSS Method 1—Air Changes per Hr____ × Room Vol. = ____C.F.H. Method 2—(Alternate)____Ft of Window Crack Refer to Data on Windows for C.F.H.					....
9	INFILTRATION____C.F.M. × TEMP. DIFF.____ × 1.08					
10	ROOM HEAT LOSS (ITEMS 4 TO 9 INCLUSIVE)					
11	OUTSIDE AIR LOSS: ____C.F.M. × Temp. Diff.____ × 1.08 × By-Pass Factor By-Pass Factor Depends on Design, Allow 0.1 if By-Pass Factor Unknown.					....
12	GRAND TOTAL HEATING REQUIREMENTS 10 + 11					

HUMIDIFICATION REQUIREMENTS:

$$\frac{\text{\_\_\_\_C.F.M. Infilt.} + \text{\_\_\_\_C.F.M.O.A.} \times \text{Item 3 (Gr. per Lb.)}}{1580} =$$

____Lb Moist. per Hour

Factor 1580 Converts to Pounds of Water per Hour

13 SUPPLY AIR TEMPERATURE DIFFERENTIAL:

$$\frac{\text{Item 10}}{1.08 \times \text{\_\_\_\_C.F.M. Air Circulation}} = \text{\_\_\_\_Temp. Diff.}$$

Temp. Heated Supply Air

Item 13 + Item 1 (Dry Bulb) = ____Degrees

FIGURE 51

CHAPTER XIV

COOLING LOAD

Air-conditioning load computations are similar to heating computations but are more intricate in that the air-conditioning engineer must be thoroughly familiar with the following elements of design:

Outside design conditions
Inside comfort conditions
Sun effect, heat lag and transmission
Ventilation requirements
Thermodynamic properties of air

In proportioning the cooling load required for a large building, where the sun effect forms a large percentage of the heat gain, the air distributing system may be divided into zones in accordance with the movement of the sun across the building. It is preferable that each exposure be zoned separately but north and east exposures may be placed in one zone and south and west exposures in another. The maximum volume of air in circulation is based on the total heat gain, which includes transmission and the sun effect apportioned to each zone. When the total refrigeration load is calculated, a search is made for the time of the peak load, but generally the roof heat gain and one exposure only need be included, as the maximum sun effect does not occur simultaneously on any two exposures.

275. Design Outside Temperature

Summer dry-bulb and wet-bulb temperatures of various locations in the United States of America are given in Table 124A.

TABLE 124A

DESIGN CONDITIONS

State and City	Heating Days S = 175–200 M = 150–175 F = Less 150	Average Latitude °	Heating Outside Dry-Bulb	Heating Inside Dry-Bulb	Cooling Outside Dry-Bulb	Cooling Outside Wet-Bulb	Cooling Inside Effective T.° Occupancy Over 40 Min	Cooling Inside Effective T.° Occupancy Under 40 Min
Ala.—Birmingham	F	32	10	70	95	78	74	75
Ariz.—Tucson	M	32	25	70	105	72	74½	75½
Ark.—Little Rock	M	35	5	70	95	78	74	75
Cal.—San Francisco	F	38½	35	70	85	65	72	72
Cal.—Los Angeles	F	34	35	70	90	70	73	73½
Col.—Denver	S	40	–10	70	95	64	73	73½
Conn.—Hartford	S	43	0	70	95	75	74	75
Del.—Wilmington	S	39	0	70	95	78	74	75
D. C.—Washington	M	39	0	70	95	78	74	75
Fla.—Miami	F	26	35	70	91	79	74	75
Ga.—Atlanta	F	34	10	70	95	76	74	75
Ida.—Boise	S	44	–10	70	95	65	73	73½
Ill.—Chicago	S	42	–10	70	95	75	74	75
Ind.—Indianapolis	S	40	–10	70	95	76	74	75
Iowa—Des Moines	S	41	–15	70	95	77	74	75
Kan.—Topeka	S	39	–10	70	100	75	74	75
Ky.—Louisville	M	38	0	70	95	78	74	75
La.—New Orleans	F	30	20	70	95	79	74½	75½
Me.—Portland	S	44	–5	70	90	72	73	73½
Md.—Baltimore	M	39	0	70	95	78	74	75
Mass.—Boston	S	42	0	70	92	75	73½	74
Mich.—Detroit	S	42	–10	70	95	75	74	75
Minn.—Minneapolis	S	45	–20	70	95	75	74	75
Mo.—Kansas City	S	39	–10	70	100	75	74	75
Mon.—Billings	S	46	–25	70	90	66	72½	72½
Neb.—Omaha	S	41	–15	70	95	77	74	75
Nev.—Reno	S	40	–5	70	95	65	73	73½
N. J.—Newark	S	41	0	70	95	75	74	75
N. Y.—New York	S	41	0	70	95	75	74	75
N. Y.—Syracuse	S	43	–10	70	93	75	74	75
Ohio—Cleveland	S	41	–5	70	95	75	74	75
Okla.—Tulsa	M	35	0	70	101	77	74½	75½
Ore.—Portland	M	45	10	70	90	68	73	73½
Pa.—Philadelphia	S	40	0	70	95	78	74	75
R. I.—Providence	S	42	0	70	93	75	74	75
S. C.—Charleston	F	33	15	70	95	78	74	75
Tenn.—Memphis	M	35	0	70	95	78	74	75
Tex.—Dallas-Ft. Worth	F	33	10	70	100	78	74½	75½
Tex.—Houston-San Antonio	F	29	20	70	95	78	74	75
Utah—Salt Lake City	S	41	–10	70	95	65	73	73½
Va.—Richmond	M	38	10	70	95	78	74	75
Wash.—Spokane	M	47	–15	70	93	65	73	73½
Wash.—Seattle	M	47	15	70	85	65	72	72
W. Va.—Charleston	S	38	0	70	95	75	74	75
Wis.—Milwaukee	S	43	–15	70	95	75	74	75

It will be noted that the temperatures are not the maximum 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 1 to 4% of the time, and in spells of such short duration that it is not practical to design a cooling system covering this maximum range. During an average year the design temperatures given are not exceeded for more than 5 to 8% of the duration of a cooling season of 1200 hours in New York City in June, July, August and September.

276. Classification

A cooling load determination consists of five divisions which may be arranged as follows:

1. Normal heat transfer through walls, windows, partitions, doors, floors, ceilings etc. as given in the topics on heat transmission in Chapter V.
2. Heat emission of occupants within the enclosure as given in Chapter XI.
3. Transfer of solar radiation through walls, windows, doors, skylights or roof as given in this chapter.
4. Heat introduced by infiltration of outside air, or controlled ventilation, as given in this chapter.
5. Heat emission of mechanical, chemical gas, steam, hot water, electrical appliances and other sources of energy within enclosures as given in this chapter. Divisions 2, 4 and 5 may be further classified as latent or sensible heat.

277. Normal Heat Transmission

Normal heat transmission, as distinguished from solar heat transmission, means the transmission of heat through walls, windows, partitions and roofs by virtue of a difference in temperature between inside and outside.

278. Mathematical Analysis of Time Lag in Heat Transmission through Building Walls

A homogeneous substance, if left standing, soon reaches a state of thermal equilibrium. If the molecular motion is slow or heating effect slight, the substance will be in thermal equilibrium. If the motion is rapid or heating considerable the distribution of molecular velocity will become distinctly non-uniform and the substance is no longer in a state of equilibrium. The rate of heat transmission is proportional to the temperature difference only if the flow of heat is steady and the temperature invariable at any point in a plane perpendicular to the direction of flow. A different problem is presented when the temperature at each point varies with the time, as is the case with a building in which a constant temperature is to be maintained. This temperature is to be 80° F dry-bulb inside when the outside wall is subject to a temperature change from 85° F at 11 A.M. to 93° F at 3 P.M. and back to 85° F at 8 P.M., as well as to variable wind velocities, and solar radiation.

The quantity of heat per unit surface area varies. Part of the heat is used up in changing the temperature of the successive layers. In the transfer of a certain quantity of heat the mean temperature difference assumed is the logarithmic mean explained in Chapter IV. Fourier's Law states that in a state of heat equilibrium

$$Q = \frac{AKZ(T_a - t_a)}{L} \tag{1}$$

where

Q = quantity of heat, in Btu
A = area of heat transfer surface
K = heat transfer coefficient, Btu per hour per square foot per ° F temperature difference per inch thickness
Z = time, in hours
T_a = initial temperature of warmer air outside
t_a = initial temperature of cooler interior surface of building wall
L = thickness of layer perpendicular to A

When a building wall heats up in summer a state of equilibrium does not obtain. Therefore assume an average thickness of ½ L.

Then

$$Q = \frac{AKZ(T_a - t_a)}{\frac{L}{2}} \qquad (2)$$

and, further simplified,

$$Q = 0.215\ (T_a - t_a)\ A\ L \text{ or } D_M AL \qquad (3)$$

where

D_M is the average mean increase in temperature of 100 theoretical layers of wall

Q is the quantity of heat absorbed by wall to depth L

$0.215 = \frac{MTD}{GTD}$ which corresponds to $\frac{LTD}{GTD} = 0.01$ in Table 38

To obtain an expression of the time elapsed while Q penetrates through A for difference $(T_a - t_a)$:

$$Q = \frac{2AKZ(T_a - t_a)}{L} = 0.215(T_a - t_a)AL$$

and

$$Z = \frac{0.215L^2}{2K} = \frac{0.1075L^2}{K} \qquad (4)$$

and

$$L = \sqrt{\frac{KZ}{0.1075}} \qquad (5)$$

If the wall is thicker than L, only part of the wall is heated from the outside and

$$t_x = \frac{0.215L(T_a - t_a)}{R} \qquad (6)$$

where

R = thickness of wall

and t_x = change in temperature, in ° F, of mass of 1 sq ft of wall.

These formulas do not apply to air space because Fourier's Law states that the temperature is constant in a plane perpendicular to the heat flow and convection currents in an air space cause different temperatures in the plane.

Table 125 illustrates formula (4) concerning the time required for heat to penetrate an ideal wall of homogeneous material with various coefficients of heat transmission. Refer to Table 131 which gives values of time lag for non-homogeneous materials based on research of the A.S.H.V.E. Laboratories.

TABLE 125

TIME PERIOD FOR HEAT PENETRATION THROUGH HOMOGENEOUS MATERIAL

K	Btu Per Hour Per Square Foot Per Degree Difference Per Inch Thickness						
L	0 2	0.3	0.4	1.00	2.41	5.00	12.6
Inches	Z Hours to Penetrate L Inches of Material						
1	0.54	0.358	0.27	0.107	0.045	0.0216	
2	2.15	1.43	1.08	0.4[illegible]8	0.178	0.086	0.034
3	4.83	2.62	2.43	0.963	0.401	0.194	0.076
4	8.6	5.85	4.3	1.71	0.715	0.346	0.136
5	13.4	9.1	6.75	2.68	1.12	0.54	0.216
6	19.3	13.2	9.7	3 85	1.62	0.78	0.306
7	26.3	17.9	13.2	5.25	2.19	1.06	0.416
8	34.4	23.4	17.2	6.85	2.86	1.38	0.545
9	43.5	29.6	24.2	8.65	3.62	1.75	0.690

A theoretical analysis of conduction through ideal layers computed by Herbert Herkimer—1922.

279. Solar Heat Transmission

The atmosphere receives heat from two sources, terrestrial heat and solar heat, the combined action of which we measure when we obtain the temperature of the air. If there were no atmosphere surrounding the earth to check radiation of heat into space, the temperature of the earth would be $-325°$ F. Solar heat is radiated from the sun and is transmitted to the earth where it maintains all animal and vegetable life. Solar heat rays are transmitted radially from the sun and the intensity of solar radiation is proportional to the angle at which the rays strike the earth, dependent upon season and time of day.

Calculations of the solar heat transmitted through walls, roofs etc. are difficult to determine because of the periodic character of heat flow and the time lag caused by the heat capacity of the construction material of the wall or roof.

Rooms having walls, glass surface or roof exposed to the sun will receive a certain amount of additional heat over and above the heat normally transmitted. This additional heat is called sun effect or heat due to solar radiation, and may run as high as 75% of the cooling load. The greatest quantity of the heat from solar radiation is transmitted when the receiving surface is perpendicular to the rays of the sun and the heat transmitted to the room is almost immediate if it comes through glass but is somewhat retarded when coming through walls or roof. This retardation is called *time lag*. Inspection of the solar radiation table indicates that when the maximum effect occurs on the east or west side, it does not occur on the other sides or roof, but when it occurs on the south side it also occurs on the roof. From this may be deduced the rule that the sun effect to be added is the largest quantity obtained in any of the three combinations below.

(1) Sun effect on West wall alone
(2) Sun effect on South wall and roof
(3) Sun effect on East wall alone

280. Solar Heat Calculations

The solar heat conduction through a wall or roof exposed to the sun may be expressed by the following equation

$$H_R = AFaI$$

where

H_R = solar heat transmission in Btu per hour

F = percentage (expressed as a decimal) of the absorbed solar radiation which is transmitted to the inside

a = percentage (expressed by a decimal) of the incident solar radiation which is absorbed by the surface

I = intensity of solar radiation striking surface in Btu per hour per square foot

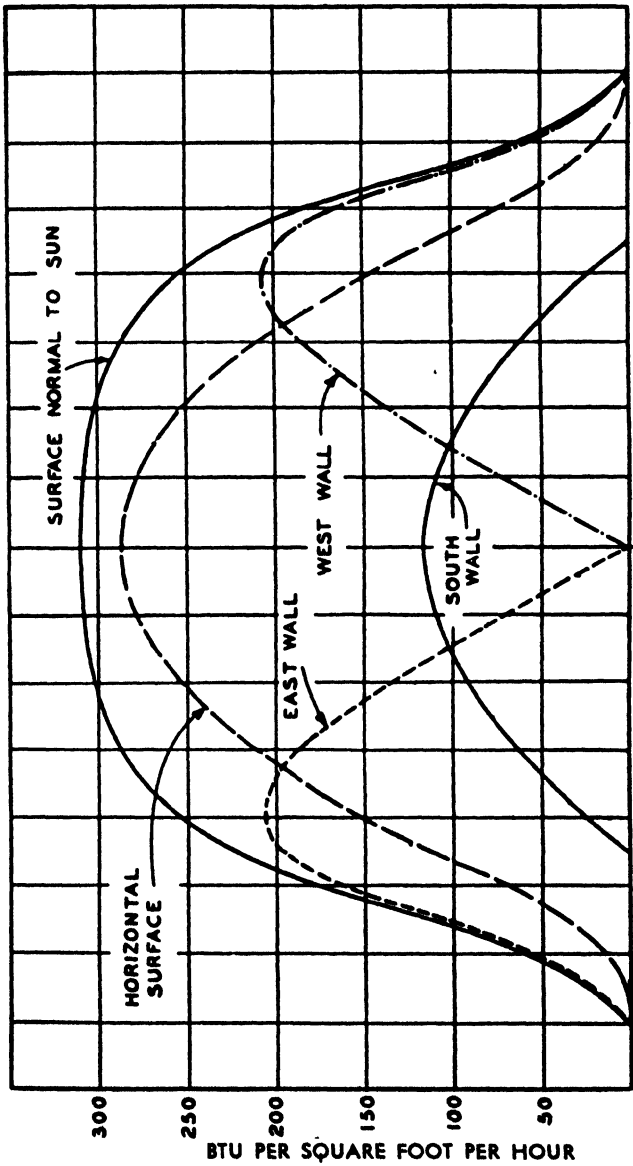


FIGURE 52

Curves Giving Solar Intensity Normal to Sun, on Horizontal Surface and on Walls for August 1

Reprinted from A.S.H.V.E. Guide

The variation in radiation intensity on surfaces facing different directions is shown in Figure 52. The greater part of the radiation is direct; however during the time the sun is shining, any surface, even a northern exposure, receives radiation of a lower intensity due to refraction and reflection. This scattered radiation varies from very small amounts to as high as 20%. The tables are for combined solar and scattered radiation.

A study of the chart will disclose a periodic relationship and wide variation in solar intensity. The values of solar intensity as given in Figure 52 must not be confused with the actual heat transmitted through the wall, for much of the radiation impinging on an outer wall, (93.5% when K = 0.33) does not pass through, but is reflected back to the outside air by radiation, conduction and convection. Table 127 gives only the solar heat transmitted and recognizes a time lag of two to three hours for the average brick veneer, tile and plaster wall having a heat transmission coefficient U of 0.33. It will be noted in Table 127a that the peak load on the roof occurs at 2 P.M. and in Table 127b that the peak load on a flat skylight occurs at noon, demonstrating a two hour lag. The maximum solar intensity strikes the west wall at 4 P.M. but the maximum interior temperature shown in the table is at 8 P.M., showing considerable time lag. However conditions are balanced by the fact that the temperature at 4 P.M. is 92° F while at 8 P.M. it is 85° F. Failure to consider this periodic property of heat flow resulting from the movement of the sun and the lag due to the heat capacity of the walls and roof, may result in a great error in load calculations.

281. Heat Transmission by Conduction (West only)

Table 126, based on a heat transmission factor (U) of 0.33 and a radiation factor of 0.065 gives the periodic heat flow by conduction only.

TABLE 126

SUN EFFECT DUE TO HEAT TRANSMISSION THROUGH WALLS

Sun time	11 A.M.	12 M.	1 P.M.	2 P.M.	3 P.M.	4 P.M.	8 P.M.
Outside temperature	85° F	86° F	90° F	93° F	93° F	92° F	85° F
Inside temperature	78° F	78° F	78° F	78° F	78° F	78° F	78° F
Difference	7° F	8° F	12° F	15° F	15° F	14° F	7° F
Btu per square feet	2.3	2.6	3.9	4.9	4.9	3.0	2.3
West Solar gain	1.6	1.7	6.1	9.8	12.7	13.7	0
Total heat gain	3.9	4.3	10.0	14.7	17.6	16.7	2.3

Peak load occurs at 3 P.M. with the type of wall assumed, and no time lag. With a time lag of 2 to 3 hours peak load would occur between 6 P.M. and 7 P.M. The total refrigeration load is based on the nature of occupancy; for a restaurant usually about 1 P.M., an office building about 11 A.M. and 2 P.M., a theater Sunday matinee about 2 P.M., a night club about midnight, etc.

If details of construction are not available use averages given in Table 126 in which 4.5 Btu was allowed for conduction through the average building wall and 5 Btu for sun effect on average building wall.

The calculation of heat transmission tables for walls and roofs does not take into consideration the heat capacity of the structure nor the consequent time lag in the transmission of heat. In the case of massive walls the time lag may amount to several hours. 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.

TABLE 127

SOLAR HEAT GAIN, BTU PER HOUR PER SQUARE FOOT OF AREA LATITUDE 40° N.—SUN DECLINATION 15°—APPROXIMATELY AUG. 1

MEDIUM DARK WALL AND ROOF (a)

Sun time	Average	11 A.M.	12 M.	1 P.M.	2 P.M.	3 P.M.	4 P.M.	8 P.M.
Outside T,°F	89.1°	85°	86°	90°	93°	93°	92°	85°
North	0.11	0.5	0.4	0.5	0.6	1.0	1.25	2.5
East	3.3	2.0	2.4	2.75	3.1	3.5	4.0	5.3
South	2.5	1.6	1.7	1.9	2.2	2.5	3.0	4.4
West	2.5	2.5	2.2	2.0	2.0	2.1	2.5	4.2
Roof	6.7	3.5	4.5	7.5	8.5	9.5	8.5	5.0

The above table shows the relation between time and heat flow through the inside surface at different orientations on a 93° design day. The wall selected has a heat transmission coefficient U of 0.33 Btu per hour per square foot per degree difference. Add the above values to the normal heat transmission to obtain the total transmission heat gain. For roof $U = 0.22$; temperature inside 78° F.

CLEAR GLASS—UNSHADED (b)

Sun time	Average	11 A.M.	12 M.	1 P.M.	2 P.M.	3 P.M.	4 P.M.	7 P.M.
Northeast	24	28	26	25.5	23.5	21	17	2.5
East	34.5	94	26	25.5	23.5	21	17	2.5
Southeast	57	144	98	41	23.5	21	17	2.5
South	97	124	128	124	103	74	29	2.5
Southwest	118	41	98	144	71	181	166	4.0
West	100	25.5	26	94	52	195	211	6.0
Northwest	62	25.5	26	28	54	102	137	5.0
Flat Skylight	245	281	290	281	244	212	160	5.0

Table (b) gives total solar plus sky intensity.

GLASS SHADED (c)

Multiply above values by following factors

Type	Finish Facing Sun	Factor
Canvas Awning	Plain	0.28
Canvas Awning	Aluminum	0.22
Inside Shade Fully Drawn	Aluminum	0.45
Inside Shade ½ Drawn	Buff	0.68
Inside Venetian Blinds—Slats—45°	Aluminum	0.58
Outside Venetian Blinds—Slats—45°	Aluminum	0.22
Glass Block—Smooth		0.16
Glass Block—Ribbed		0.15

TABLE 127 (*continued*)

LATITUDE FACTORS 40° = 1.00 AT PEAK LOADS (d)

Latitude, ° North	West Wall at 4 P.M.	Roof at 12 M.
30°	0.98	1.04
35°	0.98	1.03
45°	1.12	0.97

Table 127 is based on an average wall (U = 0.33). The following table offers a correction factor for other types of walls.

TABLE 128

SOLAR RADIATION FACTORS (F)

Factor F	0.0225	0.0425	0.065	0.0875	0.1075	0.14	0.16	0.18
U	0.1	0.2	0.3	0.4	0.5	0 6	0.7	0.8

Data from A.S.H.V.E. Research.

In Figure 52 at 10 A.M. 135 Btu per hour due to solar heat impinges on an east wall. From Table 128, if the wall has a heat transmission coefficient (U) of 0.3, then 135 × 0.065 = 8.7 Btu per hour transmitted during that period.

TABLE 129

SOLAR ABSORPTION COEFFICIENTS FOR DIFFERENT BUILDING MATERIALS

Surface Material		Absorption Coefficient (*a*)
Very Light Colored Surfaces	White stone Very light colored cement White or light cream-colored paint	0.4
Medium Dark Surfaces	Asbestos shingles Unpainted wood Brown stone Brick and red tile Dark-colored cement Stucco Red, green or gray paint	0.7
Very Dark Colored Surfaces	Slate roofing Tar roofing materials Very dark paints	0.9

Courtesy of A.S.H.V.E.

282. Solar Radiation through Glass

Windows present a problem somewhat different from that with opaque walls because they permit a large percentage of the solar energy to pass through. A small portion is reflected and some energy is absorbed by the glass. The quantity absorbed depends upon the character and thickness of the glass and the angle of incidence. The temperature of the glass is raised by the absorbed heat and this heat is then delivered to the air on each side in proportion to the temperature difference.

The A.S.H.V.E. tests [1] indicate that a single pane of double strength glass 0.127 in. thick absorbs approximately 11% 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.

The amount of solar radiation delivered to an unshaded glass surface may be obtained from Table 127b. These values must be used only for the net glass area on which the sun shines and not the entire glass area. Tests at the A.S.H.V.E. Research Laboratory have determined the percentage of heat from solar radiation actually delivered to a room with various types of outdoor and indoor shading. The data in Table 130 are taken from these tests.

The percentage values in this table were obtained by dividing

[1] A.S.H.V.E. Research Report No. 974—Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 93). A.S.H.V.E. Research Report No. 975—Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet, and J. L. Blackshaw (A.S.H.V.E. Transactions, Vol. 40, 1934, p. 101). A.S.H.V.E. Research Report No. 1180—Heat Gain Through Western Windows With and Without Shading, by F. C. Houghten and David Shore (A.S.H.V.E. Transactions, Vol. 47, 1941, p. 251).

TABLE 130

SOLAR RADIATION TRANSMITTED THROUGH SHADED WINDOWS

Type of Appurtenance	Finish Facing Sun	Per Cent Delivered to Room
Canvas awning	Plain	28
Canvas awning	Aluminum	22
Inside shade, fully drawn	Aluminum	45
Inside shade, one-half drawn	Buff	68
Inside Venetian blind, fully covering window, slats at 45°	Aluminum	58
Outside Venetian blind, fully covering window, slats at 45°	Aluminum	22

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 may be small or negative as the glass temperature is raised by the solar radiation absorbed.

In calculating the total heat gain through windows on the sunny side of buildings, it is sufficiently accurate to proceed as outlined herewith:

Consider the total heat gain as that resulting from solar radiation and neglect the heat transmission through the glass caused by the difference between the temperatures of the inside and outside air. This method should be used except at times when the calculated heat gain per square foot due to normal transmission exceeds the solar intensity. At such times, solar radiation may be neglected and the total heat gain considered as resulting from normal transmission.

Tests made at the A.S.H.V.E. Laboratory indicate that sunshine through window glass is the most important factor to contend with in cooling an office building, hotel or apartment. At times it was shown to account for 75% of the total cooling necessary.

TABLE 131

TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

Type and Thickness of Wall or Roof	Time Lag Hours
1 in. yellow pine horizontal roof, smooth black finish	1
2 in. yellow pine horizontal roof, smooth black finish	1¾
4 in. clay tile—horizontal roof—slag finish	2¼
2 in. gypsum horizontal roof—slag finish	2¼
4 in. gypsum horizontal roof—slag finish	4½
6 in. concrete horizontal roof—slag finish	5
1 in. concrete, 4 in. cinders, 1½ in. concrete, black tin	8
Wood siding, 1 in. sheathing 2 x 4 studs, lath and plaster	2
Wood siding, 1 in. sheathing 2 x 4 studs (loose fill)	5
4 in. brick, 1 in. sheathing, 2 x 4 studs lath and plaster	7
4 in. brick, 8 in. tile and plaster	10½
13 in. brick plastered	12
9 in. brick, 3¾ in. tile, 5¼ in. air space, 3¾ in. tile, plaster	16

Data from A.S.H.V.E. Guide.

283. Simple Method of Calculating Conduction and Solar Heat Gain

Careful consideration of all the complexities of the variable heat flow through building walls and roof would make the computations unduly long and tedious. Therefore, it is common practice to make approximations which simplify the problems and furnish reasonably accurate results. The following table is based on an average building having a heat transmission coefficient U of 0.33 for outside walls and 0.23 for its roof. Sometimes detailed construction of building walls is not available. In a situation of this kind an average wall and roof construction may be assumed. The glass area is generally 1/6 of the wall area and this ratio may be used in estimating glass area. It is important of course to investigate the exposures but if this information is not available the estimator should assume the maximum, which is usually the western exposure.

TABLE 132

APPROXIMATE CONDUCTION AND SOLAR HEAT GAIN

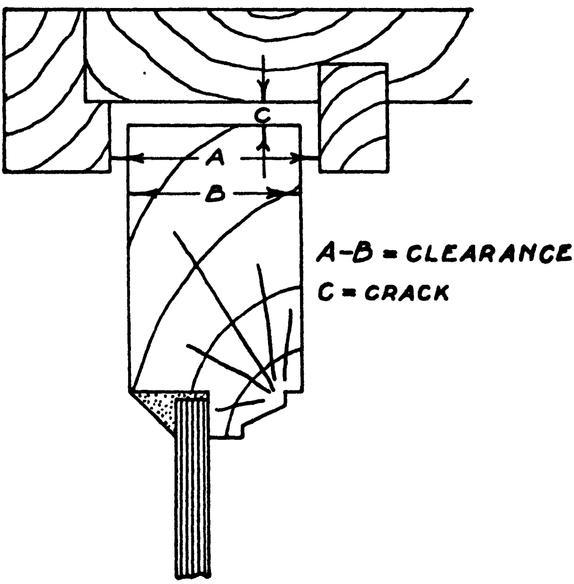
Conduction load.		
(Btu per square foot per hour per 15-degree difference)		
Walls (a) outside		4.95
Walls (b) inside		3.9
Windows and doors		17.0
Floor—uninsulated		3.5
Floor—insulated		1.7
Ceiling or roof (uninsulated)		3.6
Ceiling or roof (insulated)		1.8
Solar radiation		
Walls exposed to sun (uninsulated) additional		5
Walls exposed to sun (insulated) additional		2.5
Ceilings under roof or attic (uninsulated)		10.00
Ceilings under roof or attic (insulated)		5.00
Skylight		180
Window facing East	unshaded	55
Window facing Southeast	unshaded	75
Window facing South	unshaded	55
Window facing Southwest	unshaded	95
Window facing West	unshaded	105

With shaded glass use 1/2 of above values.

Note: If two adjacent or opposite walls are exposed to the sun, use the side with the maximum glass area.
Conditions assumed: outside 95° F Dry-bulb 75° F wet-bulb,
inside 80° F Dry-bulb 50% RH

284. Infiltration, all Seasons

Leakage of air through various apertures in buildings must be considered in heating and cooling calculations. This air leakage or infiltration is caused by cracks around doors, windows, through solid walls, fireplaces, chimneys, and by opening and shutting of doors. The displacement of heated air in buildings by unheated outside air is of two types: (1) wind pressure and (2) density difference of outside and inside air due to temperature difference. The former is generally referred to as infiltration and the latter as chimney or stack effect. The wind causes a pressure to be exerted on one side of a building. As a result air comes into the building on the side facing



DIAGRAM

Illustrating Crack and Clearance

TABLE 133

INFILTRATION THROUGH WALLS[a]

Expressed in cubic feet per square foot per hour

Type of Wall		Wind Velocity, Miles per Hour 5	10	15	20	25	30
8½ in. Brick Wall	Plain	1.75	4.20	7.85	12.2	18.6	22.9
	Plastered	0.017	0.037	0.066	0.107	0.161	0.236
13 in. Brick Wall	Plain	1.44	3.92	7.48	11.6	16.3	21.2
	Plastered	0.005	0.013	0.025	0.043	0.067	0.097
Frame Wall, with lath and plaster[b]		0.03	0.07	0.13	0.18	0.23	0.26

[a] The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms

[b] Wall construction: Bevel siding painted or cedar shingles, sheathing, building paper, wood lath and 3 coats gypsum plaster.

Courtesy of A.S.H.V.E.

the wind and a similar quantity of air leaves on the leeward side.

285. Window and Door Leakage

There are two methods of estimating air leakage through window and door cracks: (1) the crack method and (2) the air change method. The more rational crack method is generally recommended.

286. The Crack Method

This method is based on known air leakage factors for various types of windows and widths of crack and clearance. The wind velocity and length of crack are also considered when the crack method is employed. The infiltration for various types of windows is given in Table 134.

TABLE 34

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	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance. Includes wood frame leakage	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped	4.3	13.0	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance. Includes wood frame leakage	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows	Industrial pivoted, 1/16-in. crack	52	108	176	244	304	372
	Architectural projected, 1/32-in. crack	15	36	62	86	112	139
	Architectural projected, 3/64-in. crack	20	52	88	116	152	182
	Residential casement, 1/64-in. crack	6	18	33	47	60	74
	Residential casement, 1/32-in. crack	14	32	52	76	100	128
	Heavy casement section, projected, 1/64-in. crack	3	10	18	26	36	48
	Heavy casement section, projected, 1/32-in. crack	8	24	38	54	72	92
Hollow Metal, vertically pivoted window		30	88	145	186	221	242

Data from A.S.H.V.E. Research.

287. Door Leakage

Doors vary greatly in fit. The leakage for a poorly fitted double window may be used to express the leakage for a well-fitted door. If the door is poorly fitted, twice this value may be used. If it is weatherstripped the values may be reduced one-half. A single door in a store is allotted a value three times that for a well-fitted door. This extra allowance is for opening and closing losses and is kept to this value by the fact that doors are not used as much in the coldest and windiest weather.

288. Wind Pressure Closed Buildings

Flachsbar gives measurements of pressure distributions on simplified building models. They show positive pressure only on the windward perpendicular faces. Suction pressures prevail over the side, top and rear. This pressure differential results in a continuous change of air, and this is considered when calculating the heating load in winter or the cooling load in summer.

$V =$ miles per hour

$P =$ pressure, pounds per square foot

$P = V^2 \times 0.0029 =$ pounds per square foot

$P_1 = 0.00002\ V^2 =$ pounds per square inch

$P_2 = 0.00055\ V^2 =$ inches of water

TABLE 135

INFILTRATION THROUGH OUTSIDE DOORS FOR COOLING LOADS *

Expressed in Cubic Feet per Minute per Person Entering Room

Application	Cubic Feet Per Minute of Infiltration to be Assumed for Each Person (Patrons and Employees) in the Room	
	72 in. Revolving Door	36 in. Swinging Door
Bank	7.5	10.0
Barber Shop	3.5	4.5

* *Courtesy of A.S.H.V.E.*

TABLE 135 (*continued*)

Application	Cubic Feet Per Minute of Infiltration to be Assumed for Each Person (Patrons and Employees) in the Room — 72 in. Revolving Door	36 in. Swinging Door
Brokers' Office	5.0	6.5
Candy and Soda	5.0	6.5
Cigar Store	15.0	20.0
Department Store (Small)	5.0	6.5
Dress Shop	2.0	2.5
Drug Store	10.0	13.0
Furrier	2.0	2.0
Lunch Room	5.0	6.5
Men's Shop	3.5	4.5
Office (Professional)	2.5	3.0
Restaurant	2.0	2.5
Shoe Store	3.5	4.5

Infiltration When Doors Are Left Open Continuously

72 in. Revolving Door (Panels Open)	1200 cu ft per min
36 in. Swinging Door	800 cu ft per min

For vestibules with double pair swinging doors, infiltration may be assumed to be 75 per cent of swinging door values.

Infiltration for 72 in. revolving doors may be assumed 60 per cent of swinging door values.

Note: These values for swinging doors and for doors left open hold only where such doors are in one wall only, or where the doors in other walls are of the revolving type. If swinging doors are used for access, or doors are left open, in more than one wall, the infiltration cannot be estimated. The values for revolving doors hold regardless of number or location.

To determine the total cfm infiltration due to opening of doors, multiply the design number of occupants by the factor from the above table for the kind of establishment in question. When there is more than one door, treat them as though there were only one, except in case of open doors.

289. The Air Change Method

Air leakage is sometimes estimated by assuming a certain number of air changes per hour for each room, the number of changes being dependent upon the usage and location as indicated in the table below. This method may be used as a check on calculations made by the more exact crack method. The air change method is often used in preference to the crack method when estimating air leakage in vestibules.

TABLE 136

Exposure	Air Changes
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

Data from A.S.H.V.E. Guide

For simplicity, an arbitrary method of offsetting infiltration when cooling is accomplished by allowing one air change per hour, and when the quantity of outside air introduced is greater than one air change, no infiltration need be figured. If the outside air requirement is less than a one hour air change, an additional quantity of air should be added to allow for infiltration so that the total will be equivalent to a one hour air change. This minimum outside air quantity is required, in addition to any hood or vent exhaust from the conditioned space.

290. Heat Introduced by Outside Air

Allowance should be made for the moisture and heat introduced by the air for infiltration and ventilation. The volume of air entering by infiltration has been given in previous topics of this chapter. The volume of air required for ventilation is given in Chapter XI.

If the quantity of air entering an enclosure by infiltration exceeds that required for ventilation, the former should be used as a basis for determining the portion of the load contributed by outside air. When the volume of air required for ventilation exceeds that entering by infiltration it is assumed that a slight positive pressure will exist within the enclosure with a resultant exfiltration instead of infiltration. In this case, the air required for ventilation may be used in determining outside air load. However, if infiltration is a small percentage of the ventilating requirement, a safety factor is introduced by including both ventilating and infiltration air in the total cooling load.

291. Heat Emission from Appliances

Heat-generating appliances which give off sensible heat or sensible and latent heat in an air-conditioned enclosure may be divided into three general classes of equipment

(1) Electrical appliances
(2) Gas appliances
(3) Steam heating appliances.

In the first group may be found lights, fans, motors, toasters, waffle irons, etc. The wattage is generally marked on the name-plate of the appliance. To estimate heat emission in Btu per hour multiply total wattage by 3.413. Electric motors are rated in horsepower output. To determine the input some assumption of motor efficiency must be made, 75% being an average. Divide the horsepower rating of the motor by 0.75 and multiply by 2546 (Btu per horsepower-hour) to obtain output of motor in Btu. Good judgment must be employed in using data in Table 137. Only those appliances in use at the time of the peak load need be considered.

TABLE 137

HEAT GAINS FROM VARIOUS SOURCES

Source	HEAT Sensible	Latent	Total
Electric Heating Equipment[a]			
Electrical Equipment—Dry Heat	100%	0%	100%
Electric Oven—Baking	80%	20%	100%
Electric Equipment—HeatingWater—Stewing, Boiling, etc.	50%	50%	100%
Electric Lights and Appliances per Watt (Dry Heat)	3.4†	0	3.4†
Electric Lights and Appliances per Kilowatt (Dry Heat)	3,413†	0	3,413†
Electric Motors per Horsepower	2,546†	0	2,546†
Electric Toasters or Electric Griddles	90%	10%	100%
Coffee Urn—Large, 18 in. Diameter—Single Drum	2,000†	2,000†	4,000†
Coffee Urn—Small, 12 in. Diameter—Single Drum	1,200†	1,200†	2,400†
Coffee Urn—Approx. Connected Load per Gallon of Capacity	600†	600†	1,200†
Electric Range—Small Burner	*	*	3,400†
Electric Range—Large Burner	*	*	7,500†
Electric Range—Oven	8,000†	2,000†	10,000†
Electric Range—Warming Compartment	1,025†	0	1,025†
Steam Table—Per Square Foot of Top Surface	300	800	1,100†
Plate Warmer—Per Cubic Foot of Volume	850	0	850
Baker's Oven—Per Cubic Foot of Volume	3,200†	1,300†	4,500†
Frying Griddles—Per Square Foot of Top Surface	*	*	4,600†

[a] Heat gain from electric or gas residential ranges or cooking stoves depends on size of the family, socio-economic status of the individual, time of day for principal meal, and whether the equipment is manually or automatically controlled. Total heat gain will probably not exceed 40 per cent name-plate rating. Per cent sensible and latent heat will depend upon use of equipment; dry heat, baking or boiling.

* Percentage sensible and latent heat depends upon use of equipment; dry heat, baking or boiling. Above data based on continuous operation for one hour. Load factor may reach 75% in a counter lunch room at noon. When hoods are properly installed above appliances, 50% of the heat will be conveyed up the hood, which will be replaced by outside air with perhaps lower moisture content resulting in a net saving of refrigeration.

† Btu per hour.

TABLE 137 (*continued*)

Source	HEAT Sensible	Latent	Total
Hot Plates—Per Square Foot of Top Surface	*	*	9,000†
Hair Dryer in Beauty Parlor—600 w	2,050†	0	2,050†
Permanent Wave Machine in Beauty Parlor 24–25 w Units	2,050†	0	2,050†
Gas Burning Equipment [b]			
Gas Equipment—Dry Heat—No Water Evaporated	90%	10%	100%
Gas Heated Oven—Baking	67%	33%	100%
Gas Equipment—Heating Water—Stewing, Boiling, etc.	50%	50%	100%
Stove, Domestic Type—No Water Evaporated—Per Medium Size Burner	9,000†	1,000†	10,000†
Gas Heated Oven—Domestic Type	12,000†	6,000†	18,000†
Stove, Domestic Type—Heating Water—Per Medium Size Burner	5,000†	5,000†	10,000†
Residence Gas Range—Giant Burner (About 5½ in. Diameter)	*	*	12,000†
Residence Gas Range—Medium Burner (About 4 in. Diameter)	*	*	10,000†
Residence Gas Range—Double Oven (Total size 18 in. x 18 in. x 22 in. High)	*	*	18,000†
Residence Gas Range—Pilot	*	*	250†
Restaurant Range—4 Burners and Oven	*	*	10,000†
Cast-Iron Burner—Low Flame—Per Hole	*	*	100†
Cast-Iron Burner—High Flame—Per Hole	*	*	250†
Simmering Burner	*	*	2,500†
Coffee Urn—Large, 18 in. in Diameter—Single Drum	5,000†	5,000†	10,000†
Coffee Urn—Small, 12 in. Diameter—Single Drum	3,000†	3,000†	6,000†
Coffee Urn—Per Gallon of Rated Capacity	500†	500†	1,000†
Egg Boiler—Per Egg Compartment	2,500†	2,500†	5,000†
Steam Table or Serving Table—Per Square Foot of Top Surface	400†	900†	1,300†
Dish Warmer—Per Square Foot of Shelf	540†	60†	600†
Cigar Lighter—Continuous Flame Type	2,250†	250†	2,500†
Curling Iron Heater	2,250†	250†	2,500†

[b] Name-plate ratings of gas burning equipment can be obtained from a Directory of Approved Gas Appliances and Listed Accessories, January 1, 1942, obtainable from *American Gas Association* Laboratories, Cleveland, Ohio.

TABLE 137 (*continued*)

Source	HEAT Sensible	HEAT Latent	HEAT Total
Bunsen Type Burner—Large—Natural Gas	*	*	5,000†
Bunsen Type Burner—Large—Artificial Gas	*	*	3,000†
Bunsen Type Burner—Small—Natural Gas	*	*	3,000†
Bunsen Type Burner—Small—Artificial Gas (Idling)	*	*	1,800†
Welsbach Burner—Natural Gas	*	*	3,000†
Welsbach Burner—Artificial Gas	*	*	1,800†
Fish-tail Burner—Natural Gas	*	*	5,000†
Fish-tail Burner—Artificial Gas	*	*	3,000†
Lighting Fixture Outlet—Large, 3 Mantle 480 C.P.	4,500†	500†	5,000†
Lighting Fixture Outlet—Small, 1 Mantle 100 C.P.	2,250†	250†	2,500†
One Cubic Foot of Natural Gas Generates	900†	100†	1,000†
One Cubic Foot of Artificial Gas Generates	540†	60†	600†
One Cubic Foot of Producer Gas Generates	135†	15†	150†
Steam Heated Equipment[c]			
Steam Heated Surface Not Polished—Per Square Foot of Surface	330†	0	330†
Steam Heated Surface Polished—Per Square Foot of Surface	130†	0	130†
Insulated Surface, Per Square Foot	80†	0	80†
Bare Pipes Not Polished, Per Square Foot of Surface	400†	0	400†
Bare Pipes Polished, Per Square Foot of Surface	220†	0	220†
Insulated Pipes, Per Square Foot	110†	0	110†
Coffee Urn—Large, 18 in. Diameter—Single Drum	2,000†	2,000†	4,000†
Coffee Urn—Small, 12 in. Diameter—Single Drum	1,200†	1,200†	2,400†
Egg Boiler—Per Egg Compartment	2,500†	2,500†	5,000†
Steam Table, Per Square Foot of Top Surface	300†	800†	1,100†
Miscellaneous			
Heat Liberated by Food per Person, as in a Restaurant	30†	30†	60†
Heat Liberated from Hot Water used direct and on towels per hour—Barber Shops	100†	200†	300†

[c] Steam Requirements of Process Equipment, Report of the Commercial Relations Committee, *National District Heating Association* (*Heating, Piping and Air Conditioning*, November, 1942, p. 675).

292. Food Heat Gains

An allowance of 30 Btu per person in restaurants is recommended to account for sensible heat gain from food. However when the appliance heat is already included in the computations, as in a cafeteria or lunch-counter restaurant, this allowance is not considered.

293. Heat Gain from Illumination

Lighting is a great source of heat in modern stores and offices. The Table 138 gives watts used per square foot. Multiply watts by 3.413 to convert to Btu per hour.

TABLE 138

GROUP OF INTERIORS WITH RECOMMENDED MINIMUM STANDARDS OF ILLUMINATION AND THE APPROXIMATE LIGHTING REQUIREMENT IN WATTS PER SQUARE FOOT OF FLOOR AREA

	Recommended Foot-candles	Electrical Demand Watts Per Square Foot
Auditorium	5	1.5
Automobile Show Room	20	4.0
Banks		
Lobby	10	2.5
Counters (75 w per running foot)	50–100	—
Barber Shop and Beauty Parlor	20	4.0
Bowling		
Alley Runway and Seats	10	2.5
Pins (300 w per set)	30–50	—
Billiards		
General	10	2.5
Tables (450 w per table)	30–50	—

TABLE 138 (*continued*)

	Recommended Foot-candles	Electrical Demand Watts Per Square Foot
Club and Lodge Rooms		
Lounge and Reading	20	4.0
Auditorium	5	1.5
Court Rooms	10	2.5
Dance Halls (with no color or spectacular effects)	5	1.5
Drafting Rooms	30	7.0
Hotels		
Lobby (with no provision for convention exhibits)	10	2.5
Dining Room	5	2.5
Kitchen	10	2.5
Bedrooms (including allowance for table and bed lamps)	10	3.0
Corridors (10 w per running foot)	2	—
Writing Room (including allowance for portable lamps)	20	4.0
Library		
Reading Rooms (including allowance for table lamps)	20	6.0
Stack Room (12 w per running foot of facing stacks)	10	—
Moving Picture Theaters (with no color or spectacular effects)		
During Intermission	5	1.5
During Pictures	0.1	0.5
Museum		
General	10	2.5
Special Exhibits (an average of 2.5 w per square foot)	50–100	—
Office Buildings		
Private and General Office		
No close work	10	3.0
Close work	20	5.0
Conference Rooms	10	3.0
Filing and Mail Sorting	20	5.0
Reception Rooms	10	3.0
Corridors and Stairways	5	1.5
Post Office		
Lobby	10	3.0

TABLE 138 (*continued*)

	Recommended Foot-candles	Electrical Demand Watts Per Square Foot
Sorting, Mailing, etc.	20	4.0
Office, Private and General	20	5.0
File Room and Vault	10	2.5
Corridors and Stairways	5	1.5
Professional Offices		
Waiting Rooms	10	2.5
Consultation Rooms	20	5.0
Operating Rooms	20	5.0
Dental Chairs (600 w per chair)	30–50	
Restaurants, Lunch Rooms and Cafeterias		
Dining Area	10	2.5
Food Displays (100 w per running foot of counter)	30–50	—
Stores		
Main business districts—large cities	20	5.0
Neighborhood stores—large cities and all stores—small cities and towns	15	3.5
Show Cases		
40 w per running foot	50–100	—
Show Windows		
Brightly Lighted Districts, large cities 300 w per running foot of glass	200	—
Secondary business districts, large cities 150 w per running foot of glass	100	—
Neighborhood stores, large cities and for small cities and towns—75 w per running foot of glass	50	—
Lighting to reduce daylight reflections—500 w per running foot of glass	400	—
Theaters		
Auditoriums (with no color or spectacular effects)	5	1.5
Foyer	10	2.5
Lobby	15	3.5

Courtesy of A.S.H.V.E.

294. Estimating Forms

Every company in the air-conditioning industry has an estimating system of some kind. To assist the estimator in calcu-

TABLE 139

HEAT LOSSES FROM BARE AND INSULATED PIPE

Based on Still Air and Room Temperature of 90° F—Heat Losses Given in Btu Per Hour Per Linear Foot of Pipe—Coefficient "U" = 0.55 Btu Per Hour Per Square Foot Insulation Interior Surface Per Degree Difference vs. Steam and Outside Air for Thickness Required Which Varies from Standard 85% Magnesia to 4½ in. Special

Steam Temperature °F	Surface Temperature °F	Surface	Nominal Pipe Size ½	¾	1	1¼	1½	2	2½	3	3½	4	4½	5	6	8
270	125	Bare	102	127	160	202	231	288	348	425	485	546	607	675	804	1047
		Insulated	33	37	43	50	55	59	67	78	86	90	98	107	123	144
340	130	Bare	162	202	254	321	367	459	554	676	772	869	965	1073	1278	1665
		Insulated	47	53	61	71	78	84	96	113	123	127	139	152	176	204
390	135	Bare	215	269	336	425	487	608	735	897	1024	1152	1279	1423	1695	2207
		Insulated	57	65	74	87	95	84	95	109	119	110	118	130	147	180
550	140	Bare	462	577	722	913	1045	1305	1580	1924	2197	2471	2746	3055	3638	4735
		Insulated	75	84	95	108	118	126	131	149	163	165	179	194	220	250
800	145	Bare	1077	1347	1685	2131	2439	3047	3689	4490	5130	5773	6412	7132	8494	11050
		Insulated	120	134	149	169	183	155	171	185	207	215	219	228	258	307
1000	150	Bare	1760	2200	2750	3480	3990	4975	6020	7350	8370	9425	10450	11650	13880	18000
		Insulated	144	154	1157	162	164	151	148	146	151	148	145	143	143	146

Per Cent Increase in Heat Loss

Steam Temperature °F	Room Temperature 60°	70°	80°	90°
270	16.7	11	5	0
340	12	8	4	0
390	10	6.6	3.5	0
550	6.5	4.3	2.2	0
800	4.2	2.8	1.4	0
1000	3.3	2.2	1.1	0

Per Cent Heat Loss for Bare Pipe

Air Velocity Feet Per Minute	300	500	900
Surface Temperature	Per Cent Increase in Loss		
200	1.55	1.42	2.1
600	1.38	1.55	1.75
1000	1.22	1.85	1.4

Condensation: Btu loss does not change temperature but causes condensation of steam. Allow average of one lb condensation per 900 Btu.

TABLE 140

ROOM HEAT LOAD RATIOS FOR TYPICAL SUMMER COMFORT CONDITIONING

Room Heat Load Ratios [a]	Typical Classes of Room Service or Load: No. Occupants or Sources of Vapor	Private Office or Residence	Restaurant or Crowded Office	Auditorium at Capacity or Crowded Restaurant	Ballroom at Capacity
$\frac{\text{Sensible Heat}}{\text{Total Heat}}$	1.00	0.90	0.80	0.70	0.60
$\frac{\text{Total Heat}}{\text{Sensible Heat}}$	1.00	1.11	1.25	1.43	1.67
$\frac{\text{Latent Heat}}{\text{Total Heat}}$	0	0.10	0.20	0.30	0.40
$\frac{\text{Total Heat}}{\text{Latent Heat}}$	....	10.00	5.00	3.33	2.50
$\frac{\text{Sensible Heat}}{\text{Latent Heat}}$	....	9.00	4.00	2.33	1.50
$\frac{\text{Latent Heat}}{\text{Sensible Heat}}$	0	0.11	0.25	0.43	0.67
Dry-bulb Temperature of Air at Room Inlets, to Maintain Typical Room Conditions of 80° F Dry-bulb, 50% Relative Humidity					
Air entering saturated [b]	60.0	58.6	56.5	53.0	35.0
Air entering with 4° F wet-bulb depression	66.5	65.4	64.1	61.8	56.0
Air entering with 8° F wet-bulb depression	72.6	72.1	71.6	70.5	68.0

[a] The over-all heat load ratio for the dehumidifier will be different from the heat load ratio for the room. The extent of the difference will depend on the quantity and condition of the outside air used, upon the magnitude of the duct losses, and upon whether or not reheat or by-pass are used.

[b] Typical air conditions leaving the central conditioner are: With spray dehumidifier, 0 to 2° F wet-bulb depression. With surface-type dehumidifier, 1 to 6° F wet-bulb depression. With by-pass or reheat, 4 to 10° F wet-bulb depression.

Courtesy of A.S.H.V.E.

lating the cooling and heating load, an estimating form, prepared by the engineer, is essential and convenient. No single type of estimating form is recommended, since the details of the form must be arranged to meet the conditions of trade and the engineering ability of the estimator. The form in Figure 53, prepared by the authors, is offered only as a suggestion.

COOLING LOAD ESTIMATE SHEET

Estimated by ________ Date ________ Plan No. ________ Sheet No. ________

Job Name ________ Location ________

Space Conditioned ________ Length ________ Width ________ Height ________ Floor ________

Outside Design Conditions: Dry-Bulb ________ Wet-Bulb ________ Dew Point ________ Rel. Hum. ________

Inside Design Conditions: Dry-Bulb ________ Wet-Bulb ________ Dew Point ________ Rel. Hum. ________

Time ________ A.M. ________ P.M. ________ Window Shades ________ Awnings ________ Bare ________

Refer To	Item No.	Item Description	Dimensions	Area, Sq Ft	Factor U	Temp. Diff.	Sensible Heat Btu per Hour
Topic 78		Transmission Heat Gains			....	....	....
Topic 82	1	Exterior Walls (Gross)					
Table 50	2	Exterior Glass					
	3	Exterior Walls (Net)					
Table 49	4	Floor					
Table 49	5	Ceiling or Roof					
Table 50	6	Partitions (Net)					
Table 50	7	Glass in Partitions					
	8	Miscellaneous					
	9	Total Transmission Heat Gains	....	....	....	....	
Table 132		Excess Sunlight Gains					
		Walls Facing:					
Table 132	10	W ☐ E ☐ S ☐					
Table 132	11	W ☐ E ☐ S ☐					

FIGURE 53

Refer To	Item No.	Item Description	Dimensions	Area, Sq Ft	Factor U	Temp. Diff.	Sensible Heat Btu per Hour
Table 132	12	Roof					
		Glass Facing:					
Table 132	13	W ☐ E ☐ S ☐					
Table 132	14	W ☐ E ☐ S ☐					
Table 132	15	Skylights					
Table 128	16	Miscellaneous					
	17	Total Excess Sunlight Gains					
		Duct Heat Gains					
Table 55	18	Insulated Ducts					
Table 54	19	Non-Insulated Ducts					
	20	Total Duct Heat Gains	....	....	....	....	

Refer To	Item No.	Item Description	Heat Load Btu per Hour: Sensible	Heat Load Btu per Hour: Latent
Table 109		Body Heat Gains		
	21	Sensible No. of People ——— × 225		
	22	Latent (Quiet) No. of People ——— × 175		
	23	Latent (Active) No. of People ——— × 450		
	24	Total Body Heat Gains		
Table 137		Equipment Heat Gains		
Table 138	25	Electric Lights ———Watts × 3.415		
	26	Electric Motors ———H.P. × ——		
	27	Electric Equipment ———No. × ——		
	28	Gas Equipment ———No. × ——		

FIGURE 53 (*continued*)

Refer To			Heat Load Btu per Hour Sensible	Latent
Table 139	29	Warm Pipes ________ Ft. × ____		
	30	Total Equipment Heat Gains		
Topic 284		Infiltration Heat Gains Sensible Factor Latent Factor	....	....
Table 136	31	Room Volume Cubic Feet		
	32	Miscellaneous Heat Gains (Pump and Fan H.P., Etc.)		
Topic 291		Summary of Heat Gains		
		Item No. / Item		
Topic 78		9 — Transmission Load		
Topic 279		17 — Sun Load		
Topic 87		20 — Duct Load		
Topic 195		24 — Occupancy Load		
Topic 291		30 — Equipment Load		
Topic 284		31 — Infiltration		
		32 — Miscellaneous		
	33	Total Sensible Heat Gains		....
	34	Total Latent Heat Gains		←
	35	Total Heat Gains = Item 33 + Item 34		....
Table 140	36	Sensible Heat Percentage = $\frac{\text{Item } 33}{\text{Item } 35} \times 100 =$ ________ $\times 100 =$ %		
Table 110	37	Dry-Bulb Temperature of Air Supply = ________ Degrees F		
Topic 301	38	Wet-Bulb Temperature of Air Supply = ________ Degrees F		
	39	Rise in Dry-Bulb Temperature of Air Supply = Room Dry-Bulb — Item 37 = ____ — ____ = ____ Deg.		

FIGURE 53 (*continued*)

Refer To	Item No.	Item Description	
Topic 270	40	Total Air Supply $= \dfrac{\text{Item 35}}{1.08 \times \text{Item 39}} = \dfrac{\rule{2cm}{0.4pt}}{1.08 \times \rule{1cm}{0.4pt}} =$ ______ C.F.M.	
		Heat Load of Ventilating Air	
		No. of People ______ $\times$ ______ C.F.M. Per Person = ______ C.F.M. of Outside Air	
	41	C.F.M. Outside Air ______ $\times$ ______ Btu Per C.F.M. $\times$ 60 = ______ Btu Per Hour	
		Total Cooling Load on Coils and Refrigerating Apparatus	
		Item 35	
		Item 41	
	42	Total Cooling Load Btu Per Hour	
Topic 36	43	Tonnage Equivalent of Cooling Load $= \dfrac{\text{Item 42}}{12000} =$ Tons	

FIGURE 53 (*continued*)

CHAPTER XV

DEHUMIDIFICATION AND HUMIDIFICATION WITH AIR WASHERS

Dehumidifying is the process of removing moisture from the air or reducing the quantity of water vapor present in a pound of air. It will be seen that this is a *specific* or *absolute* reduction in the moisture content and has nothing whatever to do with the matter of relative humidity, although the relative humidity will be affected by dehumidifying if the temperature of the air remains unchanged. In the ordinary case, however, the temperature of the air is changed and the relative humidity is altered by this change as well. It is entirely possible, for example, to have air at a given temperature and 50% relative humidity, to remove some of the moisture and still have the air at a higher relative humidity, or even saturated, due to the drop in the air temperature which may occur at the same time.

295. Means of Dehumidifying

Dehumidifying may be accomplished by physical means, by lowering the air temperature below the dew point of the air or by chemical means, by employing a chemical such as silica gel or an aluminum salt to absorb the moisture from the air. In lowering the air temperature, cooling must be employed, while the use of chemicals requires no cooling to accomplish dehumidification.

Dehumidifying by means of chemicals usually is carried out in a three-stage cycle: first, the stage when the moisture is absorbed from the air; second, the stage when this moisture is driven out of the chemical and the chemical is brought back to its original dry state; and, third, the stage when the chemical is allowed to cool after the drying process (which usually employs heat). The chemicals used have microscopic pores in which the moisture collects, sometimes to the extent of several times the weight of the chemical; but in time they become saturated and can absorb no more moisture. At this point the chemicals must be regenerated. This process of dehumidifying is called the *adsorption system.*

296. Dehumidifying by Cooling the Air

The common dehumidifying process used in air conditioning consists of lowering the air temperature sufficiently below the dew point that the air at the lower temperature will be saturated with the exact quantity of moisture that is desired when the air is heated back to the room temperature which is to be maintained.

297. Equipment Used for Cooling Air

There are two types of devices commonly employed for cooling air in the ordinary air-conditioning system: the older and more exact type, which consists of an air washer, and the newer and cheaper type consisting of coils. In the air washer, water at the required low temperature is sprayed into the air stream and cools the air; such spray water does not evaporate and if the air is cooled below the dew point, as it nearly always is, water is actually added to the sprays by the condensation of water out of the air. In the coil-type coolers, the air is drawn across a cooling surface, which may be filled with cooled water or a refrigerant, and the air, coming into contact with the cold surface of the coils is cooled; if it is cooled below the dew point it deposits moisture on the coils, from which it drips down into a pan piped to the sewer or other means of disposal.

298. Construction of the Air Washer

The air washer is a device originally designed for cleansing the air, as its name implies. It washes the air by spraying into it atomized water which wets the dust particles and causes them to fall to the pan at the bottom, but it will not remove oily soot or dust and it does not remove smoke to any extent. It usually consists of a pan about 16 in. deep, on top of which a housing is constructed to confine the air, and the inlet and outlet ends of the housing are left open for the entrance and exit of the air. At the entrance a perforated plate often is used to distribute the air flow over the entire area of the washer, while at the exit baffle plates or *eliminators* are inserted to prevent the carry-over of entrained moisture into the system with the air. In the central portion atomizer nozzles on *trees* are so arranged that spray water is distributed over the entire cross-section of the washer. In air-conditioning dehumidifying, it is essential to have a double row of sprays or *trees* so that cooling can be more thorough. Air washers commonly are made in the sizes given in Tables 141 and 142.

In the ordinary air washer a centrifugal pump is set beside the washer; this pump takes its suction from the pan at the bottom of the washer and delivers the water to the spray nozzles where it is sprayed into the air and falls to the pan at the bottom, ready to repeat the cycle. In cooling with air washers the water is pumped through the nozzles by a similar pump, but also passes through a cooler, in which the heat, collected from the air passing through the washer, is removed from the water and the water temperature brought back to its original status.

299. Air Washer Performance

The air washer is rated on a velocity of 500 fpm through the washer for the air flow and there should be 5 nozzles per

TABLE 141

TYPICAL SPECIFICATIONS OF ONE-BANK WASHERS

Number	Air Capacity, Cubic Feet per Minute	Chamber Area, Square Feet	Dimensions, in. W	Dimensions, in. H	Dimensions, in. L	Washing Surface, Square Feet	Number of Nozzles Spray	Number of Nozzles Flooding	Water Flow, Gallons per Minute Spray	Water Flow, Gallons per Minute Flooding	Pump Size, Inches	Motor h.p.
1	4,600	9.2	33	61½	86 5/16	84.5	12	9	22	9	1½	1½
2	9,400	18.8	64½	61½	86 5/16	174	24	20	43	20	1½	2
3	14,200	28.3	95¾	61½	86 5/16	263	36	30	65	30	1½	3
4	21,900	43.7	95¾	61½	86 5/16	413	60	30	108	30	1½	3
5	29,300	58.6	127¼	61½	86 5/16	553	80	40	144	40	3	5
6	36,700	73.4	158½	61½	86 5/16	693	100	51	180	51	1½	5
7	44,100	88.2	190	61½	86 5/16	852	120	61	216	61	1½	5

TABLE 142

TYPICAL SPECIFICATIONS OF TWO-BANK WASHERS

Nominal Size, Feet	Air Capacity at 500 Ft. per Min., Cubic Feet per Minute	Area of Spray Chamber, Square Feet	Dimensions, W	Dimensions, H	Elimination Surface, Square Feet	Spray Nozzles Number	Spray Nozzles Gallons per Minute	Flooding Nozzles Number	Flooding Nozzles Gallons per Minute
5 × 4	10,000	20	5′3″	5′1″	244	80	9	18	98
5 × 6	15,000	30	5′3″	7′1″	366	120	9	18	138
5 × 8	20,000	40	5′3″	9′1″	488	160	9	18	178
5 × 9	22,500	45	5′3″	10′1″	549	180	9	18	198
6 × 5	15,000	30	6′3″	6′1″	366	120	11	22	142
6 × 7	21,000	42	6′3″	8′1″	512	168	11	22	190
6 × 9	27,000	54	6′3″	10′1″	658	216	11	22	238

thousand cfm handled; each nozzle will require a supply of 1.8 gpm of water and, in addition, eliminator nozzles for washing down the eliminator plates must be used, these requiring 1 gpm per nozzle. A ball cock is used to keep the pan full of water and an overflow allows any excess water to escape during dehumidifying. Inside the washer a marine light allows a view of the sprays and their operation. The washer may be used for humidifying as well as dehumidifying but generally employs heated water when humidifying. During humidifying the air will leave the washer at not over two degrees above the water temperature and practically saturated at that temperature.

When cooling without refrigeration, the air washer with only the flooding nozzles on the eliminators in operation and using *recirculated uncooled* water will reduce the air temperature somewhat without a noticeable increase of humidity. When the flooding nozzles and one bank of sprays are in operation the air will be cooled 70% of the wet-bulb depression and there will be an increase of humidity; when both banks of sprays are in use the air will be cooled 99% of the wet-bulb depression, but will be practically saturated as it leaves the washer.

When cooled water is supplied to the washer the entire situation changes, for, in this case, with both sprays in operation, the air will leave the washer at the water temperature *minus 10% (or slightly less) of the difference* between the temperature of the entering air and the washer. In the first case, the wet-bulb temperature of the entering air is really the limiting factor in determining how much the air can be cooled when recirculated and uncooled water is used, while, in the second case, when cooled water is used, the air can be cooled almost to the water temperature. For example, a washer using recirculated and uncooled water with all sprays in operation would be able to cool air at 95° F with a wet-bulb of 80° F

$$99\% \text{ of } (95° - 80°) \text{ or } 14.85°,$$

but with cooled water at 50° F and under the same operation of both sprays the temperature of the air would be changed

$$[(95° - 50°) \times 10\% = 4.5°,]$$

so that the air temperature would be less than 4.5° above 50° or possibly 54°.

300. Refrigeration Equivalent for Dehumidification

The condensing of water vapor in the air results in a release of latent heat caused by the changing of the water from a gaseous, or vapor, form into a liquid form and this latent heat under atmospheric pressure amounts to 970.4 Btu per pound. Owing to the fact that the condensate, too, is reduced in temperature after it is formed, it is customary in air-conditioning calculations to consider the heat released in dehumidifying as 1,000 Btu per pound, for every pound of vapor so condensed.

With chemical dehumidifying no such refrigeration load is created, as refrigeration is not employed in this process; the removal of the condensed moisture, after the dehumidifying agent has become saturated, is the difficulty and heat is usually employed for this purpose; the agent is heated until the absorbed moisture is driven off and then must be allowed to cool before it can be again introduced into the air flow. One arrangement often used is a cylindrical shell in which three equal compartments are constructed, each of which is filled with the agent to be used for dehumidifying. One of these compartments is inserted into the air flow, while the second is being heated and dried and the third compartment is cooling, preparatory to being inserted into the air flow as soon as the material in the first compartment is saturated. When the material in the first compartment is saturated, the cylinder is given ⅓ of a complete revolution, thus bringing the saturated material out of the air flow and into the dryer, while the cooled material is inserted in the air flow and the material previously in the dryer is turned to the cooling position.

301. Cooling and Dehumidification

It has been seen that in cooling the air for the purpose of dehumidifying, it may be necessary to cool the air below the temperature at which it is desired to inject the air into the room; for example, if the room is to be kept at a maximum of 80° F dry-bulb and 50% relative humidity the number of grains per pound of air in the room will approximate

$$155 \text{ gr} \times 0.50 \text{ or } 77.5 \text{ gr}$$

and air, in order to contain not over 77.5 grains saturated would have to be reduced in temperature to at least 60° F because at 60° F saturated air will contain 77.3 grains per pound. However, on days when the room is to be kept at 70° F and 50 per cent relative humidity, the number of grains per pound of air in the room must not exceed

$$110 \text{ gr} \times 0.50 \text{ or } 55 \text{ gr}$$

and the air would have to be reduced to 52° F because saturated air at about 52° F will contain the quantity of moisture which is desired.

If it is not desirable to introduce the air into the room at more than 15 degrees below the room temperature, it will be seen that, in the first case, the air must not come into the room at less than

$$80° \text{ F} - 15° \text{ F or } 65° \text{ F}$$

while, in the second case, it could not come in over

$$70° \text{ F} - 15° \text{ F or } 55° \text{ F}$$

This indicates, that, in both cases, the air after dehumidification would be too cool to deliver into the room. In the first case, the dehumidified air reaches 60° F when air at 65° F must be delivered to the room, and in the second case, the air reaches 52° F in order to dehumidify, and must not be injected into the room at less than 55° F. These conditions are made worse when there are people in the room adding moisture to the air, since dehumidification then must be carried to a greater extreme and the air cooled to a lower temperature in order to allow for the moisture increase in the room and still not carry the room humidity higher than desired.

As an example of this, let it be assumed that a room is to be kept at 80° F dry-bulb with 50% relative humidity, and the number of persons in the room is such as to increase the moisture content in the air 10 gr per lb while the air is passing through the room. To what temperature must the air be reduced in order to obtain 50% relative humidity in the room? In such cases the room humidity in grains per pound does not determine the number of grains which must be in the supply air, since the supply air must be able to pick up 10 grains in the room and then, after picking up this 10 grains, to have the required 77.5 gr per lb. As a result, the entering air must be 10 gr per lb short of the number of grains desired in the room so that, after picking up the 10 grains in the room, the air will have the desired 77.5 grains. This means that the number of grains per pound in the entering air must be 67.5 gr and as air saturated at 56° F will hold approximately this number of grains, it is evident that the air, in order to be properly dehumidified, must be reduced to 56° F even when it cannot enter the room at a temperature under 65° F.

When all of the air does not pass through the dehumidifier and part is by-passed around the air washer the temperatures, to which the portion of the air that is dehumidified must be reduced, are still lower and often reach 45° F. As a result of the necessity of reducing the air temperatures so low in order to remove sufficient moisture, the air must be re-heated after it has been dehumidified, either by inserting heaters in the air flow, or by by-passing part of the air around the dehumidifying apparatus so that this uncooled air may be used to reheat the dehumidified air and thus render heaters unnecessary. It is always possible to find a point where the quantity of untreated and uncooled air used for by-passing, when mixed with the dehumidified air, will yield the temperature and relative humidity desired in the room air and finding such a combination is one of the finer points in air-conditioning design.

302. Operation of the Air Washer in Humidifying

When the washer is humidifying, the evaporation of water from the sprays into the air is accompanied by a reduction of the air temperature because the latent heat of evaporation is largely supplied from the air passing through. In order to secure accurate humidifying the temperature is raised by

A. Heating the spray water.
B. Heating the incoming air.
C. A combination of A and B.

A small steam heater is used to raise the temperature of the water spray and to heat the air the preheater has a second row added so that the air temperature may be made as high as desired.

303. Spray Heads and Spray Water

The spray heads are specially designed to prevent clogging and to atomize the water finely, and require about 20 lb pressure to function properly. The heads are supported on vertical pipes, or *trees,* which are fed from horizontal headers at the bottom, to which the discharge from the circulation pump is connected.

304. Efficiency of Air Washers

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% unless the concentration of dust is high. Its effectiveness in removing greasy microscopic dust is negligible as is 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 to note here that heating is generally accomplished by passing the water through closed hot water heaters or by injecting steam into the water circuit and 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.

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the overall 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.

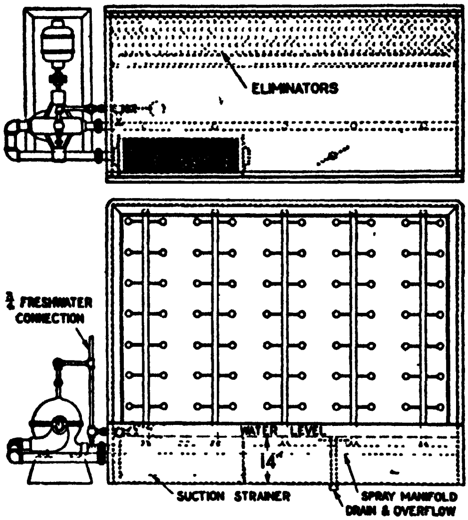


FIGURE 54

Typical Single-Bank Air Washer

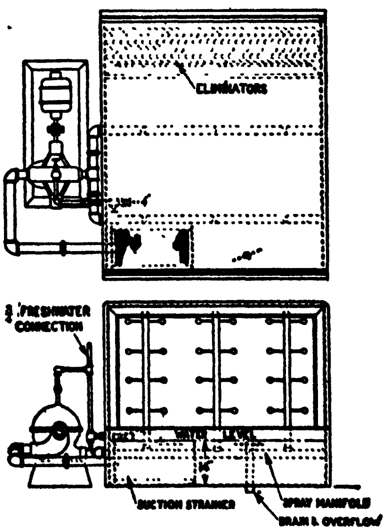


FIGURE 55

Typical Two-Bank Air Washer

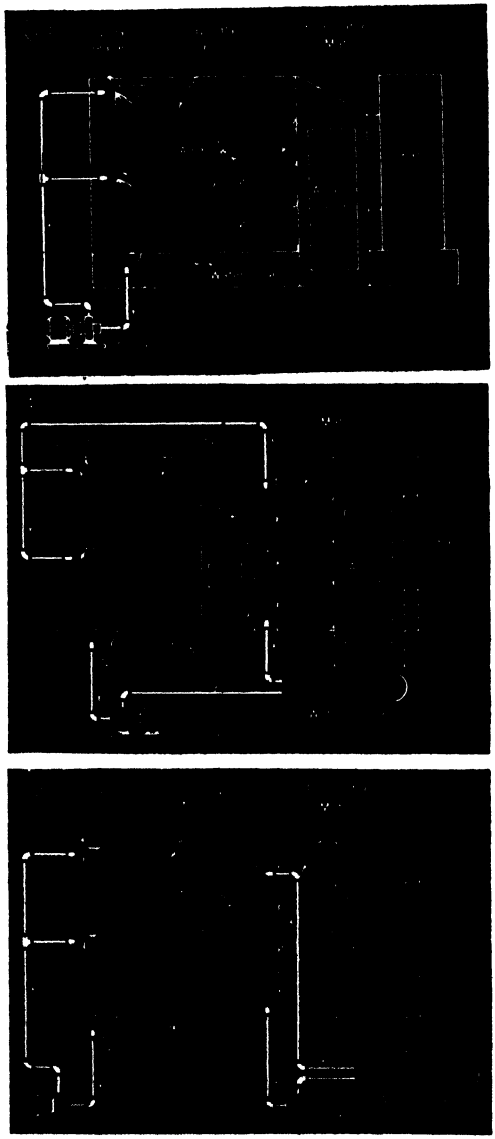


FIGURE 56
Capillary Air Conditioner

1 Class I Capillary Conditioners

As a simple air washer, humidifier or evaporative cooler, the Class I Capillary conditioner using concurrent flow of air and water, is usually applied. For these applications the units are usually selected for the full air capacity ranging up to 1100 cfm per cell maximum. It is to be noted that the pump simply recirculates the water from the tank over the Capillary cells.

As stated previously, 3 gallons of water per cell per minute is required at 6 pound nozzle pressure. The total required pumping head, including friction loss, is 22½ ft plus the height of the unit above the tank or 2½ ft per tier of two cells.

4 Class III Capillary Conditioners

These units involve the use of cooling coils incorporated in the casing of the Capillary unit. Such coils may be for direct expansion refrigeration or for the circulation of cold liquids.

Illustration (4) shows the Class III arrangements where the cooling water is passed first through the coil then over the Capillary cells. This arrangement offers counter flow performance as in a 2 stage Capillary. Whereas a coil ranging from 6 to 12 rows in depth may be required to obtain a desired water temperature rise, the capillary cells may be used after the water has left the coil to carry from 50 to 75 per cent of the load and thus reduce the number of rows of coils correspondingly.

5 Class III with Closed Circuit Coil

The closed circuit of cooling fluid shown in illustration (5) is applied where direct expansion is applied or where, for some reason, the water or brine circuit may not be opened.

In each instance the Capillary in advance of the coil keeps the coil clean, provides the economy of evaporative cooling when entering wet-bulb conditions permit, offers year-round humidity control.

by Courtesy of W. L. Fleischer

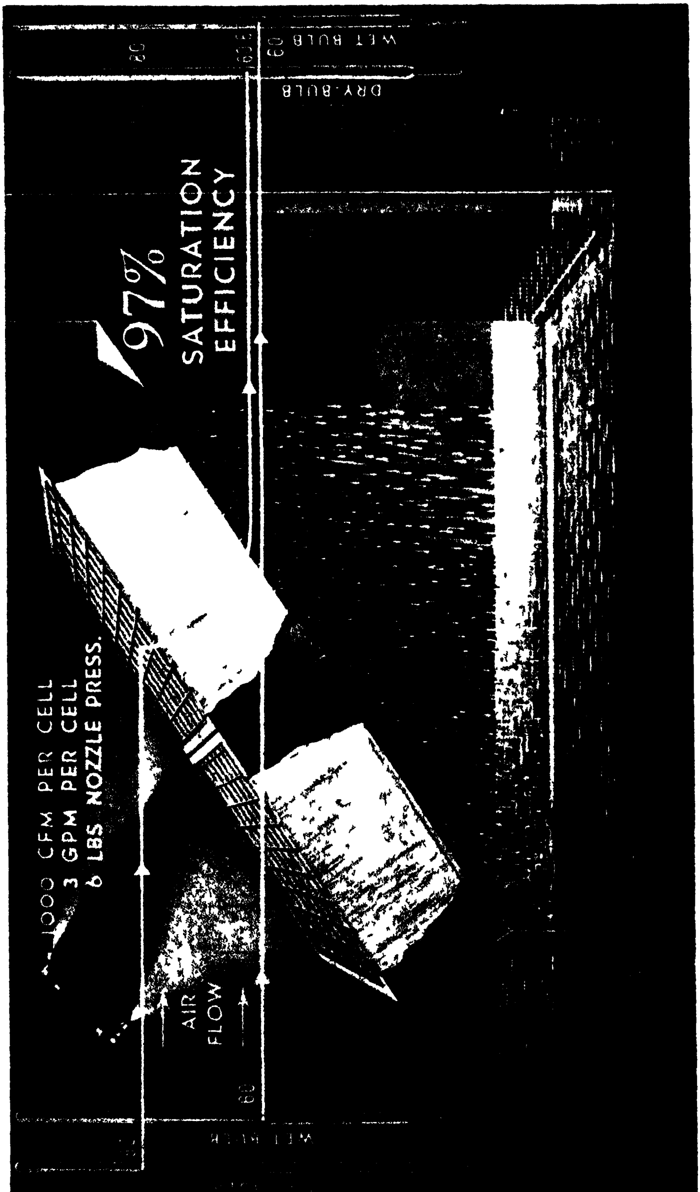


FIGURE 56A
Capillary Air Conditioner

305. Capillary Air Conditioners

The capillary air conditioner offers a combination system adapted to the humidification as well as cleaning of large quantities of outside air for use in textile mills, chemical plants, biological laboratories, hospital operating rooms and many industrial applications. Instead of pumping large quantities of atomized water at a comparatively high pressure, it furnishes intimate and efficient contact between air and the cooling or moistening fluid at lower pressure and with lower consumption of power. The contact is obtained by dividing both water and air into fine streams flowing through low resistance channels, generally consisting of Fiberglas cells. Basic patents were granted on this revolutionary method. Air and Refrigeration Corporation is licensed to manufacture Capillary Conditioners under U. S. Patents 2054809, 2132457, 2139675, and 2149593.

CHAPTER XVI

SPRAY SYSTEMS AND COOLING TOWERS

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its temperature and moisture content, and then it is conducted 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 utilization. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess of air is used to cool a fixed quantity of water.

The increasing number of cooling-for-comfort installations presents very definite problems to municipalities, especially in metropolitan areas, concerning condenser water supply and disposal. So great has this problem become in certain cities that legislation has been enacted to limit the use of city water for condensers. It follows that water-conserving equipment must be developed for cooling-for-comfort purposes. Such equipment includes means for transferring the heat removed from buildings directly to the outside air using water as the heat-carrying medium.

Refrigerating apparatus used in cooling-for-comfort installations performs the function of a heat pump in extracting heat from air at a low temperature level and in discharging it at a higher temperature level. Various refrigerants may be employed by the pump for conveying the heat. In the last stage of the transfer, water commonly is used. Where water costs are low, the easiest course has been to discharge this heat-laden water into the sewer. Water cooling towers are devices in which an intimate contact of water with atmospheric air is achieved, thereby promoting an exchange of heat from the water to the air. The water is cooled in the process, to be used again in the refrigeration cycle.

A successful and economical cooling tower must be capable of simultaneously breaking up the incoming condenser water so as to present the largest possible surface to the air, and bringing into intimate contact with this water the largest possible quantity of air for the least expenditure of power.

The extent of cooling is limited by the wet-bulb temperature of the air. Hence the efficiency of a device as a water cooler is expressed as a percentage as follows:

$$\text{Per Cent Efficiency} = \frac{(\text{hot water temperature} - \text{cooled water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of incoming air}}$$

Water cooling towers may be classified as:

1. The atmospheric type which depends on wind motion for operation.

2. The natural draft type, which employs stack or thermal action for producing air movement through the tower.

3. The mechanical draft type, which employs forced or induced draft fans for creating air movement through the tower. The humidifying air washer is a modification of an induced draft cooling tower.

4. The evaporative condenser, which is a combination of a refrigerating condenser and a mechanical draft cooling tower, described in Chapter XVIII.

306. Commercial Engineering Considerations

Primary considerations in the design of both mechanical and natural draft water cooling towers include the number of gallons per minute to be cooled from a given temperature to a desired temperature and the mean maximum summer temperatures and humidity of the atmosphere at the place where the tower will be located. Then, by applying the natural laws of heat liberation through conduction, convection and radiation and by evaporation, the amount of air required to produce the desired water cooling can be accurately determined. This, in turn, determines the size of the tower, for the temperature of the air leaving the tower should be as close as possible to the temperature of the incoming water, in order to utilize the full cooling potentialities of the air.

307. Water Cooling by Conduction, Convection and Radiation

The transfer of heat from the water to the air by means of sensible heat is not large in quantity. One pound of dry air absorbs only about 2.4 Btu as its temperature is raised 10 degrees Fahrenheit. Therefore, about 417 pounds (about 5600 cubic feet) of dry air must be heated 10 degrees Fahrenheit by the water in order to remove 1000 Btu from the water. Extremely large quantities of air remove comparatively small quantities of heat in this manner.

308. Water Cooling by Evaporation

Water loses its heat to the surrounding air principally by evaporation. One pound of air entering a cooling tower at a temperature of 70° F and a relative humidity of 60% and leaving the cooling tower with a temperature of 95° F and a relative humidity of approximately 90% has, while passing through the tower, absorbed about 6 Btu because of its rise in temperature, but has carried off about 24 Btu because of the water which has been evaporated. In this particular instance, approximately 30 Btu have been removed from the water by means of one pound of air. Of these 30 Btu, only 20% has been by direct transfer of heat, but 80% has been due to evaporation. Each instance of varying air temperature and relative humidity will differ from others, but this proportion will usually vary between 25:75 and 15:85.

In the case of a water cooling-tower serving a steam condenser, the heat absorbed by the circulating water in condensing the steam must equal the heat removed from the circulating water in the cooling system. Therefore, from the statement above, the water evaporated in the cooling system must equal from 75 to 85% of the steam condensed (condensate) in the condenser. It will thus be evident that the make-up to the circulating water supply must also equal from 75 to 85% of the steam condensed (condensate) in the condenser.

Air will evaporate water until it is completely saturated, and the amount of water it will absorb depends upon its initial temperature and its initial saturation or relative humidity. The amount of vapor required to produce complete saturation increases as the air temperature increases.

The wet-bulb temperature is the lowest temperature to which water can possibly be cooled by air. It represents a theoretical limit that is never reached commercially. If the initial temperature of the water to be cooled is 100° F, and the wet-bulb is 75° F, the maximum cooling range theoretically possible is 25 degrees. If the water is actually cooled to 80° F the efficiency of the water-cooling system is 80%.

309. Cooling or Heating of Water by Air or Steam

Sheets, drops or jets of water falling or flowing through air or steam reduce the temperature of water, not instantaneously, but gradually, to a point within three to four degrees of the heating or cooling medium.

The process is described under the subject of adiabatic saturation in an earlier topic. A certain period of time is required for sensible heat to enter or leave a film of water. Therefore the diameter of drop or thickness of sheet of water must be considered in designing cooling towers, air washers, baudelot coolers, atmospheric condensers, etc. The next topic is a purely theoretical mathematical analysis of the subject.

310. Size and Surface of Drops

$$\text{Ratio} = \frac{\text{surface of sphere}}{\text{volume of sphere}} = \frac{6}{d} \tag{1}$$

d = diameter

TABLE 143

SIZE AND SURFACE OF DROPS

Diameter of drop in inches	$\frac{1}{32}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{5}{32}$
Diameter of drop in millimeters	0.7938	1.588	2.381	3.175	3.968
Surface of drop in square inches	0.00302	0.01227	0.02761	0.04909	0.07670
Volume of drop in cubic inches	0.000016	0.000128	0.00043	0.00102	0.002
Ratio $\frac{\text{surface}}{\text{volume}}$	192	96	64	48	38.4

Example 1: If 1728 cu in. of water is broken up into drops $\frac{1}{32}$ in. in diameter the total external surface of all the drops may be rapidly calculated by multiplying total $1728 \times 192 = 331{,}776$ sq in. or 2300 sq ft.

Example 2:

If one cubic foot of water is broken into drops $\frac{5}{32}$ in. in diameter what is the total surface in square feet?

1 cu in. gives a surface of 38.4 sq in.
1728 cu in. give a surface of 1728×38.4 sq in.
1 cu ft gives a surface of 460.8 sq ft

The importance of a fine spray for rapid transfer of heat is demonstrated.

TABLE 144

PRECIPITATION VALUES

Popular Name	Diameter Water Drop in Inches	Velocity Feet Per Second	Weight Water in Grains Per Cubic Foot Air
Fog	0.0004	0.009	11
Mist	0.004	0.75	100
Drizzle	0.008	2.25	200
Light Rain	0.018	6	600
Moderate Rain	0.04	12	600
Heavy Rain	0.06	15	1000
Excessive Rain	0.084	18	1200
Cloudburst	0.12	21	1200

311. Size and Surface of Sheets

The ratio of flowing sheet surface to volume is $\frac{1}{d}$ where d = thickness, because only one side is exposed. For falling sheets, the ratio equals $\frac{2}{d}$ because both sides are exposed.

TABLE 145

Thickness of sheet, inches	$\frac{1}{32}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{5}{32}$
Ratio $= \frac{\text{surface}}{\text{volume}} = \frac{1}{d}$ (flowing)	32	16	10.7	8	6.4
Ratio $= \frac{\text{surface}}{\text{volume}} = \frac{2}{d}$ (falling)	64	32	21.4	16	12.8

Example. A cubic inch of water flowing in sheets ⅛ in. thick will expose 8 sq in. of surface. A cubic foot of water ⅛ in. thick exposes 1728 × 8 sq in. or 96 sq ft.

312. Heat Transfer Through Sheets of Water

Symbols:

For water $K_F = 0.155$ Kgcal per sq m per sec per ° C per mm thickness

For water K_B = 4.56 Btu per hr per sq ft per ° F per inch thickness

For water K_M = 0.03168 Btu per sec per sq ft per ° F per mm thickness

For water K_1 = 0.0012673 Btu per sec per sq ft per ° F per inch thickness.

In calculations involving heat transfer through thin films of water a thermal conductivity constant in seconds and millimeters (1/25 in.) thickness is most convenient; therefore in the following K_F is used, and all units reduced to the metric system. Where

S = area of heat transfer surface in square meters

N = thickness in millimeters of layer perpendicular to S and in direction of heat flow

Z = time in seconds

C = quantity of heat in Calories

θ_a = initial difference in temperature, in ° C, between exterior surface of film exposed to air and the air

θ_e = final difference in temperature between interior surface of film and air after a small period of time

Fourier's law proves that in a state of heat equilibrium

$$C = \frac{SK_FZ\theta_a}{N} \tag{2}$$

However, in cooling or heating there is by no means a state of equilibrium.

Assume an average thickness of ½N.

Then $$C = \frac{SK_fZ\theta_a}{\frac{N}{2}} \text{ (no equilibrium)} \tag{3}$$

Assume a thin layer divided into 100 thinner layers.

Where

t_a = temperature air surrounding sheet of water

t_w = temperature of all 100 layers at beginning of fall through air

$$t_a - t_w = \theta_a$$

(Water is heated by air in this problem if cooling effect assumed is $t_w - t_a = \theta_a$.)

At the end of the falling period the outer layer will approach the temperature t_a but the inner layers will not have been heated as rapidly as the outer layers. It is obvious that the length of time required for heat to penetrate the 100 theoretical layers should be known.

What is the mean increase of temperature of the 100 t layers where

t_a = initial temperature of warmer substance (air)
t_w = initial temperature of cooler substance (inner layer)
t_a = final temperature of warmer substance (excess air)
t_e = final temperature of innermost layer

$$t_a - t_w = \theta_a$$
$$t_a - t_e = \theta_e$$

Assume $$\theta_e = \frac{\theta_a}{100}$$

From log mean difference table (Table 38)

when $$\frac{\theta_e}{\theta_a} = \frac{1}{100} \quad \text{then} \quad \theta_M \text{ for } (\theta_a = 1) = 0.215$$

$$T_{EM} = 0.215\ \theta_a \tag{4}$$

T_{EM} is the average mean increase in temperature when heating 100 ideal layers of water. Therefore $C = 0.215\ \theta_a\ SN = T_{EM}\ SN$ which is heat absorbed by water heated to depth N. To obtain an expression of time Z_s during which quantity of heat C has penetrated through surface S for constant temperature difference θ_a

$$C = \frac{2SK_FZ\theta_a}{N} = 0.215\theta_a SN \text{ in Calories}$$

$$Z = \frac{0.215N^2}{2K_F} = \frac{0.1075N^2}{K_F} = 0.694N^2 \quad (5)$$

$$N = 1.2\sqrt{Z} \text{ in millimeters} \quad (6)$$

N = thickness of a sheet of water one meter square when heated by air or steam on one side to acquire temperature of air on that side and just beginning to show signs of activity of getting warm on other side in time Z seconds.

$$N = 0.048\sqrt{Z} \text{ in inches.} \quad (7)$$

313. Heating of Mass of Water

The sheet may be thicker than the depth of penetration, since heat in a certain time only penetrates to N. Assume sheet is d mm thick and d is three times as thick as N. The balance of mass d — M is not heated. Therefore the temperature change of the mass is ⅓ the temperature rise of the sheet to depth N.

$$\text{Heat in N layer} = T_{EM} \times N$$
$$\text{Heat in d layer} = T_{EP} \times d$$

T_{EP} is an unknown quantity

$$T_{EP} = \frac{T_{EM} \times N}{d} \quad (8)$$

$$T_{EP} \text{ for sheets} = \frac{N\theta_a \times 0.215}{d} \quad (9)$$

$$T_{EK} \text{ for drops} = \frac{1.29\theta_a N(d - 0.4N)^2}{d^3} \quad (10)$$

where

T_{EK} = change in temperature of mass of drop
d = diameter drop

314. Problem Involving Sheets and Drops of Water

Problem (1). A sheet of water flowing over a flat, nearly horizontal surface is ½ in. thick and is exposed to the air for 5

TABLE 146

HEAT TRANSFER THROUGH FALLING SHEETS OF WATER

Period in Seconds Z	Height Free Fall in Feet	N = Distance Heat Penetrates Milli-meters	Inches	C_M Kilocalorie Per Square Meter Per Degree Difference C	C_B Btu Per Square Foot Per Degree Difference F
0.1	0.16	0.38	0.0152	0.082	0.0168
0.2	0.64	0.532	0.02128	0.116	0.02378
0.3	1.45	0.66	0.0264	0.141	0.0289
0.5	3.30	0.85	0.0339	0.183	0.0375
0.7	7.89	1.0	0.04	0.215	0.0440
1.0	16.10	1.2	0.048	0.26	0.0533
1.2	27.18	1.315	0.0526	0.282	0.0578
3.0	Use Baffles	2.1	0.084	0.452	0.0926
5.0	Use Baffles	2.68	0.107	0.58	0.1189
7.0	Use Baffles	3.2	0.128	0.69	0.1414
9.0	Use Baffles	3.6	0.144	0.775	0.1589
11.0	Use Baffles	4.0	0.16	0.86	0.1763
13.0	Use Baffles	4.35	0.174	0.935	0.191
15.0	Use Baffles	4.65	0.186	1.00	0.205
20.0	Use Baffles	5.376	0.215	1.155	0.236

$C = 0.215\ N$ when $\theta_a = 1$

$N = 1.2\sqrt{Z}$ millimeters $\qquad$ $N = 0.048\sqrt{Z}$ inches

CONVERSION FACTORS

One Millimeter = 0.03937 Inches
One Calorie = 3.969 Btu
One sq Meter = 10.76 sq ft
One °C = 1.8 °F
$C_M \times 0.205 = C_B$

seconds. What is the change in temperature of the mass of water due to sensible heat exchange if the temperature of the

air at the beginning is 30° C (86° F) and the temperature of the water is 20° C (68° F)? Give the solution in metric units.
Solution. From Table 146, heat penetration in 5 seconds is 2.68 mm. One-half inch reduced to mm gives a sheet thickness of 12.5 mm. $\theta_a = 10$. Substituting in formula (9)

$$T_{EP} \text{ for sheets} = \frac{2.68 \times 10 \times 0.215}{12.5} = 0.45° \text{ C rise in temperature.}$$

It is evident from the small change in the temperature of the mass, that the sheet is too thick. By trial and error

$$T_{EP} = \frac{2.68 \times 10 \times 0.215}{4} = 1.44° \text{ C}$$

$$\text{For d} = 1 \text{ mm, and } T_{EP} = 5.65° \text{ C}$$

If drops were used instead of flowing sheets more surface would be exposed for the same volume of water, giving a greater change in temperature.
Problem (*2*). If drop diameter is ⅛ in. or 3.175 mm, the fall is 16.1 ft, air temperature 30° C and drop temperature 20° C, what is the rise in temperature of the mass during the period of fall?

Solution. From Table 146

$$Z = \text{one second}$$
$$N = 1.2 \text{ mm}$$

Substituting formula (10) for drops gives

$$T_{EK} = \frac{1.29 \times 10 \times 1.2(3.175 - 0.4 \times 1.2)^2}{3.1753} = 3.5° \text{ C}$$

315. Operation and Types of Cooling Towers

Commercial cooling tower design is guided by the following requirements: Suspension of the water in the air for a reasonable length of time, with a pumping head as small as possible; maximum area for free admission of air with the lowest corresponding loss of water; structure as light in weight as possible, especially if placed on roof; moisture-resistant construction.

The cooling of the water is mostly by evaporation, but radia-

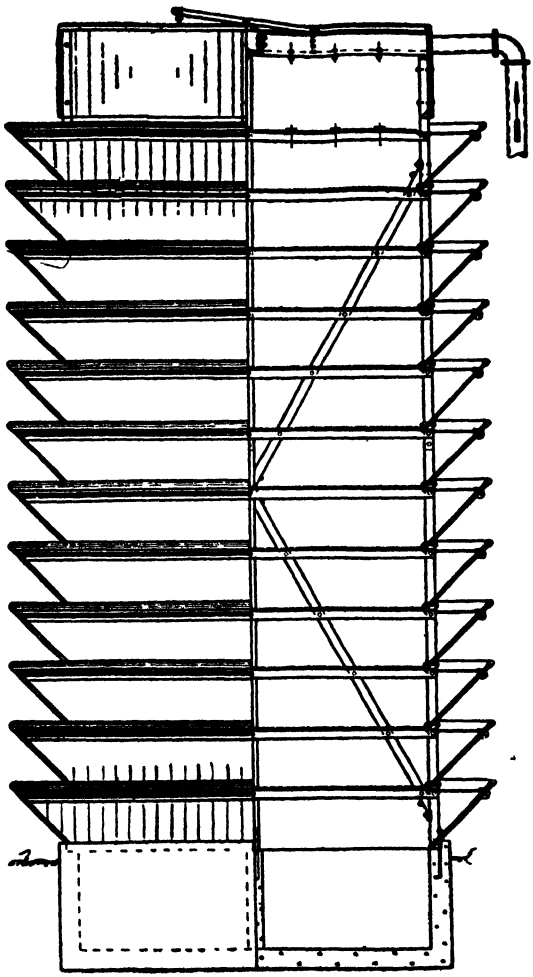


FIGURE 57

Half-section and Elevation, Sectional Portable Atmospheric Cooling Tower

Induced Draft Cooling Tower of redwood construction.

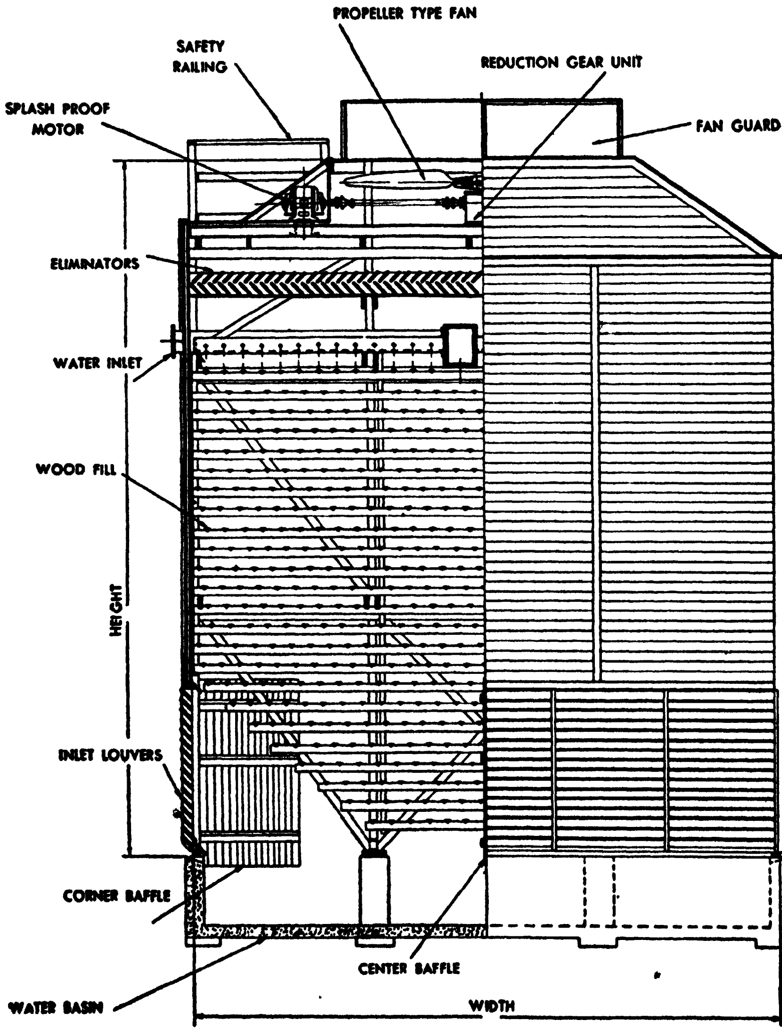


FIGURE 58

Induced Draft Cooling Tower

Courtesy of C. H. Wheeler, Philadelphia

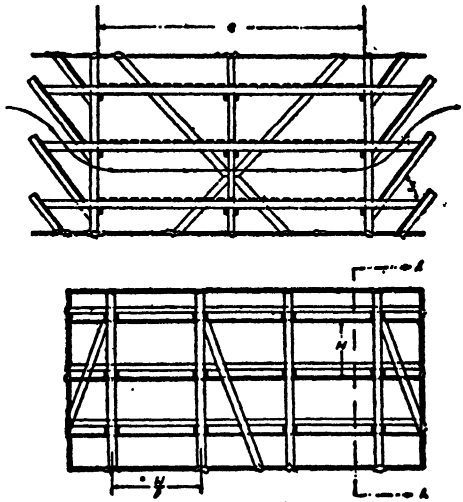


FIGURE 59

Atmospheric Splash-Deck Tower

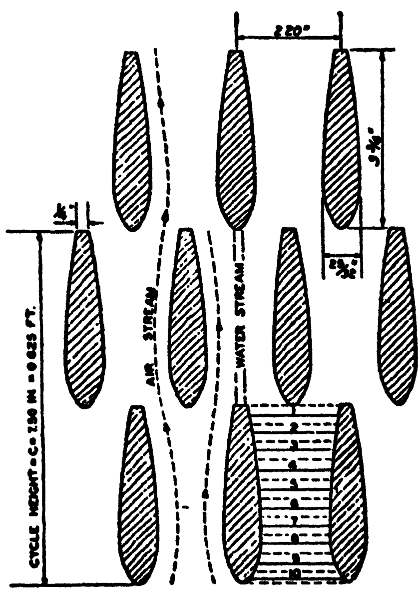


FIGURE 60

Streamlined Checkerwork

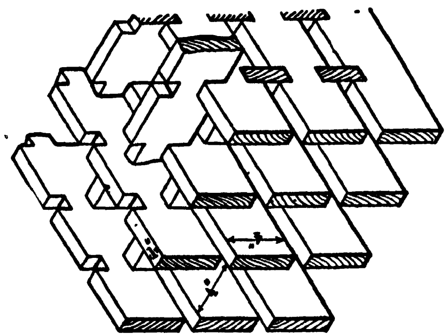


FIGURE 61

Cooling Tower Checkerwork

tion, convection and conduction also enter into the heat exchange; consequently heat exchange is most efficient when the water is exposed as a very fine spray or sheet. Expulsion of the air after it passes through the spray is as important as the free admission of the air. The air circulation mechanism in the tower may be mechanical, i.e., a fan, or natural, i.e., horizontal atmospheric air currents and internal vertical currents caused by the temperature differences. In order to utilize all or part of these currents different types of towers have been evolved.

316. Atmospheric Towers

These are least expensive hence most popular, depending on horizontal atmospheric currents for air admission and by internal vertical currents which also assist in cooling. The vertical current is a small factor for small temperature differences, but where temperature differences are larger, vertical currents play an important part.

The atmospheric tower must be so placed as to be exposed to the prevailing winds in summer, must be of light weight and must be braced for a wind velocity of 90 mi per hr. The area of its horizontal cross-section is a factor of its capacity; its height, a factor of its efficiency. Various methods are used to break up the water by stages or decks. After the water has passed through from deck to deck it is collected in a pan at the bottom. Almost every commercial cooling tower builder has his individual arrangement of louvers on the outside of the tower. These should be constructed to give free admission to horizontal air currents and to prevent the water from blowing out of the tower. Some manufacturers use only spray nozzles instead of cooling decks inside the tower. This type of tower is nothing more than an elevated spray pond and loses a large percentage of the suspended water. The atmospheric tower is recommended wherever there is room for such a structure.

317. Forced Draft Towers

Where floor space is limited and the location not exposed to prevailing winds a mechanical tower must be used. This consists of a solidly built shelf with a fan at the bottom; cooling surface inside, above the fan and an additional height to accommodate eliminators. The water enters at the top and passes over the cooling surface while the fan blows the air upward, giving a counter-current effect. Since the fan delivers a known quantity of air the results which may be expected in a forced draft tower can be correctly calculated.

318. Cooling Tower Formulas

Designers of cooling towers, humidifiers, air washers and spray ponds resort to experimental data which gives an empirical equation of the following order:

$$S = \frac{C_S}{K_S T_M} \tag{11}$$

where

S = area of wetted surface in square feet
T_M = log mean temperature difference, air and water
K_S = Btu per hour per square foot per degree difference
C_S = sensible heat transfer in warming or cooling

Note: C_S does not include heat of evaporation.

distributed. Sheets of water 1/8 in. thick, exposed on both sides, may be assumed. With atmospheric draft, assume a wind velocity of one mile per hour. Ideal cooling tower design tends towards a maximum of falling water, broken up into as fine sheets, jets and drops as is practical.

In an atmospheric tower, free area for vertical air movement is as important as free area for horizontal movement. Hence distributors are spaced to allow a free area of 60% of the total area between distributors.

In a mathematical analysis of the heat exchange in an atmospheric tower, assume that the air inlet is at the bottom and the outlet at the top; the water inlet at the top and the outlet at the bottom, giving a counter-current effect, as in a forced draft tower. The actual heat exchange in an atmospheric tower is a transverse effect, instead of counter-current, but for simplicity of calculation assume a counter-current effect.

The average total fall of water from top deck to bottom deck varies from 22 to 30 ft; in the case of a forced draft or induced draft tower additional height must be allowed for fan and eliminators. In both types of towers the sheets of water have an assumed length of 6 ft per deck for at least five decks, giving a total of 30 ft. In 0.6 sec a body falls about six feet and has a velocity of 19.32 ft per sec at the end of the fall; therefore, the average velocity of the water between decks is 10 ft per sec. In an atmospheric tower with, e.g., horizontal wind velocity of 4 ft per sec and water velocity of 10 ft per sec a relative velocity of 10.8 ft per sec; at one mile per hr wind velocity a relative velocity of about 10.9 ft per sec; at 15 miles per hour wind velocity 22.4 ft per sec relative velocity. In a forced draft tower the relative velocity is the sum of air velocity and water velocity. In the analysis only air velocity and not relative velocity is considered. Air movement due to air velocity is more important, as the air carries off the vapor after it has done its work of evaporation; the ideal tower removing the vapor as soon after it is formed as possible. Hence height in direction of air and width in direction of air is limited.

321. Cooling Tower Problem (Cool Weather Conditions)

Given: An atmospheric tower 12 ft by 16 ft with a pan area of 200 sq ft. The warm water enters at the top and flows over five decks of distributors in a theoretically vertical sheet. Total length of fall is 30 ft. The water is to be cooled at the rate of 6,000 gal per hour from 122 to 68° F. The air enters at 37° F wet-bulb. What area, in square feet, of wetted surface is required?

Solution: Assume a counter-current effect, air entering at 37° F wet-bulb. Tests show that in cool weather air leaves the tower within ten degrees of the water inlet temperature and in a saturated condition; in this case air outlet temperature is 112° F.

(Note: In summer air leaves a tower at within three to five degrees of temperature of incoming warm water, depending upon the humidifying efficiency of the design.)

Total heat content of 1 lb air at 112° F =	92.10 Btu
Total heat content of 1 lb air at 37° F =	13.87 Btu
Total heat gained per pound air	78.23 Btu

Total heat abstracted from water equals 6,000 gal × 8.3 lb × (122° F − 68° F) = 2,700,000 Btu per hr. 2,700,000 divided by 78.23 Btu gives 34,500 as the number of pounds of air required to remove heat.

Water is partly cooled by evaporation. However, in this problem, we are only concerned with the sensible heat given up to the air by the water by conduction in determining the area of cooling surface.

Sensible heat per pound air at 112° F =	26.88 Btu
Sensible heat per pound air at 37° F =	7.68 Btu
Sensible heat absorbed per pound of air	19.20 Btu

34,500 lb air $\times$ 19.2 = 660,000 Btu per hour sensible heat given up to air by water by conduction.

$$S = \frac{660{,}000}{K_S \times T_M}$$

$$T_M = \text{Log mean temperature difference}$$

$$T_a = 122 \qquad T_e = 68 \qquad t_a = 37 \qquad t_e = 112$$

$$\theta_e = T_a - t_e = 10 \qquad \theta_a = T_e - t_a = 31$$

$$\frac{\theta_e}{\theta_a} = 0.322$$

$$\theta_M \text{ for } (\theta_a = 1) = 0.60$$

$$\theta_M = 18.60 = T_M$$

In winter assume $V = 4$, therefore $K_S = 3.1$

$$S = \frac{660{,}000}{18.6 \times 3.1} = 11{,}500 \text{ sq ft wetted surface}$$

322. Total Surface (Winter)

10,850	sq ft area of sheets of falling water (both sides)
200	sq ft pan surface
450	sq ft approximate horizontal distributor surface
11,500	total wetted surface (natural draft)

Summary

$\frac{10{,}850}{2}$ = 5,425 sq ft area of sheets.

Area of one sheet = 30 $\times$ 16 = 480 sq ft

$\frac{5{,}425}{480}$ = 12, number of sheets, requiring 12 distributors

Length of distributor, 12 ft

Total length of distributors per deck, 144 ft

100 gpm over 144 ft distributor = ¾ gpm per foot of distributor.

Area (horizontal) per deck of distributors = 40% of 200 (Note: distributors may be cut down to a 2½ in. width if necessary).

6,000 gallons of water spread over 5,425 sq ft will theoretically furnish a film 9/64 in. thick.

However the sheet in falling will be broken up into drops and jets.

Active horizontal surface = 80 sq ft

$\frac{100}{80}$ = 1¼ gallon per minute per sq ft active horizontal surface.

Cooling Tower Problem (Summer Conditions)

Total heat content per pound air entering	44.41 Btu at 80° sat.
Sensible heat content per pound air entering	21.10 Btu at 80° sat.
Latent heat content per pound air entering	23.31 Btu at 80° sat.
Total heat content per pound air leaving	56.93 Btu at 90° sat.
Sensible heat content per pound air leaving	24.54 Btu at 90° sat.
Latent heat content per pound air leaving	32.39 Btu at 90° sat.

6,000 gal × 8.3 × 13 = 647,400 Btu withdrawn from water by both evaporation and conduction.
56.93 − 44.41 = 12.5 Btu increase per pound air
647,000/12.5 = 52,000 lb air per hour
52,000/60 = 870 lb air per minute
870 × 14.4 = 12,528 cu ft air per minute
32.39 − 23.31 = 9.08 Btu increase latent heat per pound air
52,000 × 9.08 = 472,160 Btu absorbed by vapor per hour
1,043.6 Btu latent heat per pound
472,160/1,043.6 = 456 pounds water lost per hour
24.54 − 21.10 = 3.44 Btu increase in sensible heat per pound of air
3.44 × 52,000 = 178,880 Btu total increase of sensible heat or heat given up by water to air by conduction:

$$S = \frac{C_S}{K_S T_M}$$

where

S = wetted surface
K_S = coefficient = 2.25 at 1½ miles per hour wind velocity
T_M = log mean temperature difference
C_S = sensible heat absorbed = 178,880

To find T_M or log mean temperature difference

$$T_a = 98° \text{ F} \qquad t_e = 90° \text{ F} \qquad T_e = 85° \text{ F}$$

$$t_a = 80° \text{ F (wet-bulb)}$$

$$\theta_a = T_a - t_e = 8 \qquad\qquad \theta_e = T_e - t_a = 5$$

$$\frac{\theta_e}{\theta_a} = \frac{5}{8} = 0.625$$

$$O_M \text{ for } (\theta_a = 1) = 0.8$$

$$T_M = 0.8 \times 8 = 6.4° \text{ F}$$

$$S = \frac{178{,}880}{6.4 \times 2.25} = 12{,}300 \text{ sq ft wetted surface}$$

For summer conditions the 12,300 sq ft of surface would be apportioned as follows:

200 sq ft pan surface
450 sq ft horizontal distributor surface
11,650 sq ft vertical sheets of water

The data above refer to conditions assumed in summer for refrigerating purposes. A safe allowance is 3 sq ft horizontal active surface per ton of refrigeration, or ¾ gal minimum to 1½ gal maximum per minute per foot length of distributing trough. Refrigerating engineers allow 4 gpm per ton of refrigeration at Latitude 40.

323. Atmospheric Cooling Tower Problem (Summer)

Problem.

Assume outside air conditions are 95° F dry-bulb, 80° F wet-bulb and 51% relative humidity. Cool 6,000 gallons of water per hour from 98° F to as low a temperature as possible with a natural draft tower. Find the wetted surface of an atmospheric tower.

Solution.

Tests have shown that water may be cooled to within 3 to 5° F of outside wet-bulb temperature when there is a minimum flow of 1⅓ gallons of water per minute per sq ft of horizontal active surface or from ¾ gallon to 1½ gallons per minute per foot of distributor. Tests show that in the case of properly designed towers air leaves the top nearly saturated after being heated to within 5° F of the temperature of the warm water entering, assuming that, regardless of outside dry-bulb, the air enters at the bottom at outside wet-bulb.

T_a = 98° F temperature of water entering at top
t_e = 90° F temperature of saturated air leaving at top
T_e = 85° F temperature of water leaving at bottom
t_a = 80° F wet-bulb temperature of air entering bottom

21.10 − 18.00 = 3.10 Btu, increase in sensible heat per pound of air

51,000 × 3.1 = 158,100 Btu per hour, increase in sensible heat

$$S = \frac{C_s}{K_s\,T_M}$$

where

$T_M = 5$
$K_s = 2.24$
$C_s = 158{,}100$
$S = 14{,}200$ sq ft wetted surface

15,900 sq ft total surface would be required for a ten degree range with outside wet-bulb temperature at 70° F.

The following table gives an idea of temperature variations to be expected with a standard commercial atmospheric tower.

TABLE 148

1⅓ GAL PER MINUTE PER SQUARE FOOT ACTIVE HORIZONTAL TOWER SURFACE

Cooling Range	Wet-Bulb 60° F	65° F	70° F	75° F	80° F
	Temperature of Water off Tower				
10°	66	70.3	74.3	78.7	83
9°	65.7	69.9	74	78.3	82.7
8°	65.2	69.4	73.6	78.2	82.6
7°	64.5	68.8	73.3	77.6	82.1

324. Forced Draft Towers

A forced draft tower occupies about ⅕ the space of a natural draft tower. Referring to the preceding problem of cooling 6000 gallons per hour by forced draft under the same atmospheric conditions of wet-bulb, but 95° F entering water temperature, apply the same mathematical analysis. Howevet, due to higher air velocities the required surface will be less. Good practice demands 500 ft per minute air velocity through water and 1000 feet per minute between distributors. A narrower distributing trough will be required, not over 2½ inches wide

$$K_f = 0.4 + 1.36\sqrt{8.3} = 5.9 \text{ for forced draft}$$
$$S = \frac{178{,}880}{5 \times 5.9} = 6000 \text{ sq ft wetted surface}$$

$$T_M = 5 \text{ (since water enters at } 95° \text{ F)}$$

First trial use sheets 30 ft high.

$6000 \div 2 = 3000$ sq ft sheets (both sides exposed)

(horizontal pan surface and horizontal surfaces not considered).

Area one sheet 30 ft $\times$ 5 ft $= 150$ sq ft

$3000 \div 150 = 20$ sheets of water

20 distributors per deck 5 ft long = 100 ft of distributors

One gallon per minute per foot of distributor

Horizontal surface distributors (one deck) $= 100 \times 0.2 =$ 20 sq ft

Minimum of 5 decks = 100 sq ft horizontal surface

20 Distributors 2½ in. wide = 50 in. total width

The minimum width of distributor is 2½ in. Therefore with a tower 5 ft $\times$ 5 ft using 20 distributors 2½ in. wide will not give enough area between distributors for free air flow. The net free area between distributors should be 60% of the total area.

50 in./0.40 = 125 in. = (approximately) 10 ft

Therefore, the minimum dimension of tower to permit proper flow of 12,528 cu ft of air per minute is 10 ft $\times$ 10 ft.
12,528/100 = 125 ft air velocity

$$\text{New } K_S = 0.4 + 1.36\sqrt{1} = 2$$

$$\frac{178{,}800}{5 \times 2} = 17{,}880 \text{ sq ft, surface}$$

$$\frac{17{,}880}{2} = 8{,}940 \text{ sq ft, sheet surface}$$

$$\frac{8{,}940}{20} = 447 \text{ sq ft, area of one sheet}$$

$$\frac{447}{10} = 44.7 \text{ ft height of sheet}$$

Therefore, the tower must be increased in height from 30 to 44.7 ft and, in addition, allowance must be made for the height of the eliminators and fan housing. An eliminator is absolutely necessary to prevent excessive loss of water.

325. Forced vs. Induced Draft

The purpose of any cooling tower is to mix sufficient volumes of outside air with the water to be cooled, in such a manner that the air carrying away the heat of the water does not carry entrained water, which would deposit as rain on surrounding objects. Further, the warmed air should leave the tower in such a manner that it cannot be drawn back and recirculated, for this would reduce the water cooling efficiency of the tower.

Forced draft cooling towers are the oldest types of mechanical draft towers, the fans being located at the base of the tower so that they force air into the side or sides of the tower, upward through the falling water and thence out through the top of the tower.

Induced draft cooling towers efficiently and thoroughly eliminate the difficulties experienced by many users of the forced draft type, the fans being located at the top of the tower draw air, through louvers on all sides near the base of the tower, upward through the falling water, and discharge it to the atmosphere.

326. Advantages of Induced Draft

The construction and operating characteristics of induced draft cooling towers adapt them to every installation and operating condition. They are the type most used today because they have many advantages over forced draft cooling towers; some of these advantages may be enumerated as follows:

Non-recirculating. Air leaving the tower is at high velocity, thus it is discharged vertically upward to a considerable distance before it mixes with the atmosphere. This prevents the recirculation of warmed air through the tower which frequently occurs on forced draft towers due to high-velocity inlet air and low-velocity outlet air.

Non-freezing. The induced draft fan is in the path of the warm used air, thus the temperature of fan parts is maintained well above freezing, and there is little hazard of ice accumulation, even in sub-freezing weather.

Minimum noise. Location of fans at the top of the tower results in upward vertical projection of any fan noise, so that it cannot be objectionable to surrounding areas.

Minimum stratification. Cooling effect is uniformly produced across the entire cross-section of the tower, due to air suction at entrance on all sides at the base of tower.

Minimum space. The area required for the cooling tower is less because no additional space is required for fan hood and mechanical drives since all of these are on top of the induced draft type tower.

327. Practice in Atmospheric Spray Cooling Towers

TABLE 149

Number	Nominal Rating Gallons per Minute	Dimensions, Feet Width Length Height	Pipe Size Inlet	Pipe Size Outlet	Make Up
	5	2 × 3 × 6	1½	2	½
(a)	25	4 × 4 × 8	2	2½	½
(b)	25	6 × 6 × 8	2½	3	½
(c)	100	6 × 8 × 12	3	4	¾
(d)	200	8 × 16 × 12	4	6	1
(e)	300	12 × 18 × 12	6	8	1
(f)	500	12 × 30 × 12	6	8	1¼
(g)	1000	12 × 66 × 12	8	10	1½

For refrigeration, allow a 6° F rise through the condenser. The ratings above are based on a 70° F wet-bulb temperature and a wind velocity of 3 miles per hour.

TABLE 150

CORRECTION WHEN WET-BULB IS ABOVE 70° F

Temperature, ° F	Percentage to be Added	Factor
72.5	2.5%	1.025
73.5	4.0%	1.036
75.0	5.0%	1.05
77.5	6.5%	1.065
80.0	8.0%	1.08

TABLE 151

CORRECTION WHEN WET-BULB IS BELOW 70° F

Temperature, ° F	Percentage Deduction	Factor
67.5	2.5%	0.975
65.0	5.0%	0.95
62.5	6.5%	0.935
60.0	8.0%	0.92

328. Rating of Atmospheric Towers On Preceding Examples

TABLE 152

Cooling Range, °F	Factor 0.7	1	1.3
	Approach to Wet-Bulb		
5	3.5	5	9
10	6.5	9	15.5
15	9	12.5	20
20	12	15.5	24
25	14	18	28
30	16	19	31

Note: For capacities between two towers use *larger* size.

Example. It is necessary to cool 100 gpm from 90° F to 80° F with air at 73.5° F wet-bulb. Select the proper tower from the table.

Solution. Cooling range equals $90° F - 80° F = 10° F$; an approach to wet-bulb of $80 - 73.5 = 6.5° F$ which gives a factor of 0.7. Correction factor is 1.036.

$$\text{Normal rating} = \frac{\text{capacity}}{\text{product of factors}} = \frac{100}{0.7 \times 1.036} = 138 \text{ gpm}$$

TABLE 153

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

TABLE 153 (*continued*)

	Cooling Pond	Spray Pond	Spray Tower	Deck Tower	Mechanical Draft	Indoor Tower
Weight per square foot	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

x Not comparable

329. Spray Cooling Ponds

The spray pond consists of a basin above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist, because the individual drops must be heavy enough to fall back into the basin and not drift away with the air movement. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in temperature between the water and the wet-bulb air temperature, 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 to 7 ft above the edge of the basin, to supply from 5 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 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 by 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.

TABLE 154

EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

Equipment	Cooling Efficiency—% Minimum	Usual	Maximum
Spray Ponds	30	40 to 50	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

330. Mathematical Analysis of Spray Pond Water-Cooling

Problem. To cool 6,000 gallons of water per hour through a range of 5° F when outside dry-bulb is 98° F and wet-bulb 80° F. Initial temperature of the water is 88° F and it is cooled to 83° F, i.e., within 3° F of the wet-bulb. With a nozzle pressure of 7 lb per sq in., what is minimum diameter of drops assuming 100% effective cooling?

Solution.

Reduce all quantities to the metric system.

$$\theta_a = 98°\text{ F} - 88°\text{ F} = 10°\text{ F} = 5.5°\text{ C}$$

$$T_{EK} = 88°\text{ F} - 83°\text{ F} = 5°\text{ F} = 2.75°\text{ C}$$

7 lb pressure = 14 ft rise and 14 ft fall

$$Z = 1.86\text{ seconds}$$

$$N = 1.2\sqrt{1.86} = 1.63\text{ mm}$$

$$d = 2$$

Substitute in formula (8)

$$T_{EK} = \frac{1.29\theta_a N(d - 0.4N)^2}{d^3}$$

Solve for d^3

$$d = 1.2 \text{ millimeters}$$

331. Spray Pond Data

The average of several careful tests of a spray pond of 1,600 sq ft area, spraying 100 gpm through four nozzles carrying 25 gpm each, 1½ in. pipe size shows the following:

TABLE 155

SPRAY POND DATA

Cooling Range	Wet-Bulb 65° F	Wet-Bulb 70° F	Wet-Bulb 75° F	Wet-Bulb 80° F
	Temperature of water leaving nozzles			
5°	70°	74°	78.4°	83°
6°	70.4°	74.4°	79°	83.4°
7°	71.4°	75°	79.4°	84°

TABLE 156

SPRAY POND SPACE REQUIREMENTS

System Number	Minimum Space Required, Unenclosed	Number of Groups	Capacity Gpm	Lineal Feet of 12-ft Redwood Louver Fence	Minimum Space Required Enclosed Inside Fence	
1	40 ft × 40 ft	1	100	144	36 ft × 36 ft	Single Line systems
2	40 ft × 52 ft	2	200	168	36 ft × 48 ft	
3	40 ft × 64 ft	3	300	192	36 ft × 60 ft	
4	40 ft × 76 ft	4	400	216	36 ft × 72 ft	
5	40 ft × 88 ft	5	500	240	36 ft × 84 ft	
6	52 ft × 76 ft	8	800	240	48 ft × 72 ft	Double Line systems
7	52 ft × 88 ft	10	1000	264	48 ft × 84 ft	
8	52 ft × 112 ft	14	1400	312	48 ft × 108 ft	

332. Summary

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. Atmospheric wet-bulb temperature at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air.)
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service (refrigerating, internal combustion engine and steam condensing).
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for water-cooling 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 a quantity depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigeration tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving-hot-water-temperature within 3 or 4° F of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10° F difference.

Table 157 lists several gases with data of 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 counterflow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

The temperature range, once the hot water temperature is approximately known, depends upon:

1. The maximum wet-bulb temperature at which the full quantity of heat must be dissipated.

2. Efficiency of the atmospheric cooling equipment considered.

Design wet-bulb temperatures are given in Chapter XIV.

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

TABLE 157

CONDENSER DESIGN DATA

Gas	Maximum Pressure Desired in Condenser	Gas Temperature in Condenser, ° F	Leaving Hot Water Temperature, ° F: Best Condenser Design	Leaving Hot Water Temperature, ° F: Average Condenser Design
Steam	28 in. vacuum	101.2	97	93
Steam	27 in. vacuum	115.1	110	105
Steam	26 in. vacuum	125.9	120	114
Ammonia	185 lb gage head pressure	96.0	92	88
Carbon dioxide	1030 lb gage head pressure	86.0	83	81
Methyl chloride	102 lb gage head pressure	100.0	96	92
Dichlorodifluoromethane	117 lb gage head pressure	100.0	96	93

Courtesy of A.S.H.V.E.

wet-bulb temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

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 154 although unusual designs may operate outside these ranges.

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various processes of the cooling equipment are:

Compressor refrigeration	220 to 270	Btu per minute 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.

See Chapter XXIV for approximate costs of cooling towers.

Chapter XVII

DRYING

Drying, in its broader sense, is the removal of a volatile liquid, usually water. The common usage of the word drying refers to the removal of water or other liquid or solvents, by evaporation from a solid material. The process of drying air has been discussed under the subject of dehumidification and absorption. When the solid to be dried contains large amounts of free water the actual drying process is often preceded by the removal of part of the water by mechanical means, such as centrifuging, settling or filtration. Removal by mechanical means is less expensive than removal by evaporation.

333. Classification by Method of Heat Application

Driers may be classified as follows:

1. Direct heat (radiation or convection) with heating coils.
2. Indirect heat with gravity circulation, mechanical circulation with distributing ducts, induced circulation by water sprays, steam jets or air jets.
3. Radiant heat. (Infra-red lamps).

Driers may be classified according to the methods of moisture removal as follows:

1. By ventilation, either gravity or mechanical.
2. By condensation, either direct cold surface or cold spray.
3. By absorption with calcium chloride, lithium chloride, etc.
4. By adsorption on silica gel, activated alumina, etc.
5. By transpiration, through cloth walls.

An adiabatic drier is one in which all heat is supplied by air externally heated. The air temperature decreases as heat is transferred to the material being dried.

A constant temperature drier is one in which part or all of the heat is supplied by steam coils or other heat sources within the drier itself. Driers using little air for heating with a high temperature drop are difficult to keep at uniform temperatures; the more air used, the easier it is to secure accurate control of temperature and humidity.

334. Drying Stages

The theory of the drying cycle may be divided into two stages:

1. Constant rate period
2. Falling rate period

These are based on an assumption of uniform air velocity and distribution at a constant temperature and humidity.

335. Constant Rate Period *

The constant rate period occurs while the material being dried is still very wet, and continues as long as the water in the material comes to the surface so rapidly that the surface remains thoroughly wet, and evaporation proceeds at a constant rate, precisely as from a free water surface. The material assumes a temperature corresponding to the wet-bulb temperature of the surrounding air, or slightly higher, due to radiation and conduction from dry surfaces adjoining the material. The constant rate period continues until the moisture no longer comes to the surface as fast as it is evaporated. This point is called the *critical moisture content.*

336. Falling Rate Period

As the drying proceeds, a period of uniform falling rate is entered. During this period, the surface of the material is gradually drying out, and the rate of drying falls as the remaining wet surface decreases in area. This period is also known as unsaturated surface drying.

* *Courtesy of A.S.H.V.E.*

As drying continues, the surface is completely dry and the water from the interior evaporates and comes through the surface as vapor. As the plane of water recedes, the diffusion of the vapor becomes more difficult and hence the period is known as *varying falling rate period,* or sub-surface drying.

As drying progresses, another point, called the *equilibrium moisture content,* is reached. At this point the vapor pressure of the moisture in the air equals the vapor pressure of the moisture in the material, and drying is complete.

337. Moisture in Substances

Water is associated with solids in the following manner:

1. Water containing solids in solution
2. Water containing solids in suspension
3. Water occupying and permeating the pores in inorganic substances, and the cells in organic substances
4. Water adhering to surfaces
5. Crystalline water (ice)
6. Hygroscopic moisture
7. Water causing fungus growth (still air generally above 75% relative humidity)

Moisture may be classified as follows:

1. Capillary or free
2. Hygroscopic or combined.

Free moisture occupies the capillary spaces between fibers of materials. The loss of this moisture only changes the weight of the material. Combined or hygroscopic moisture, when removed, changes physical properties of the material. The quantity of hygroscopic moisture is limited, and this limit is called the *fiber saturation point.*

Care must be exercised to avoid shrinkage or hardening when material is dried below this point. All hygroscopic substances have a definite equilibrium moisture content dependent on temperature and humidity. Materials are often dried to a lower moisture content than the surrounding air and allowed to regain the necessary moisture after leaving the drier. The fibers used in textiles are very hygroscopic, i.e., they readily absorb or release moisture, maintaining a moisture regain normal or not in proportion to that of the surrounding air. Moisture affects the flexibility, pliability or brittleness of substances. Tobacco is hygroscopic. Paper becomes dry and brittle or soft and pliable with changes in the atmospheric moisture. Excess moisture causes candy to become sticky. Sometimes drying may be accomplished by simply heating the air, so as to lower the relative humidity and increase its capacity for absorbing moisture. Some substances which must be dried cannot be subjected to heated air. These are substances which melt at temperatures above, e.g., 90° F. Glue, gelatin and matches are among these. The drying of such products in summer involves the double problem of reducing the relative humidity of the air and at the same time preserving a temperature equal to atmospheric.

338. Drying of Matches

In the manufacture of a product containing glue, e.g., matches, a dry-bulb temperature not over 72° F is recommended in order to keep the viscosity of the glue at the proper point, and to avoid conflagration. However a low humidity must be maintained for satisfactory drying. Research and experience have determined that 50% relative humidity corresponding to a 52° F dew point is ideal for the progress of matches through the machines. At a dew point of 50° F, 4.4 grains of moisture is contained in a cubic foot of air. Research has established that a pound of water must be removed from 50,000 matches. Every cubic foot of air introduced will have to absorb about 0.7 grain of moisture, so that the air introduced may contain only 3.8 grains per cubic foot, which is equivalent to a dew point of 48° F. The reduction in dew point is a refrigerating process. Air cooled by refrigeration to 48° F saturated is heated by steam coils to 72° F and then introduced into the match machines.

A match machine delivering about 7,000 gross per hour requires about 20 tons of refrigeration and a net air circulation of 4,500 cubic feet per minute, 25% being fresh air and the balance being recirculated air.

339. Drying of Lumber

Moisture is a natural concomitant of wood. Kiln drying removes part of the moisture, the natural moisture remaining as sap. In the sap are dissolved various chemicals. The percentage of wood sawdust soluble in cold water varies from 1.77% dry weight of balsa wood to 10.6% of western larch. Moisture is held in wood in three ways

(1) Moisture in cell cavities
(2) Moisture in cell walls
(3) Free moisture between cell walls.

The time consumed in drying wood should be enough to avoid the too rapid drying causing *case hardening,* uneven drying, shrinking, checks, warping, etc. The period of time spent in the kiln by some woods, dependent upon species and size, is prolonged for ten days to two weeks. About 2.5 lb of steam per pound of moisture removed from soft wood is required to reduce the moisture content to 5% of bone-dry weight. In the slow drying of hard woods at low temperature, 15 lb steam per pound of moisture is often required. Air in the kiln is recirculated at the rate of about 24 changes per hour.

Humidity is regulated in the drying kiln by returning the air through a spray of water thermostatically controlled at a constant dew-point temperature; the air then is forced over steam coils or some other heating surface, heated to a higher temperature, and recirculated over and between the planks of lumber.

TABLE 158

HYGROSCOPIC MOISTURE IN LUMBER

Water Content Per Cent Dry Weight	Relative Humidity
3	10
6	30
9	50
14	70
22	90
30	100

Rate of absorption or evaporation depends upon species.

340. Regulated Drying with Conditioned Warm Air

Substances may be dried by passing over them a blast of superheated steam or warm air. In this topic the warm air blast only is discussed. The warm air blast may be

(1) vented to the atmosphere and wasted as the air becomes more and more saturated; it must be recognized that the more nearly air approaches saturation the lower is its ability to remove moisture.

(2) returned to the heater after it has been passed through a spray of cooler water to condense the accumulated moisture. If the spray of water is thermostatically controlled to give a set dew point, this system may be called *Regulated Drying.*

The drying may be a continuous process, the substance being placed on conveyors which progress at a set rate through the drying chamber; the drying may be a discontinuous or *bulk* process, the substance being placed in a chamber and allowed to remain there until finished. The latter process is discussed in this topic, as it offers a more easily analyzed plan. Continuous drying calculations are simply the summation of the calculations for a series of bulk drying processes.

341. The Product in the Storeroom

Research and practical experience usually determine the retained moisture permissible in a product after drying. If the product is wood the hygroscopic table in a previous topic gives the effects at an air condition of 30% relative humidity (the relative humidity in the drier is not permitted to drop below 30%). After the product leaves the drier it must be stored before fabrication. Allowance must therefore be made for some regain of moisture. Research and experience have determined that in the case of furniture a regain up to 8% will meet average conditions in the commercial field, e.g., in the retail store, or the home. In other words, furniture containing 8% moisture will give the least trouble from shrinkage in winter or swelling in summer. In practice the center of the log would be reduced to 8%, while the surface would be reduced to 5%. Surface regain would occur during the normal period of storage, except in extremely cold weather.

Humidity control in the factory or storeroom is needed only to the extent where it prevents stock from regaining much above 8% or whatever experience determines.

Wood finishings, lacquers and varnishes fill the pores of the wood and slow down regain. Cheap furniture usually has no finish on the unexposed surfaces; this is a source of trouble, such as swelling or shrinking. Furniture given at least four coats of varnish, will not absorb moisture through the varnished surface.

Research and experience determine the balance between average atmospheric moisture and average moisture content for

TABLE 159

GAS COMBUSTION CONSTANTS[a]

Gas	Chemical Formula	Molecular Weight	Cubic Feet per Lb	Heat of Combustion Btu per Pound: Gross	Heat of Combustion Btu per Pound: Net	Pounds per Pound of Combustible, Required for Combustion: O_2	Required for Combustion: N_2	Required for Combustion: *Air*	Flue Products: CO_2	Flue Products: H_2O	Flue Products: N_2
Carbon	C	12.000	....	14,140	14,140	2.667	8.873	11.540	3.667	....	8.873
Hydrogen	H_2	2.015	187.723	61,100	51,643	7.939	26.414	34.353	....	8.939	26.414
Oxygen	O_2	32.000	11.819	....	....	....	....	....	....	....	....
Nitrogen	N_2	28.016	13.443	....	....	....	....	....	....	....	....
Carbon Monoxide	CO	28.000	13.506	4,369	4,369	0.571	1.900	2.471	1.571	....	1.900
Carbon Dioxide	CO_2	44.000	8.548	....	....	....	....	....	....	....	....
Methane	CH_4	16.031	23.565	23,912	21,533	3.992	13.282	17.274	2.745	2.248	13.282
Ethane	C_2H_6	30.046	12.455	22,215	20,312	3.728	12.404	16.132	2.929	1.799	12.404
Propane	C_3H_8	44.062	8.365	21,564	19,834	3.631	12.081	15.712	2.996	1.635	12.081
Sulfur Dioxide	SO_2	64.060	5.770	....	....	....	....	....	....	....	....
Water Vapor	H_2O	18.015	21.017	....	....	....	....	....	....	....	....
Air	....	28.900	13.063	....	....	....	....	....	....	....	....

[a] All gas volumes corrected to 60° F and 30 in. mercury barometric pressure dry.

Table 160
DRIERS FOR EVAPORATION OF WATER

Type	Kind	Materials Handled	Means of Handling	Temp. Range °F	Heat Supply	Uses and Remarks
Batch or Intermittent	Compartment	Paper, Leather, Yarns, Lumber, Foodstuffs	Suspended, Truck, Tray	80 to 180	Steam Coils, Air, Electricity	When production does not warrant continuous drier
	Agitated	Chemicals too sticky for Rotary Drier	Shoveled into Drum or Pan	100 to 330	Water, Steam Jacketed, may have Vacuum on top	Where dust must be saved
	Vacuum	Chemicals, Explosives, Pharmaceuticals, Food Products	Tray Basket, Tumbling Drum	80 to 300	Water, Steam	Cost of operation high, for expensive materials
Continuous	Tunnel	Ceramics, Chemicals, Lumber, Food Products	Truck, Tray, Belt	100 to 350	Steam Coils, Air, Electricity, Products of Combustion	For High Production
	Rotary	Bulk	Cascades through	80 to 500	Air, Steam, Products of Combustion	Where material will stand rough handling and is not subject to balling up
	Drum	Liquids, Slurries	Flowed on Drum, Dry Material Scraped off	to 310	Steam, may have Vacuum on Top	Hygroscopic materials dried with vacuum, and packed immediately
	Cylinder	Paper, Textiles, Chemicals	Continuous Sheets, Endless Chain Belt	to 350	Steam inside of Drum	Where material comes in sheets or rolls, and will stand direct contact with heating surface
	Festoon	Paper, Chemicals	Continuous Sheets, Suspended on Metal Screens	to 200	Air, Steam Coils	Where one side cannot come in contact with supports until dry
	Tower or Column	Grains, Sand	Falls through by Gravity	125 to 250	Air, Steam Coils	Where headroom is available
	Spray	Solutions over 30% Solids	Sprayed into Chamber	120 to 350	Air, Products of Combustion	Drying is almost instantaneous
	Induction	Metals, for removal of traces of Water	Placed in High Frequency Field	to 400	Electricity	Where heating of metal from inside out is important

Courtesy of A.S.H.V.E.

TABLE 161

DRYING TIME AND CONDITIONS FOR REPRESENTATIVE MATERIALS

Material	Temperature, ° F	Per Cent Relative Humidity	Drying Time
Apples	140–180		6 Hr
Armatures Varnish	200		2.5 Hr
Banana Food ¼ in. Thick	140		4–6 Hr
Barrels	300		15 Min
Beans	140		18 Hr
Bedding	150–190		
Blankets	120		40 Min
Brake Lining	325		12 Min
Brick continuous	350 to 90		24 Hr
Briquets	1100		108 Min
Cabbage, Raw	150		4.5 Hr
Candied Peel	165		2 Hr
Casein	180		5 Hr
Cereals	110–150		
Ceramics before firing	150	70 to 20	24 Hr
Chicle	95–100		
Coco-fiber mats	170–210		10 Hr
Cocoanut	150–200		4–6 Hr
Coffee	160–180		24 Hr
Conduit (Enamel)	400 Max		2 Hr
Cores, Oil sand for molding			
½—1 in. thick	300		30 Min
Black sand with goulic binder about 0.6 of time: 3 in. thick	480		2.5 Hr
Black sand with goulic binder about 0.6 of time: 8 in. thick	480		4.5 Hr
Black sand with goulic binder about 0.6 of time: 16 in. thick	700		10 Hr
Cores, Crank case (in continuous ovens)	525–600		2–3 Hr
Cores, Radiator (in continuous ovens)	275–450		1.5 Hr
Cornstalk Board	150		2 Hr
Cotton Linters	180		
Enamels, synthetic			
Finish coat on autos	225		2 Hr + Air Dry
Ice boxes all metal (white)	290–425		1 Hr
Ice boxes wood inside (white)	225		3 Hr
Enamel, not synthetic			
Fence posts green	200		1 Hr
Golf balls (white)	90–95	40–50	18–36 Hr
Small parts (auto) black	450		1 Hr

Courtesy of A.S.H.V.E.

TABLE 161 (*continued*)

Material		Tempera-ture, ° F	Per Cent Relative Humidity	Drying Time
Steel furniture		225–300		30–350 Min
License plates		250		1.5 Hr
Feathers		150–180		
Films, Photographic		85–110		20–30 Min
Fruits and Vegetables		140		2–6 Hr
Furs		110		
Gelatin		110		
Glue, bone, thin sheets on wire trays		70–90		6–9 Days
Glue, skin		70–90		2 Days
Glue size on furniture		130		4 Hr
		150		
Gypsum board ⅜ in. thick	Start Wet	350		60 Min
	Finish	275		
Gypsum block		350–190		8–16 Hr
Hair felt		180–200		
Hair goods		150–190		1 Hr
Hanks on poles		120		2 Hr
Hats, felt		140–180		
Hides thin leather		90		2–4 Hr
Hides heavy		70–90		4–6 Days
Hops		120–180		
Ink printing		70–300		
Japan beds		300		1.5–2 Hr
Japan cash register		300–450		1.5 Hr
Japan metal shelving		200		30 Min
Knitted fabrics		140–180		
Leather mulling		78–95	85	
Leather thick sole		90	70	4 Days
Leather uppers		80		2–3 Days
Linoleum varnish		110–145	10–30	6–10 Hr
Lithographing on tin color work		250–270		18–25 Min
Lithographing on tin Japan		350		
Lumber green hardwood		100–180		3–180 Days
Lumber green soft wood		160–220		2–14 Days
Macaroni		90–110		7.5–8 Hr
Matches		140–180		
Matrix		350		15 Min
Milk and other liquid foods spray dried		135–300		Instantaneous
Millboard sheets		95		10 Hr
Molds green sand C.I. flasks (one surface only exposed)	8 in. thick	600		6 Hr
	13 in. thick	700		13 Hr
Motors, field coils		180		6 Hr

TABLE 161 (*continued*)

Material	Temperature, ° F	Per Cent Relative Humidity	Drying Time
Motors, stators	250		6.5 Hr
Noodles	90–95		
Nuts	75–140		24 Hr
Oil cloth	150		
Paint, wood wheels	150	35	8–24 Hr
Paint, on sheet metal	350–140	22–30	2.5 Hr
Paper, machine dried	180		
Paper, air dried	90–200		
Paper wall, ground coat	140		3 Min
Paper wall, varnished	140–160	45	15 Min
Paper cardboard, spirit varnish	150		1–2 Min
Peaches	135		26 Hr
Pears	140		24 Hr
Peas	150		6 Hr
Potatoes sliced	85		4 Hr
Potatoes steamed	170		6.5 Hr
Prunes	140		
Rags	180		
Ramie fiber	140		10 Hr
Rice	150		
Rock wool insulation	300		8 Hr
Rubber	85–90		6–12 Hr
Rubber reclaimed	140–200		1–2 Hr
Rugs	190		4–8 Hr
Salt	350		Rotary Drier
Sand loose 1 in. deep	300		10–15 Min
Sausage casings	110		5 Hr
Shade cloth	240		1–2 Hr
Shirts	120		20 Min
Soap	100–125		12–72 Hr
Starch	180–200		1–4 Hr
Stock feed mixed	180–220		20–30 Min
Storage battery plates	100–110	90 for	24 Hr
	250	Low for	6 Hr
Sugar	150–200		20–30 Min
Tannin and other chemicals (spray dried)	250–300		Instantaneous
Terra Cotta (air drying in conditioned room)	150–200		12–96 Hr
Tobacco leaves	85–130		12 Hr
Tobacco stems	180–200		12 Hr
Varnish refrigerator boxes	110	35	5–7 Hr

TABLE 161 (*continued*)

MATERIAL	TEMPERATURE, ° F	PER CENT RELATIVE HUMIDITY	DRYING TIME
Varnish steering wheels	110–140	25–35	Overnight
Veneer ¼ in. 3-ply	120–130	35–40	6–8 Hr + 2 Hr acclimation
13⁄16 in. 5-ply	120–130	35–40	16–18 Hr + 4 Hr acclimation
1¼ in. 5-ply	120–130	35–40	20–24 Hr + 4 Hr acclimation
Vitreous Enamel sheets before firing	170		
Wallboard pasted plywood	300		15–20 Min
Wallboard fiber insulating, roller type drier	300–385		2½–3 Hr
Wallboard fiber insulating, truck type drier	300–385		24–48 Hr
Walnuts	100		24 Hr
Wheat, corn, oats, rice, barley	180		
Wire cloth Japan	200		20 Min
Wool	105		

given temperatures and humidities. The equilibrium moisture content of certain woods at 8% is equivalent to 40% relative humidity at a temperature of 70° F. At 70° F dry-bulb, 40% relative humidity, the dew point is 44° F. Therefore a humidifying system will be required in the storeroom in order to maintain this dew point in winter. In the summer when outside air is at a high humidity, e.g., 80° F dry-bulb and 75° F wet-bulb, equivalent to 80% relative humidity the storeroom must be heated to 90° F to reduce conditions to 50% relative humidity.

342. Rapid Estimating Rules

The quantity of steam required to evaporate a pound of water will vary from 1.5 lb to 3 lb per pound of water evaporated. The temperature will drop 8.5° F per grain of water evaporated per cubic foot of air or approximately 0.62° F per pound of air at any temperature. Air will drop 55° F per cubic foot for each Btu extracted. Generally, air will absorb from 2 to 5 grains per cubic foot in one passage through an air dryer, depending on the temperature and the degree of contact with the material.

CHAPTER XVIII

COOLING METHODS AND REFRIGERATION

The early Greek poet, Simonides, while at a banquet, observed that the liquor served to the other guests was cooled by snow. Whereupon he expressed his dissatisfaction in the following ode:

"The cloak with which fierce Boreas clothed the brow
  Of high Olympus, pierced ill-clothed man
While in its native Thrace; 'tis gentler now,
  Caught by the breeze of the Pierian plain.
Let it be mine: for no one will commend
  The man who gives hot water to a friend."

History's pages also show that the ancient Egyptians knew the secret of cooling by evaporation, as practiced by the native of India today—filling with water shallow trays of porous material placed on beds of straw, and leaving them exposed to the night winds, with the result that dawn finds a thin film of ice formed on the surface.

On a very early page we find that the Emperor Nero had slaves bring snow down from the mountains to cool his wines. Alexander the Great had trenches dug for storing snow. Hundreds of kegs of wine were cooled there, with the result that his phalanxes entering battle the next day didn't care much what became of them, just so it was a good battle.

Marco Polo, the great Italian navigator, brought recipes for water and milk ices from Japan and China in the thirteenth century.

When Catherine de Medici left Florence, Italy, to go to France, in the sixteenth century, she took with her the best of the chefs to make sure that she would be supplied with frozen creams and ices every day.

Sir Walter Scott told how Saladin, leader of the Mohammedan armies, sent a frozen sherbet to Richard, the Lion Hearted, much to the amazement of that doughty monarch.

As Lord Bacon commented in his *Sylva Sylvarum:*

> "Heat and cold are Nature's two hands whereby she chiefly worketh, and heat we have in readiness in respect of the fire, but for cold we must stay till it cometh or seek it in deep caves or mountains, and when all is done, we cannot obtain it in any great degree, for furnaces of fire are far hotter than a summer's sun, but vaults and hills are not much colder than a winter's frost."

Bacon knew what a useful thing it would be if man could have the same command of cold as of heat. Scientist that he was, he undertook experiments into its possibilities. This led to unfortunate results, as he caught his death of cold by alighting from his carriage one winter day and stuffing snow into a chicken to see if it would keep.

In 1755 Dr. William Cullen invented the first machine which produced ice by purely mechanical means, his achievement being followed by those of Vallance, of France, (1824) and Jacob Perkins, an American, then residing in England, who is given credit for the forerunner of the modern compression apparatus, his model being patented in England in 1834, with ether as the refrigerant employed. Other early workers in this field of science were Professor A. C. Twining, of New Haven, Connecticut, and Dr. John Gorrie, of Appalachicola, Florida.

In the years of 1873–75 the first successful ammonia compression machines were introduced by C. P. G. Linde, of Germany, and David Boyle, of the United States. From 1875 to 1890 many new forms of apparatus were produced and certain improvements were made.

In 1890 a natural ice shortage started increased production of artificial ice-making, cold storage plants, and commercial refrigerating units. In 1920 began the period of new building construction, creating tremendous new demands of the industry for comfort cooling, industrial air conditioning and domestic mechanical refrigeration. The late economic depression starting in 1930 did not affect the growth of the industry to any extent.

343. Cooling and Dehumidifying

Dehumidifying and cooling are closely related in air-conditioning engineering. Comfort is generally produced in summer by a reduction in both temperature and humidity. Whenever air is cooled below the dew point a reduction in moisture content takes place. There is a rise in temperature whenever moisture is removed from the air by adsorption or absorption. Thus cooling and dehumidification are associated and not to be considered as two separate problems, although each process can be accomplished separately. The addition of heat (whether sensible or latent) to air or abstraction of heat from it, requires not only a medium of the temperature or vapor pressure required to produce a flow of heat or moisture, but also sufficient contact between the air and the medium to obtain the desired final condition. The medium may be solid or liquid. It may be used directly, as in a brine or water spray, or indirectly, as with a steam radiator or direct expansion cooling coil. The contact occurs through the use of exposed surfaces with which the molecules of air are brought into intimate contact, thereby producing the heat interchange. This chapter discusses in detail the engineering and economic principles involved in the application of refrigeration and cooling.

344. Definitions *

While this chapter is principally concerned with refrigeration, the basic methods of otherwise producing the required difference in temperature or vapor pressure for cooling or dehumidification are briefly defined below.

Cooling of air involves its reduction in temperature due to the

* *Courtesy of A.S.H.V.E.*

abstraction of sensible heat. It is always a result of contact with a medium held at a temperature lower than that of the air. Cooling may be accompanied by moisture addition (evaporation), by moisture extraction (dehumidification), or by no change of moisture content whatever. Such moisture change, if present, is considered as a secondary or by-product effect. As previously stated, the medium may be directly in contact with the air (as water, brine, or ice), or indirectly through a barrier wall (as cooling surface). When the latter method is used, and the surface temperature is held above the air dew point, only cooling occurs and not moisture interchange.

Evaporative cooling involves the adiabatic exchange of heat between air and a water spray or wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger. No heat is added to or abstracted from the medium (water), which is continually recirculated. Cooling of the air occurs due to the temperature difference between entering air, and water at the wet-bulb temperature. Humidification occurs as a result of the vapor pressure exerted by the water which is higher than that corresponding to the entering air dew point. Since this is an adiabatic exchange, the enthalpy of the air remains constant, while the dew point rises and the dry-bulb falls, and the loss of sensible heat exactly equals the gain in latent heat (neglecting radiation losses). The maximum available temperature reduction is the total difference between entering dry- and wet-bulbs (wet-bulb depression). Equipment achieving the complete reduction is termed *completely* saturating or 100% efficient, since the air leaves in a saturated state. Equipment utilizing only a portion of the wet-bulb depression is termed *partially* saturating.

Evaporative cooling is being used advantageously in many parts of the country. It is particularly applicable (1) in districts where the relative humidity is normally low during the cooling season, and (2) in applications where the cooling load is principally a sensible load.

Dehumidification of air, in its broadest connotation, means simply the removal of moisture. Usage in the art has restricted the application of the term, so that the former broad meaning is now properly covered by the complementary names dehumidification and dehydration. *Dehumidification* usually refers to the *condensation* of water vapor from air due to its contact with a chilled medium (see Cooling). This type of heat exchange invariably includes temperature reduction due to removal of sensible heat, which reduction may be considered a by-product effect.

Dehydration refers specifically to the removal of water vapor from air due to its contact with a dehydrating agent. The primary distinction between dehumidification and dehydration is the vapor pressure exerted at the surface of the contacting medium. In the case of dehumidification, this surface vapor pressure is always the same as that which would be exerted by a body of water (or ice) at that same surface temperature. In the case of a dehydrating agent, the surface vapor pressure is always *lower* than that exerted by water at the same temperature, and the effectiveness of the medium as a desiccant is largely a function of the amount by which this vapor pressure can be lowered at the working temperature involved.

Thus it is evident that the primary function of a dehydrating agent is to establish a vapor pressure difference between the air and the medium in order to secure thereby a removal of moisture (latent heat) from the air. In the simplest type of process, no heat is abstracted from the medium itself, and the process is essentially an adiabatic one in which the latent heat lost by the air is converted to sensible heat which raises the air temperature by an equivalent amount. This process is therefore an energy exchange, similar to, but the reverse of, adiabatic saturation.

Combination Methods. It is evident that two or more of the above processes—cooling, evaporative cooling, dehumidification and dehydration—may be combined by the proper application of interchangers in sequence. Such combinations are dictated by the availability of prime sources of energy and the economic justification of each.

345. Evaporative Cooling

The cooling of air by passing it through a continuous circulating spray of water that is neither cooled nor heated by outside means, is called evaporative cooling. Air enters in an unsaturated condition and evaporates part of the water at the expense of sensible heat. As the process is an adiabatic heat 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. The resulting air temperature is governed entirely by the entering wet-bulb temperature of the outside air. Often it is possible to reduce the dry-bulb temperature by as much as 20°. There are times, however, when the relative humidity of the outside air is too high, so that the air temperature may only be reduced 2 to 3°. For many applications, whenever the outdoor wet-bulb temperature is 60° F or below, effective cooling and indoor comfort can be maintained by evaporative cooling. In some localities where the wet-bulb temperature rarely exceeds 60° F, evaporative cooling suffices at all times without refrigeration or other types of cooling. There is a vast difference in the wet-bulb conditions encountered in typical mountain and coastal regions. The Air and Refrigeration Corporation has compiled a very useful chart which shows that evaporative cooling could be successfully used, e.g., at Denver, Colorado during all but 800 hours in a season, and at Hartford, Connecticut, during all but 2200 hours. Only during these times would refrigeration be required. In contrast, a surface cooler, without evaporative spray cooling, would call for refrigeration to deliver 60° F air during 2750 hours in a season at Denver and 3000 hours at Hartford. This emphasizes one factor in favor of the spray system of cooling.

In most air-conditioning applications for human comfort the design conditions call for capacity to maintain 80° F dry-bulb and 50% relative humidity when outdoor dry- and wet-bulb temperatures are at the expected maximum for the given territory. The above temperature and humidity would call for a dew-point temperature of 59.7° F in the conditioned space. The dew-point temperature of the air supplied to the space, usually termed the apparatus dew point, must be sufficiently below the above condition to absorb the moisture from people and other sources. For most applications, where there is no unusual internal latent heat load, this would be about 58° F.

Since the simple recirculation of water through the spray chamber will reduce air to approximately the prevailing outdoor wet-bulb temperature, it is obvious that no refrigeration or other external source of cooling would be required to maintain the above conditions whenever the outdoor wet-bulb temperature is at 58° F or below.

346. Refrigeration

The heating of air offers no particular difficulty but the problem of cooling air requires some special means of cooling, usually something of a heat-removing nature, such as cold well water, ice or mechanical refrigeration. As the temperatures required in cooling air seldom go below 45° F and may be kept as high as 50° F if necessary, it follows that the refrigeration used needs quantity rather than extremely low temperatures.

Perfectly satisfactory air conditioning may be accomplished with an adequate supply of well water provided that this water has a temperature close to 50° F throughout the summer season. Ice cooling is used in some cases although it usually is more troublesome than either well-water or mechanical refrigeration.

347. Well-water Cooling

Well-water cooling is often a gamble unless the capacity of the well can be accurately determined beforehand. If this can be done, and if the temperature is at or near 50° F, the well may be used for air cooling without difficulty. The reason that water as low as 50° F must be obtained is that it is necessary to reduce air to approximately that temperature if adequate dehumidification is to be obtained. The quantity of water required will be in accordance with the load and with the temperature rise permissible in the water. Thus, if a rise of 10° F in the temperature of the water can be accommodated, then each pound of water will pick up 10 Btu while, if only 5° F can be used, the water will pick up per pound only 5 Btu. As a ton of refrigeration is equivalent to the removal of

288,000 Btu per day
12,000 Btu per hour
or
200 Btu per minute

it follows that the quantity of water in pounds per minute required for a given amount of cooling will be

$$\frac{\text{Tons of Refrigeration} \times 200 \text{ Btu}}{\text{Temperature Rise in Water}}$$

and that the number of gallons per minute required will be

$$\frac{\text{Tons of Refrigeration} \times 200 \text{ Btu}}{\text{Temperature Rise in Water} \times 8.33 \text{ lb}}$$

since there are 8.33 lb of water to a gallon.

Thus a 50 ton installation using water with a 10 degree rise in passing through the air washer or coils will require

$$\frac{50 \text{ tons} \times 200 \text{ Btu}}{10^\circ \text{ F} \times 8.33 \text{ lb}} \quad \text{or} \quad 120 \text{ gpm}$$

This water, after passing once through the cooling apparatus, must be wasted, since its temperature has risen 10° F and it is not possible to utilize it for further cooling.

348. Cooling with Ice

With ice cooling the ice is purchased and stored in an ice storage room; as required the cakes are introduced into a spray chamber, over which the water is sprayed.

The melting of the ice by the water results in the ice picking up 144 Btu per pound melted for the latent heat of fusion and this heat comes from the water, reducing the temperature thereof. In such systems the water may be re-used and the heat picked up from the air will be removed from the water by the melting ice so that the water is reduced back to its original cooling temperature of around 50 degrees. While ice cooling is cheaper to install than mechanical refrigeration, its operating expense is likely to run higher due to the cost of the ice and the uncertainty as to when it will be required. The amount of ice used varies with the outside temperature and with the occupancy, especially in theaters where the audience will vary greatly during different hours of the day.

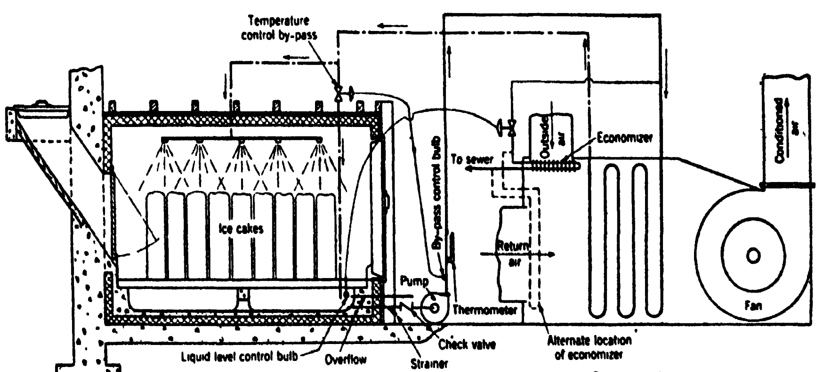


FIGURE 64

Typical Ice System Showing Bunker Details

349. Cooling Air and the Refrigeration Load

In cooling air the refrigeration load consists of two distinct items; first, the quantity of cooling to be done on the air itself, and, second, the quantity of vapor to be condensed out of the air (if any). If the cooling process or temperature to which the air is cooled brings the air below its dew point a certain quantity of vapor will have to be condensed and the latent heat of this condensation will constitute a load on the refrigeration plant because the refrigeration must take up the latent heat of condensation.

As the specific heat of air is 0.24 Btu per pound per degree of temperature change, it follows that the refrigeration load for cooling a certain number of pounds of air per hour through a given temperature range may be expressed by the formula:

$$\frac{\text{quantity, in pounds, of air cooled} \times \text{number of degrees cooled} \times 0.24}{12{,}000 \text{ Btu}} = \text{tons of Refrigeration}$$

and, of course, the quantity of air expressed in pounds per minute would modify the formula to:

$$\frac{\text{quantity, in pounds, of air cooled} \times \text{number of degrees cooled} \times 0.24}{200 \text{ Btu}}$$

which will give the same result.

350. Dehumidifying Air and the Refrigeration Load

If the air is reduced below the dew point and a certain quantity of moisture or vapor is removed from the air by condensing the excess moisture, then the refrigeration load may be determined approximately from the formula

$$\frac{\text{pounds of air} \times \text{grains water removed} \times 1{,}000 \text{ Btu}}{7{,}000 \text{ grains} \times 12{,}000 \text{ Btu}} \text{ equals tons of Refrigeration}$$

if the pounds of air is taken as pounds per hour; if the pounds of air is taken as pounds per minute, the formula becomes:

$$\frac{\text{pounds of air} \times \text{grains removed} \times 1{,}000 \text{ Btu}}{7{,}000 \text{ grains} \times 200 \text{ Btu}}$$

Refer to Chapter IV and Chapter IX for more accurate computations.

351. Examples in Approximate Calculations

Six thousand pounds of air per hour are to be cooled 40° F with a reduction of 70 grains per pound in the vapor content; how this is determined will be explained more in detail in the discussion of dehumidifying. How many tons of refrigeration capacity will be required?

First the refrigeration for cooling will be calculated: using the formula for cooling air and substituting the proper values gives

$$\frac{6{,}000 \text{ lb} \times 40° \text{ F} \times 0.24 \text{ Btu}}{12{,}000 \text{ Btu}} = 4.80 \text{ tons}$$

or, calculating on the minute basis and substituting in the minute formula, gives

$$\frac{100 \text{ lb} \times 40° \text{ F} \times 0.24 \text{ Btu}}{200 \text{ Btu}} = 4.80 \text{ tons}$$

which is exactly the same.

Now calculate for dehumidifying: using the hourly formula and substituting the proper quantities gives

$$\frac{6{,}000 \text{ lb} \times 70 \text{ gr} \times 1{,}000 \text{ Btu}}{7{,}000 \text{ gr} \times 12{,}000 \text{ Btu}} = 5.00 \text{ tons}$$

or, using the minute formula, gives

$$\frac{100 \text{ lb} \times 70 \text{ gr} \times 1{,}000 \text{ Btu}}{7{,}000 \text{ gr} \times 200 \text{ Btu}} = 5.00 \text{ tons}$$

and the total refrigeration required will be

For cooling the air	4.80 tons	(sensible heat)
For dehumidifying the air	5.00 tons	(latent heat)
Total Refrigeration	9.80 tons	(total heat)

352. Rapid Estimating of Comfort Cooling Systems

It is recommended that an approximate refrigeration capacity be estimated as described above, before accurate calculations are made. A rough idea of the refrigeration capacity enables the engineer to form a judgment as to the size and cost of the job and also offers a rough check on the calculations. Not only should the rough calculation be made before estimating the job, but also a quick estimate for a check-up after the calculations are made.

The following approximate data, if used with discreet engineering judgment, will prove of value in making quick estimates of the approximate tonnage of refrigeration required. The amount of the contract will naturally vary with the locality.

Refer to Chapter XXIV for estimated costs.

353. Approximate Refrigeration Capacity

Broadcasting studios—allow 1 ton per 800 sq ft.

Theaters less than 1500 seats—allow 1 ton for 15 seats.

Larger theaters—allow 1 ton for 17 seats.

Beauty Parlors—allow 1 ton per booth, plus 1 ton per 12 persons in waiting room.

Barber Shops—below ground—allow 1 ton per chair.

Barber Shops—above ground—allow 1 ton per chair.

Drug Stores with Lunch Counter—allow 1 ton per 125 sq ft floor area.

Drug Stores without Lunch Counter—allow 1 ton per 225 sq ft floor area.

Haberdashery Shops—allow 1 ton per 250 sq ft floor area.

Ladies' Dress Shops—allow 1 ton per 250 sq ft floor area.

1-Story Office Building—allow 1 ton per 450 sq ft floor area.

High Office Building—allow 1 ton per 600 sq ft floor area.

General Offices—allow 1 ton per 300 sq ft floor area.

Restaurants and Coffee Shops—allow 1 ton per 8 people.

Hotel Dining Rooms—allow 1 ton per 10 people.

Night Clubs—allow 1 ton per 6 people.

Cafeterias—allow 1 ton per 10 people.

Department Stores—It is necessary to know the number of people per hour passing through the store during the busiest season and the warmest weather. Department stores come in different classifications as follows:

1 person per 25 sq ft
1 person per 45 sq ft
1 person per 60 sq ft
1 ton per 10 people

Large department stores, doing a large volume of business, allow 1 person per 20 sq ft of floor area used for sales purposes.

Allow 1 ton per 10 people.

Five & Ten Cent Stores allow 1 ton per 250 sq ft floor area.

Allow 1 person per 10 sq ft of aisle space maximum.

354. Approximate Air Circulation

Allow 350 cu ft of total circulation per ton of refrigeration per minute, of which 25% should be fresh air and 75% recirculated air.

355. Refrigerating Machines

A refrigerating plant is an apparatus for maintaining a low temperature in a desired region by removing heat from that region and transferring it to a region of high temperature. A refrigeration machine is the converse of a heat engine, since it transforms work into heat, and then rejects the heat into a region of high temperature. Like the heat engine, the refrigerating machine employs a working fluid and causes this working fluid to undergo a thermodynamic cycle. Since the object of refrigeration is the transfer of heat, and not the performance of work, it is customary to consider the efficiency of a refrigerating system, as the ratio of the heat transferred to the work done. Since the mechanical equivalent of heat transferred is always several times as great as the work done, the efficiency of a refrigeration plant is usually greater than unity. A refrigerating machine may employ as a working fluid either a gas or a vapor. On shipboard air was formerly employed as a working fluid, since the leakage of air within the ship's confined engine room was not harmful. However the air system was heavier and not as efficient as the carbon dioxide vapor machine which ultimately replaced the air machine. At present, the carbon dioxide system is obsolete, having been replaced by systems using Freon and other safe refrigerants. Refrigerating systems may be divided into three classes:

1. Systems using a permanent gas as the working fluid. After being compressed and cooled the working fluid, e.g., air, is expanded adiabatically and its temperature reduced.

2. Vapor compression machines which liquefy a vapor in a condenser by application of pressure and cooling, and, by subsequent re-evaporation of the liquid under lower suction or evaporation pressure, the desired temperature is obtained.

3. Absorption systems, which generally use as a working fluid, a volatile liquid, such as ammonia, which is absorbed by water or some other liquid and then driven off under high pressure by heat in such a manner that it may be subsequently cooled and condensed. It is then evaporated under low pressure, thus producing a low temperature.

356. Refrigeration System Classification by Classes

The American Standard Safety Code for Mechanical Refrigeration as sponsored by The American Society of Refrigerating Engineers under the rules and regulations of the American Standards Association as Project B_9, in Section 4, classifies refrigeration systems as follows:

4.10 *Refrigerating Systems* shall be divided into classes, descriptive of the method employed for extracting heat as follows:

4.20 *Direct System* is one in which the evaporator is in direct contact with the material or space refrigerated or is located in air circulating passages communicating with such spaces.

4.30 *Unit System* is one which has been assembled and tested prior to its installation and which is installed without connecting any refrigerant containing parts.

4.40 *Indirect System* is one in which a liquid, such as brine or water, cooled by the refrigerant, is circulated to the material or space refrigerated or is used to cool air so circulated. Indirect Systems which are distinguished by the type or method of application are as follows:

4.41 *Indirect Open Spray System* is one in which a liquid, such as brine or water, cooled by an evaporator located in an enclosure external to a cooling chamber, is circulated to such cooling chamber and is sprayed therein.

4.42 *Indirect Closed Surface System* is one in which a liquid, such as brine or water, cooled by an evaporator located in an enclosure external to a cooling chamber, is circulated to and through such a cooling chamber in pipes or other closed circuits.

4.43 *Indirect Vented Closed Surface System* is one in which a liquid, such as brine or water, cooled by an evaporator located in a vented enclosure external to a cooling chamber, is circulated to and through such cooling chamber in pipes or other closed circuits.

4.44 *Double Indirect Vented Open Spray System* is one in which a liquid, such as brine or water, cooled by an evaporator located in a vented enclosure, is circulated through a closed circuit to a second enclosure where it cools another supply of a liquid, such as brine or water, and this liquid in turn is circulated to a cooling chamber and is sprayed therein.

4.45 *Indirect Absorptive Brine System* is an Indirect Vented Open Spray System in which the brine will chemically absorb the refrigerant in the system, and the chemical compound so formed in the solution will be stable at temperatures up to 100° F. The brine shall be of such quantity and concentration that it will absorb twice the total quantity of refrigerant in the system. An approved automatic device shall be provided for shutting down the system when the brine concentration becomes such that the brine will absorb only one and one-half times the total quantity of refrigerant in the system.

4.46 *Double Refrigerant System* is one in which a refrigerant

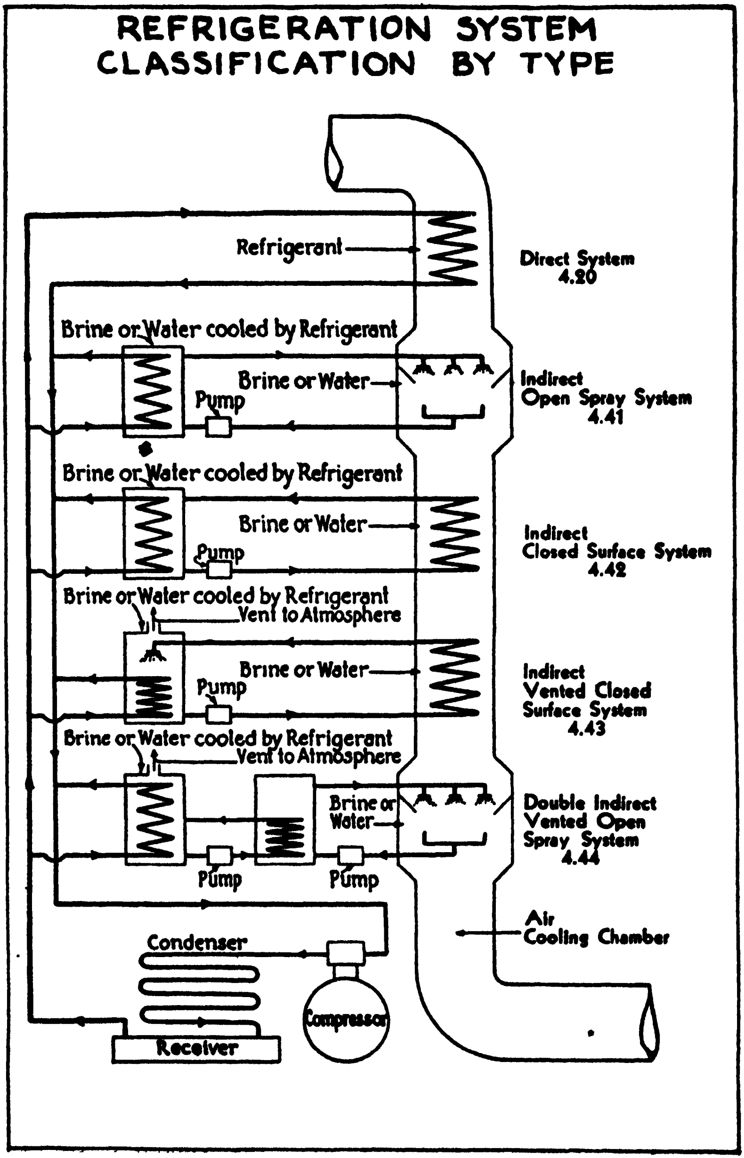

REFRIGERATION SYSTEM CLASSIFICATION BY TYPE
Refrigerant
Direct System 4.20
Brine or Water cooled by Refrigerant
Brine or Water
Pump
Indirect Open Spray System 4.41
Brine or Water cooled by Refrigerant
Brine or Water
Pump
Indirect Closed Surface System 4.42
Brine or Water cooled by Refrigerant
Vent to Atmosphere
Brine or Water
Pump
Indirect Vented Closed Surface System 4.43
Brine or Water cooled by Refrigerant
Vent to Atmosphere
Brine or Water
Pump
Pump
Double Indirect Vented Open Spray System 4.44
Air Cooling Chamber
Condenser
Compressor
Receiver

FIGURE 65

is used in the secondary circuit instead of brine or water. For the purpose of this Code, each circuit shall be considered as a separate Direct Refrigerating System.

The Direct and various Indirect Systems referred to above are illustrated in Figure 65.

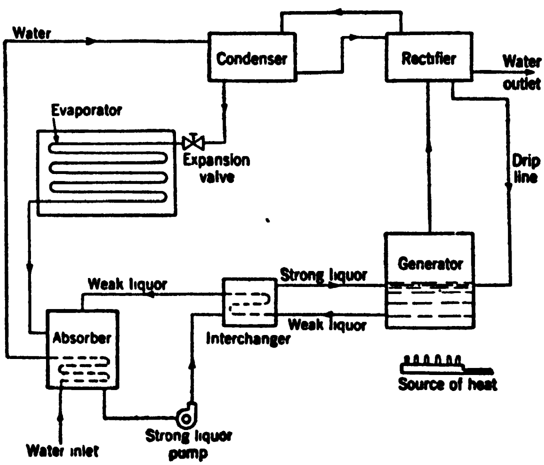


FIGURE 66

Closed Absorption System

357. Application of Refrigeration to Air Conditioning

When mechanical refrigeration is used it may be one of several types, such as the ordinary direct expansion system, the steam jet system or the water vapor system. The most common type is that employing direct expansion, where a power-driven piston-type compressor is used to compress the gas, a condenser to liquefy the gas, and a cooling coil in which the gas expands after entering the coil via an expansion valve. Various other accessories are added such as a receiver, oil separator, liquid trap, and so on. At the present time Freon is the refrigerant most commonly used in such systems, because of its comparative harmlessness, although ammonia is a more desirable refrigerating medium from a refrigeration standpoint.

The cooling accomplished in the water jacket of the compressor and in the condenser requires a supply of water which preferably should not have a temperature over 70° F and certainly not over 80° F.

The quantity of water usually required runs to 3.5 gal per minute per ton of refrigeration produced. This water cannot be re-used in the system unless the heat gain in the water, after passing once through the equipment, is removed by means of a cooling tower, spray pond, evaporative cooler or some other mechanism.

358. Reciprocating Compressors

This group covers all fields of application since it is applicable to either indirect or direct expansion systems and can be driven by steam, gas engines or electric motors. The quantity of condensing water required is less than for any other system with the exception of the centrifugal compressor (which requires the same quantity). Unit or packaged systems usually consist of reciprocating compressors. Reciprocating units are available in a wide range of sizes. However many manufacturers limit the size to not exceeding 100 tons for Freon-12. The most common refrigerants used with reciprocating compressors are dichlorodifluoromethane (Freon-12), carbon dioxide, methyl chloride, ammonia and sulfur dioxide.

Any packaged unit containing not more than ten pounds of refrigerant is limited by the safety code to Freon-12 which operates at a comparatively low pressure suited to the encased crank reciprocating compressor design. Carbon dioxide does not permit of encased crank, splash lubrication design and therefore is obsolete in modern automatic applications.

359. Refrigeration

The unit of refrigeration is the ton, which is equivalent to the heat required to melt one ton (2000 pounds) of ice from and at 32 degrees Fahrenheit in 24 hours.

One ton of refrigeration = 288,000 Btu per 24 hours
One ton of refrigeration = 12,000 Btu per hour
One ton of refrigeration = 200 Btu per minute

360. Weight of Refrigerant per Ton of Refrigeration

The weight of refrigerant handled per minute to produce one ton of refrigeration will depend upon the refrigerating effect per pound of refrigerant. This weight is equal to 200 divided by the refrigerating effect per pound and may be expressed by the formula:

$$W = \frac{200}{H_1 - h_2} \tag{1}$$

where W = weight of refrigerant per ton per minute
200 = Btu per minute equivalent to one ton of refrigeration
H_1 = heat content of vapor leaving the evaporator, Btu
h_2 = heat of the liquid refrigerant ahead of expansion valve, Btu

361. Cubic Feet Circulated per Ton of Refrigeration

The theoretical volume of refrigerant that must be circulated per minute to produce one ton of refrigeration is equal to the weight of refrigerant multiplied by the specific volume of the vapor at suction pressure. This may be expressed as follows:

$$V = \frac{200v}{H_1 - h_2} \tag{2}$$

where $V =$ cubic feet circulated per minute per ton
$v =$ specific volume (cu ft per lb) of vapor leaving the evaporating coils.

The actual displacement of a compressor to produce one ton of refrigeration is always greater than that calculated by the above formula, due to wire drawing, leakage, clearance and superheat in the suction gas. In actual practice we must therefore take into consideration the volumetric efficiency. The actual piston displacement of a compressor may be found by dividing the theoretical volume required per ton per minute by the volumetric efficiency. This may be expressed by the formula:

$$\text{P.D.} = \frac{V}{E_a} \tag{3}$$

where P. D. = piston displacement in cubic feet per minute per ton
$V =$ theoretical displacement in cubic feet per minute per ton
$E_a =$ volumetric efficiency.

The volume of refrigerant handled is important in the use of reciprocating compressors, as it determines the size of the compressor; but with centrifugal compressors a large volume is not objectionable and may be a positive advantage for small units. A high compression is undesirable in reciprocating compressors from the standpoint of clearance losses, and may make compound compression necessary. The large volumes required for methylene chloride, F-11, dieline and water vapor can be delivered by centrifugal compressors. When the suction pressure

Table 162

PERFORMANCE OF REFRIGERANTS

Refrigerant	Operating Temperature Range° F	Suction Press. Gage " Vacuum # Lb per Sq In.	Condenser Press. Gage " Vacuum # Lb per Sq In.	Ratio of Condenser Pressure to Suction Pressure	Per Ton of Refrigeration with Saturated Vapor: Weight of Vapor, lb per min	Piston Displacement CFM	Theoretical Horse Power	Temperature End of Compression, ° F	Per Ton 80% V.E. Practical Displacement CFM
Water H_2O	32–86	29.66"	28.59"	6.92	0.196	646	0.60	282	807
	32–100	29.66"	27.92"	10.68	0.199	655	0.78	354	819
	40–100	29.59"	27.92"	7.77	0.198	483	0.66	313	604
Trichloro-Ethylene C_2HCl_3	5–86	29.51"	26.34"	9.2	2.18	500		210	625
	20–100	29.41"	27.26"	5.2	2.34	358		150 *	597
	40–100	28.92"	24.94"	4.98	2.14	183.6		157 *	229
Dieline $C_2H_2Cl_2$	5–86	28.01"	14.82"	8.2	1.47	109.0		180 *	137
	20–100	27.2"	10.82"	7.2	1.48	60.7		160 *	76
	40–100	25.29"	10.82"	4.18	1.48	48.7		170 *	61
Freon-113 $C_2Cl_3F_3$	5–86	27.92"	13.93"	8.02	3.74	101.0		180 *	127
	20–100	26.80"	8.59"	4.15	3.8	67.7		160 *	85
	40–100	24.52"	8.59"	3.93	3.6	38.4		170 *	48
Methylene Chloride CH_2Cl_2	10–86	27.12"	5.67"	9.07	1.46	74.0	0.96	205	93
	20–100	26.01"	2.97"	6.90	1.52	47.7	0.91	157	60
	40–100	23.04"	2.97"	3.92	1.49	27.8	0.63	167	35
Methyl Formate $C_2H_4O_2$	5–86	29.47"	1.62"	7.9	1.09	50.0			63
	20–100	24.43"	3.56#	6.7	1.08	33.5			43
	40–100	20.45"	3.56#	3.92	1.06	20.0			25

* Approximately

TABLE 162 (*continued*)

Refrigerant	Operating Temperature Range° F	Suction Press. Gage " Vacuum # Lb per Sq In.	Condenser Press. Gage " Vacuum # Lb per Sq In.	Ratio of Condenser Pressure to Suction Pressure	Per Ton of Refrigeration with Saturated Vapor: Weight of Vapor, lb per min	Piston Displacement CFM	Theoretical Horse Power	Temperature End of Compression, ° F	Per Ton 80% V.E. Practical Displacement CFM
Freon-11	5–86	23.95"	3.6#	6.24	3.058	37.0	0.94	113	47
F-11	20–100	21.08"	8.9#	5.47	3.086	26.2	0.89	122	33
CCl_3F	40–100	15.61"	8.9#	3.36	2.943	16.08	0.63	114	21
Freon-21	5–86	19.25"	16.5#	5.83	2.364	20.87	0.94	99	26
F-21	20–100	14.25"	25.3#	5.01	2.404	15.67	0.94	110	20
$CHCl_2F$	40–100	4.84"	25.3#	3.22	2.315	10.19	0.68	106	13
Freon-114	5–86	15"	21.1#	4.94		18.9			24
F-114	20–100								
$C_2Cl_2F_4$	40–100								
Sulfur	5–86	5.88"	51.8#	5.63	1.415	9.08	0.97	191	11.3
Dioxide	20–100	2.48#	69.8#	4.92	1.453	6.52	0.92	193	8.2
SO_2	40–100	12.4#	69.8#	3.12	1.444	4.17	0.63	163	5.2
Methyl	5–86	6.4#	80#	4.47	1.331	6.20	0.99	155	7.7
Chloride	20–100	14.46#	102#	4.14	1.362	4.67	0.95	167	5.9
CH_3Cl	40–100	28.5#	102#	2.70	1.338	3.10	0.64	144	3.9
Freon-12	5–86	11.8#	93.2#	4.07	3.916	5.81	1.00	100	7.3
F-12	20–100	21.05#	116.9#	3.68	4.054	4.54	0.97	113	5.7
CCl_2F_2	40–100	36.98#	116.9#	2.54	3.880	3.07	0.67	108	3.9
Ammonia	5–86	19.6#	154.5#	4.94	0.421	3.44	0.99	210	4.3
NH_3	20–100	33.5#	197.2#	4.40	0.421	2.49	0.94	213	3.1
	40–100	58.6#	197.2#	2.80	0.427	1.70	0.65	176	2.1
Carbon	5–86	317.2#	1028#	3.16	2.6	0.69	2.12	160	0.83
Dioxide	20–87.8	407#	1055#	2.55	3.1	0.63			0.8
CO_2	40–87.8	553#	1055#	1.92	4.0	0.58			0.72

is below atmospheric, as is the case with methylene chloride, dieline, ethyl chloride, Freon-11, Freon-21 and water vapor, air leaks into the evaporator. When the suction pressure is above atmospheric pressure, the refrigerant leaks into the atmosphere. In the case of leaks, odorless and safe refrigerants are desirable, provided the leak can be discovered by simple test methods, such as the halide torch. It is beyond the scope of this book to give complete tables of all refrigerants. A complete table of properties is given in the Data Book published by the American Society of Refrigerating Engineers.

362. Volumetric Efficiency

The volumetric efficiency of a compressor is the ratio of the actual weight of vapor pumped by the machine to the weight calculated from the piston displacement by the gas entering

TABLE 163

PROPERTIES OF SATURATED FREON-12 AT STANDARD CONDITIONS

Temperature	Gage Pressure, Pounds Per Square Inch	Vapor Specific Volume v, Cubic Feet Per Pound	Heat Content, (Enthalpy) In Evaporator, Btu Per Pound H_1	Liquid Before Expansion Valve h_2
In evaporator				
—10	4.5	2.003	77.05	
5 (standard)	11.81	1.485	78.90	
20	21.05	1.121	80.49	
40	36.98	0.792	82.71	
In condenser				
79	81.6	—	—	26.00
86 (standard)	93.2	—	—	27.72
96	109.8	—	—	30.18

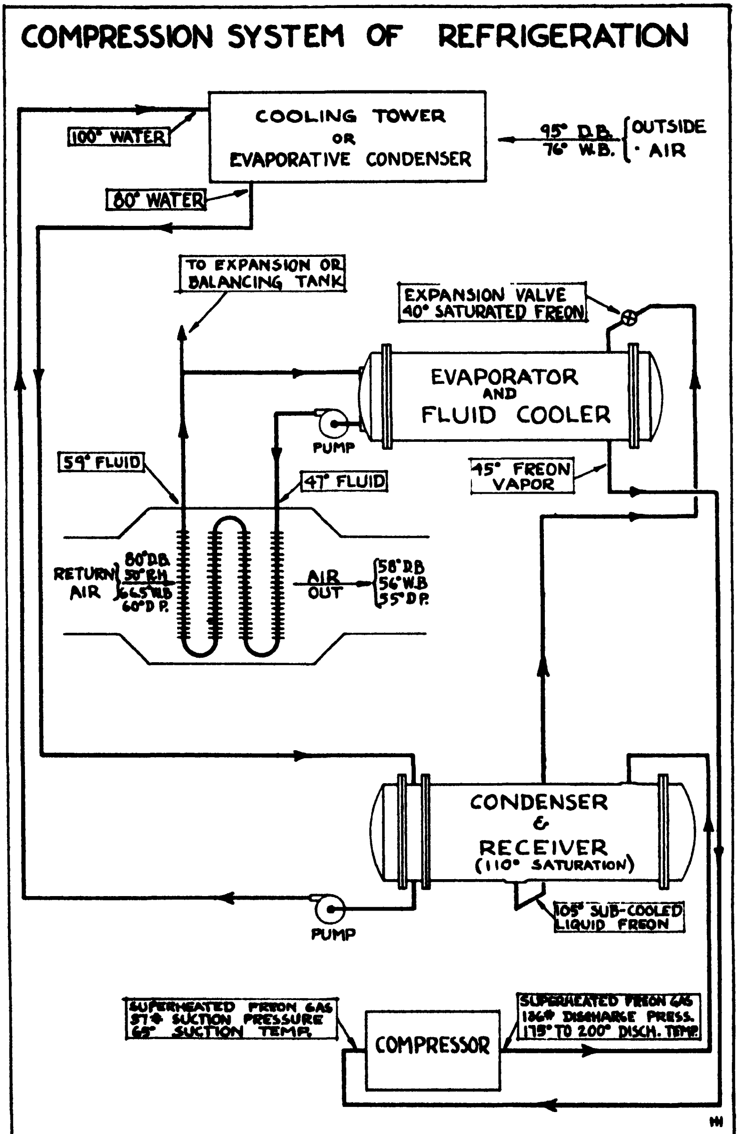

COMPRESSION SYSTEM OF REFRIGERATION
COOLING TOWER OR EVAPORATIVE CONDENSER
100° WATER
80° WATER
95° D.B. 76° W.B. OUTSIDE AIR
TO EXPANSION OR BALANCING TANK
EXPANSION VALVE 40° SATURATED FREON
EVAPORATOR AND FLUID COOLER
PUMP
59° FLUID
47° FLUID
45° FREON VAPOR
RETURN AIR
80° D.B.
50° R.H.
66.5 W.B.
60° D.P.
AIR OUT
58° D.B.
56° W.B.
55° D.P.
CONDENSER & RECEIVER (110° SATURATION)
105° SUB-COOLED LIQUID FREON
PUMP
SUPERHEATED FREON GAS 57# SUCTION PRESSURE 65° SUCTION TEMP.
COMPRESSOR
SUPERHEATED FREON GAS 136# DISCHARGE PRESS. 175° TO 200° DISCH. TEMP.

FIGURE 67

saturated at its evaporating pressure. This ratio is also called the conventional volumetric efficiency. This efficiency, because of the heating of the gas entering the cylinder, the re-expansion from clearances and the leakage from valves and pistons, varies from 50 to 85%. Compressor manufacturers will guarantee a volumetric efficiency of 80 to 85% in units of 50 tons and over, and 75 to 80% in smaller units. Theoretical values of compressor displacement are given in Table 162 above, also practical values, assuming 80% volumetric efficiency.

Problem. Find theoretical displacement of Freon-12, in cubic feet per ton, under air-conditioning operation, with 86° F liquid.

Solution. At 40° F and substituting in equation (2)

$$H_1 = 82.71 \text{ Btu}$$
$$h_2 = 27.72 \text{ Btu}$$
$$v = 0.792 \text{ cu ft}$$
$$V = \frac{200 \times 0.792}{82.71 - 27.72} = 2.64 \text{ cu ft per minute per ton}$$

Problem. Assuming a volumetric efficiency of 88% what is the actual piston displacement required?

Solution. 2.64/0.88 = 3 cu ft per minute.

363. Refrigeration Standards

The following table outlines briefly Standards of Rating which have been adopted by the American Society of Refrigerating Engineers, as described in Circular No. 14, Methods of Rating and Testing Mechanical Condensing Units. This standard refers to motors, compressors and condensers in a manufacturer's assembly. The temperature conditions of the eight groups of standard A.S.R.E. ratings particularly refer to air-conditioning units.

TABLE 164

COMPRESSOR TEST TEMPERATURES, ° F

Group No.	Saturated Refrigerant Vapor Temperature Entering Compressor	Refrigerant Vapor Temperature Entering Compressor	Saturated Refrigerant Vapor Temperature Leaving Compressor
I	—10	65	100
II	5	65	100
III	20	65	105
IV	40	65	110
V	—10	15	95
VI	5	30	95
VII	20	40	100
VIII	40	55	105

The temperature of the refrigerant vapor entering a compressor depends primarily on the arrangement of the refrigerant circuit, the size of the compressor and the characteristics of the refrigerant.

Groups No. I, II, II and IV temperatures specified for the refrigerant vapor entering compressors are intended as a rating basis for compressors which are usually applied with the refrigerant vapor entering them at the specified temperatures.

Groups No. V, VI, VII and VIII temperatures specified for the refrigerant vapor entering the compressor are intended as a rating basis for compressors which cannot usually be applied with the refrigerant vapor entering them at the temperatures specified in Groups No. I, II, III and IV.

Courtesy of A.S.R.E.

364. Practical Selection of Reciprocating Compressor Capacity for Air-Conditioning Operation

Assuming 40° F saturation evaporator temperature, corresponding to 36.98 lb gage suction pressure, and condenser gage pressure of 93.2 lb a theoretical displacement of 2.64 cu ft per minute of Freon-12 per ton of refrigeration is required. Practical considerations however require that this amount be increased to 3.00 cu ft per minute per ton, because of piston and

Table 165

TEMPERATURE CONDITIONS FOR RATING CONDENSING UNITS

Group No.	Saturated Refrigerant Vapor Temperatures	Water Cooled Condenser: Cooling Water Temperature In-going	Water Cooled Condenser: Cooling Water Temperature Out-going	Water Cooled Condenser: Refrigerant Vapor Temperature Entering Condensing Unit	Evaporatively Cooled Condenser: Air Entering Temperature Dry-Bulb	Evaporatively Cooled Condenser: Air Entering Temperature Wet-Bulb	Evaporatively Cooled Condenser: Refrigerant Vapor Temperature Entering Condensing Unit	Air Cooled Condenser: Air Entering Temperature Dry-Bulb	Air Cooled Condenser: Refrigerant Vapor Temperature Entering Condensing Unit
I	Minus 10° F	75° F	85° F	65° F	90° F	75° F	65° F	90° F	80° F
II	Plus 5° F	75° F	90° F	65° F	90° F	75° F	65° F	90° F	80° F
III	Plus 20° F	75° F	90° F	65° F	90° F	75° F	65° F	90° F	80° F
IV	Plus 40° F	75° F	95° F	65° F	90° F	75° F	65° F	90° F	80° F
V	Minus 10° F	75° F	85° F	15° F	90° F	75° F	15° F	90° F	15° F
VI	Plus 5° F	75° F	90° F	30° F	90° F	75° F	30° F	90° F	30° F
VII	Plus 20° F	75° F	90° F	40° F	90° F	75° F	40° F	90° F	40° F
VIII	Plus 40° F	75° F	95° F	55° F	90° F	75° F	55° F	90° F	55° F

The temperature of the refrigerant vapor entering a condensing unit depends primarily on the arrangement of the refrigerant circuit, the size of the condensing unit and the characteristics of the refrigerant.

Groups No. I, II, III and IV temperatures specified for the refrigerant vapor entering the condensing units are intended as a rating basis for condensing units which are usually applied with the refrigerant vapor entering the condensing unit at the specified temperatures.

Groups No. V, VI, VII and VIII temperatures specified for the refrigerant vapor entering the condensing units are intended as a rating basis for condensing units which cannot usually be applied with the refrigerant vapor entering the condensing unit at the temperatures specified in Groups No. I, II, III and IV.

Courtesy of A.S.R.E.

Table 166

DESIGN CHARACTERISTICS OF REFRIGERANTS

Refrigerant	Displacement per Standard Ton, Cu Ft/Min 1	Weight Circulated per Standard Ton, Lb/Min 2	Refrigerating Effect at 5 to 86° F, Btu/Lb 3	Differential Heat of Liquid, 86 to 5° F 4	Latent Heat at 5° F 5	Ratio, Col. 5 to Col. 4 6	Standard Cycle, hp/ton 7	Coefficient of Performance 8	Efficiency, % (Carnot) 9
1. Ammonia	3.44	0.422	474.5	90.6	565.0	6.23	0.991	4.85	84.5
2. Carbon dioxide	0.943	3.528	55.50	62.0	117.6	1.89	1.843	2.56	44.6
3. Dichloroethylene	—	—	—	21.7	136.1	6.29	0.918	5.14	89.4
4. Ethyl chloride	—	—	—	34.7	177.6	5.12	—	—	—
5. Freon-11	36.33	2.961	67.54	16.46	84.0	5.10	0.935	5.04	87.8
6. Freon-12	5.82	3.916	51.07	18.4	69.5	3.77	0.997	4.72	82.2
7. Freon-21	20.43	2.237	89.41	19.93	109.34	5.48	0.926	5.09	88.8
* 8. Freon-113	100.75	3.672	53.67	16.95	70.62	4.17	1.002	4.79	83.5
9. Freon-114	19.37	4.589	43.58	18.88	61.98	3.29	—	—	—
10. Isobutane	11.50	1.794	111.5	48.0	159.5	3.32	—	—	—
11. Methyl chloride	6.09	1.345	150.3	30.46	180.71	5.93	0.963	4.90	85.3
12. Methyl formate	49.9	1.056	189.2	41.7	231.0	5.54	—	—	—
13. Methylene chloride	74.45	1.492	134.1	27.5	162.1	5.89	0.965	4.9	85.3
14. Sulfur dioxide	9.08	1.414	141.4	28.01	169.4	6.03	0.995	4.73	82.5
15. Water	—	—	—	—	—	—	1.15	4.10	71.5
16. Carnot cycle	—	—	—	—	—	—	0.8214	5.74	100.0

Courtesy of A.S.R.E.

TABLE 167

TYPICAL AIR-CONDITIONING RATINGS OF COMPRESSION SYSTEMS USING FREON-12

Horse-Power	Cooling Tons	Cooling Btu/hr	Power Input Kw	Water Gal/hr	Speed rpm
5	5.42	65,100	4.33	465	460
7½	7.68	92,200	6.2	660	650
10	10.3	123,700	9.55	910	620
15	14.0	168,100	12.45	1245	420
20	20.4	244,400	18.7	1800	620
25	25.1	301,000	22.35	2220	750
30	31.4	377,000	27.7	2780	470
40	42.2	509,500	37.1	3740	640
50	50.3	604,000	43.0	4450	750

Note: The above ratings are given under the following conditions:

Cooling Water Temperature	75° F
Suction Saturated Refrigerant Temperature	40° F
Suction Refrigerant Temperature	65° F
Ambient Temperature	90° F
Water Temperature Rise	20° F

valve leakage, wire drawing, clearance, cylinder heating, impurities and foreign gases in the refrigerant. The ratio $\frac{2.64}{3.00}$ gives a volumetric efficiency of 88%. An allowance of 0.9 kilowatt per ton input or one horsepower per ton of refrigeration will assure a motor of ample capacity to start up the compressor. Supplying 1.43 gal of 80° F water per minute per ton generally results in an operating head pressure of 135 lb gage with pure Freon-12. If the quantity of 80° F water were increased to 3.3 gal per minute per ton the condensed pressure would drop to approximately 110 lb and the capacity would be increased about 5% or to 1.05 tons, or 3.25 cu ft per minute displacement per ton.

If the water temperature is increased to 90° F, and an allowance made of only one gallon per minute of condenser water per ton, the condenser pressure would be increased to 170 lb gage and the capacity would be decreased about 10%, requiring a displacement of 3.8 cu ft per minute per ton. It is obvious, therefore, in deciding upon the practical allowance for compressor displacement, that the temperature and quantity of condenser water available are of prime consideration; these are dependent upon outside wet-bulb air temperature with cooling tower or evaporative condensation. The following table is offered for estimating purposes.

TABLE 168

CUBIC FEET PER MINUTE ACTUAL DISPLACEMENT OF FREON-12 PER TON OF REFRIGERATION

(40° F Saturation Temperature in Evaporator)

Cu Ft/ min/ton	Condenser Pressure lb/sq in. Gage	Outlet Water Temperature °F	Inlet Water Temperature °F	Water gal/min/ton	Cooling Tower Design Temperature Wet-Bulb °F
2.94	80	70	60	3.05	55
3.18	115	90	60	0.87	50
3.07	90	80	70	2.05	65
3.38	130	100	70	1.12	55
3.17	100	85	75	4.00	70
3.48	145	105	75	0.87	60
3.28	110	90	80	3.30	75
3.52	150	110	80	0.93	70
3.35	115	95	85	4.9	80
3.74	160	115	85	0.96	80
3.44	125	100	90	4.6	85
3.78	170	120	90	1.0	85

Based on a volumetric efficiency of 88% at 130 lb condenser pressure and 35 lb suction gage pressure; 4 × 4 twin cylinder unit at 850 rpm. For smaller units, assume a lower volumetric efficiency.

365. Centrifugal Compressors

The centrifugal compressor utilizes a centrifugal impeller somewhat similar to that employed in the ordinary centrifugal pump; this impeller compresses the gas on the discharge side and forces the compressed gas through a condenser where the gas is liquefied. The liquid then passes into the cooler where the liquid re-evaporates, the action being quite similar to that obtained with the piston compressor, except that the operation is carried on at pressures below that of the atmosphere.

Usually both evaporator and condenser work below atmospheric pressure. Water and trichloromonofluoromethane (Freon-11) are the refrigerants commonly used in centrifugal compressors. This type of compressor is inherently suitable for large volumes of refrigerant at low pressure differentials. Two or more stages are required and high speeds are necessary. Centrifugal compressors are seldom built for lower than 50 tons capacity, but, if the demand warrants, the industry may expect units of lower capacity.

Centrifugal compressors are used for large installations, i.e., 400 tons and over, and where indirect systems are required.

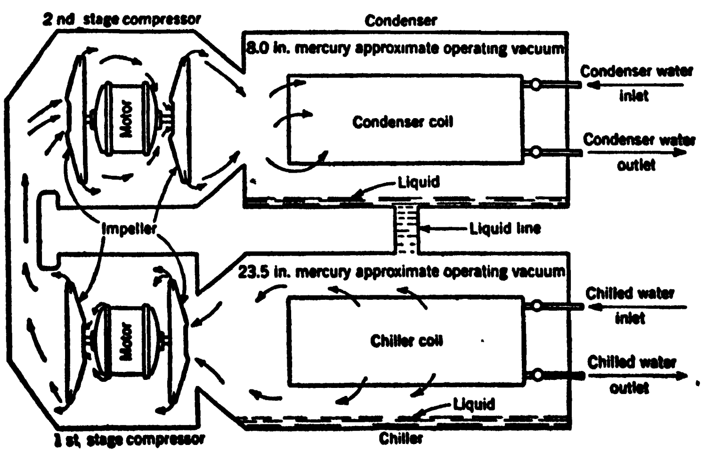


FIGURE 68

Enclosed Type Centrifugal Condensing Unit

366. Steam Ejector Refrigeration

It may seem unusual to speak of using hot steam in the process of producing refrigeration, but this is practical since the steam is only used to produce the necessary high vacuum under which the water is vaporized.

In this type of machine a hermetically tight reservoir is used. The cooling water, or water to be cooled is sprayed into the reservoir, where a vacuum of 29.75 in. is maintained by the action of a steam ejector, the suction of which is connected to the top of the reservoir. When water is sprayed under this high vacuum, approximately 4% evaporates and is removed by the steam ejector. This 4% of the water, in evaporating, will draw the latent heat required from the remaining water, and as water evaporating under this vacuum will require over 1,000 Btu per pound, it follows that for every pound of water sprayed there will be

$$1000 \text{ Btu} \times 0.04 \text{ or } 40 \text{ Btu}$$

removed from the balance of the pound of water which is not evaporated so that the water is cooled between 30 to 40 degrees. While this would not be sufficient for ordinary refrigeration work it is entirely adequate for air-conditioning cooling.

The steam discharging from the ejector, together with the 4% of the evaporated water, is carried to a water-cooled condenser where it is condensed; air and non-condensative gases are removed from the condenser by a secondary ejector of smaller size and delivered to a secondary condenser at 22 in. vacuum, from which they are again removed by a still smaller ejector and delivered at atmospheric pressure into a final condenser from which they may pass off to the atmosphere. This machine takes a larger quantity of condenser water because it condenses the steam used in the ejector as well as removes the heat from the spray water. It is therefore used where steam and large quantities of cooling water are available.

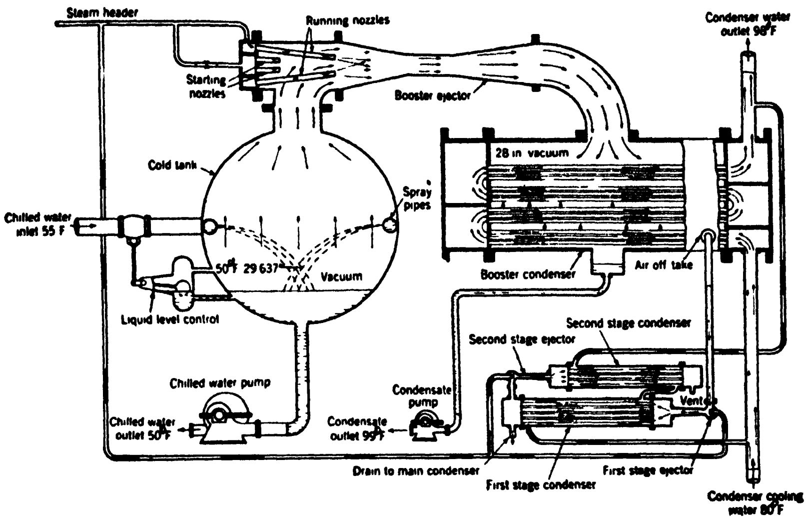


FIGURE 69

Diagrammatic Arrangement of Steam Jet Vacuum Cooling Unit

367. Water-vapor Machine

This type of refrigeration machine is similar to the centrifugal compressor type previously described, but utilizes water as a refrigerant and runs under a vacuum which must be quite high in order to evaporate the water to a sufficient extent to produce the required cooling.

Speeds for this machine run from 7,000 to 10,000 rpm, the power being delivered by an electric motor through a *step-up gear* which increases the speed from that of the motor to that required for the machine. It is claimed that water-vapor machines compare with other types of refrigeration machines in regard to power required and water used.

Water-vapor refrigeration is similar in principle to any of the well-known vapor compression systems except that the water acts as its own refrigerant, thus eliminating the need of any chemical refrigerant. It is particularly suitable for moderate chilled-water temperatures of 40 to 60° F and in some cases as low as 35° F.

The water which is to be cooled enters an evaporator or flash chamber in which there is maintained a very high vacuum (low absolute pressure) corresponding to the desired chilled water temperature. A small portion of the relatively warm entering water immediately boils or flashes into vapor, the latent heat of evaporation being furnished by the remainder of the water, the temperature of which is thereby lowered.

The latent heat of vaporization is approximately 1065 Btu per lb at 0.178 lb per sq in. absolute pressure (i.e., 50° F evaporation temperature) so only approximately 1% of the entering water is evaporated for each 10° of cooling.

As the vapor is formed, it must be removed and compressed to a lower vacuum (higher absolute pressure) at which its temperature will be sufficiently increased so it can be condensed with cooling water at whatever temperature it is available.

A compressor for this service must have large displacement capacity at very high vacuum. A steam-jet booster, which operates on the well-known ejector principle meets the requirements in every particular.

The following table shows some of the properties of water vapor applicable to its use as a refrigerant. For additional data, see steam tables.

TABLE 169
PROPERTIES OF WATER VAPOR

Chilled Water Temperature, ° F	Suction Pressure Inches Hg, Absolute	Specific Volume cu ft/lb	Evaporation Rate (Theoretical)	
			lb/hr/ton	cfm/ton
35	0.2039	2948.5	11.20	550
40	0.2478	2445.1	11.23	458
45	0.3005	2038.8	11.26	382
50	0.3624	1704.9	11.29	321
55	0.4359	1432.9	11.31	270
60	0.5214	1208.1	11.34	228

TABLE 170
PROPERTIES OF WATER

Saturation Temperature, ° F	Absolute Pressure Lb/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

Courtesy of A.S.H.V.E.

368. Coolers and Evaporators

The actual cooling of a refrigerating system occurs at the evaporator, a vessel or pipe coil in which the refrigerant liquid vaporizes or boils continuously. As the liquid evaporates it is constantly being replaced by more liquid from the receiver. The vapor is carried away through suction piping to the compressor. The medium to be cooled, air, water or brine must be warmer than the boiling refrigerant. The types of coolers used in air-conditioning applications fall into three general groups

1. Direct cooling of water
2. Direct cooling of air
3. Cooling of brine for circulation to cool either water or air.

One method used with carbon dioxide systems was the installation of direct expansion coils in the spray chamber; water was sprayed over the coils and into the air stream. Operating with a refrigerant temperature of 30° F and 40 ft of 1¼ in. E. H. coil per ton of refrigeration laid out in eight banks, a dew point of 55° F with 80° F outside wet-bulb was obtained, provided each coil was served with water by a distributor header in addition to one bank of spray nozzles.

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. There are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used because of fire or other risks, especially in densely populated areas, the brine can be cooled in an isolated room or building and can then be circulated through the air-conditioning equipment. This arrangement eliminates any possibility of direct contact between the air and refrigerant.

See Figure 65.

Heat transfer factors and methods of computing evaporator capacities are given in Chapter VI.

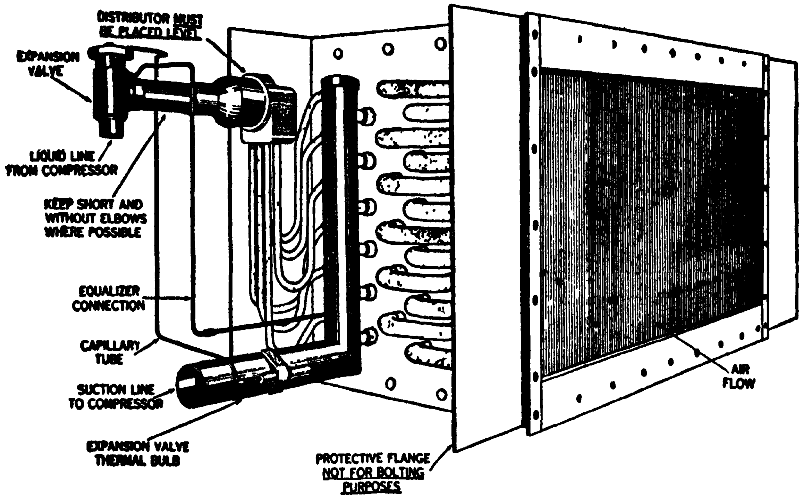


FIGURE 70

Piping Details of Direct Expansion Coils

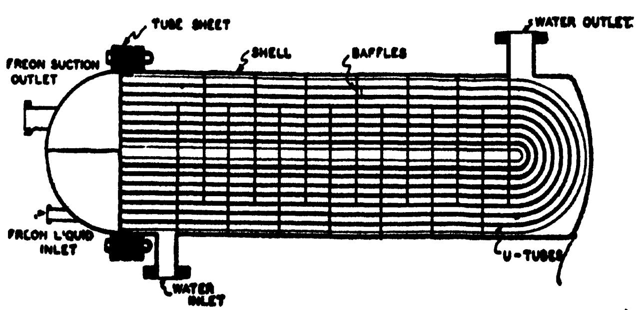


FIGURE 71

Dry Expansion Freon Water Cooler

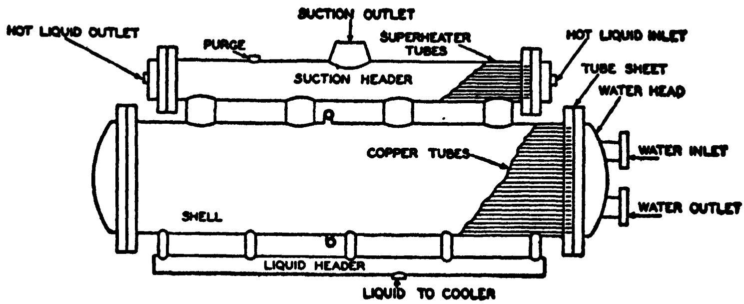


FIGURE 72

Typical Freon Flooded Water Cooler

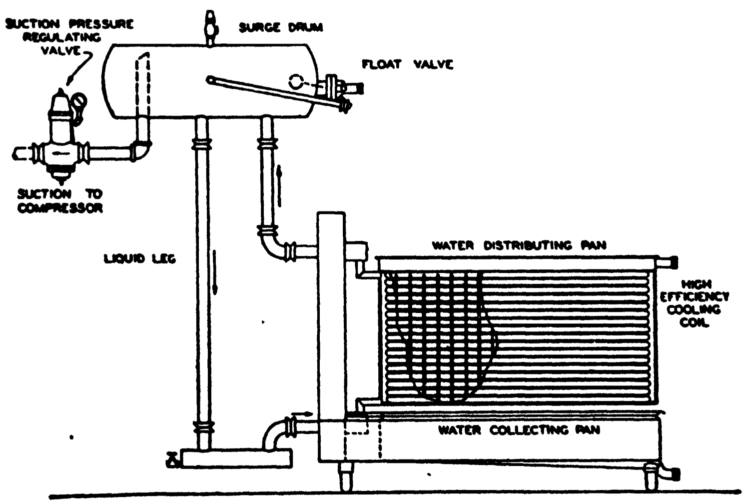


FIGURE 73

Baudelot Cooler

369. Condensers

Three general types of condensers are used in air-conditioning work:

(1) Air-cooled
(2) Water-cooled
(3) Evaporative (combination air and water).

370. Air-cooled Condensers

This type is seldom used for capacities above 3 tons, unless water cannot be obtained. Air-cooled condensers must be located in a well ventilated space, so that the heated air may escape. For this reason a packaged unit, placed in the room to be cooled, is equipped with a water-cooled condenser.

371. Water-cooled Condensers

The commercial types appear in three designs:
(1) Double pipe
(2) Shell-and-tube
(3) Shell-and-coil
The temperature and quantity of condensing water determine the condensing pressure and the power required to operate the compressor.

Municipal water departments are limiting the supply of water available for air conditioning; this limitation necessitates the installation of cooling towers or evaporative condensers. Water coolers produce the warmest water at the time when the load on the system is greatest, consequently the refrigeration equipment must be designed to meet this situation. With a cooling tower or evaporative condenser the water temperature may rise as high as 88° F. Shell and coil condensers are in general use for sizes under 30 tons, and consist of a coil of tubing mounted inside a shell. The cooling water passes through the coil while the shell acts as a receiver.

372. Condenser Standards

The following tables outline briefly the Standards of Rating which have been adopted by the American Society of Refrigeration Engineers.

Water-cooled refrigerant condensers are classified by types as follows:

A. Closed shell-and-tube
B. Closed shell-and-coil
C. Open shell-and-tube
D. Double-tube.

TABLE 171

TEST CONDITIONS FOR CONDENSERS

Test Group No.	Saturated Temperature of Entering Refrigerant Vapor, ° F	Minimum Actual Temperature of Entering Refrigerant Vapor, ° F	Temperature of Entering Water, ° F	Temperature of Leaving Water, ° F	Temperature of Ambient Air, ° F
I	105	125	75	98	90
II	105	125	85	98	90
III	100	200	75	93	90
IV	100	200	85	93	90

TABLE 172

EVAPORATIVE CONDENSER TESTS

A.S.R.E. Rating Condition	Value
Barometric pressure	29.92 in. mercury
Dry-bulb temperature of air entering unit	90° F
Wet-bulb temperature of air entering unit	75° F
Dry-bulb temperature of ambient air	90° F
Temperature of water entering unit	75° F
Saturated temperature of dry refrigerant vapor entering unit	105° F
Actual temperature of dry refrigerant vapor entering unit	Not less than 125° F

Ratio of *Condensing Heat Rejection Effect* to the *Condensing Refrigeration Effect* shall be arbitrarily assumed as 1.20 for the above rating conditions.

373. Evaporative Condensers

The evaporative condenser is an economical means of condensing refrigerant vapors. In many situations it replaces the customary cooling tower and shell-and-tube condenser. The hot compressed refrigerant gas passes through the condenser coil and by the combined action of the spray water and the large volumes of air passing over this coil, the hot gas is reduced to a liquid. The initial cost of an evaporative condenser is less than the combined cost of a cooling tower and shell-and-tube condenser. The total horsepower requirements are lower than for a forced-draft cooling tower located on the roof. Air leaving the evaporative condenser is almost saturated, and must be discharged into the atmosphere to prevent condensation. These condensers are available in units from 8 tons to 110 tons capacity, and use only a small portion of the water required for a water cooled condenser. Each pound of water evaporated extracts 1000 Btu from the refrigerant, whereas when the water-temperature rise is 20° F, each pound of water extracts only 20 Btu from the refrigerant. The evaporative condenser occupies more space, than the city-water-cooled condenser, has a higher first cost and must be located where air is available, but where water is restricted or expensive, the evaporative condenser has been accepted by the industry.

TABLE 173

MANUFACTURER'S RATING AND DIMENSION TABLES FOR TYPICAL EVAPORATIVE COOLER UNITS

	Rating—Tons* at Condensing Temperature		Approximate Overall Dimensions In Inches		
Model No.	Ammonia (95° F)	Freon (105° F)	Height	Width	Length
1	17.1	20.0	79	36	75
2	18.9	23.0	77	32	95
3	25.0	29.0	90	32	98
4	30.9	36.0	90	32	98
5	34.2	40.0	90	32	123
6	41.1	48.0	90	32	123
7	51.7	60.0	99	52	134
8	68.4	80.0	126	62	139
9	82.0	96.0	130	72	139

* Ratings are based upon 76° F Wet-Bulb and 40° F Suction Temperature.

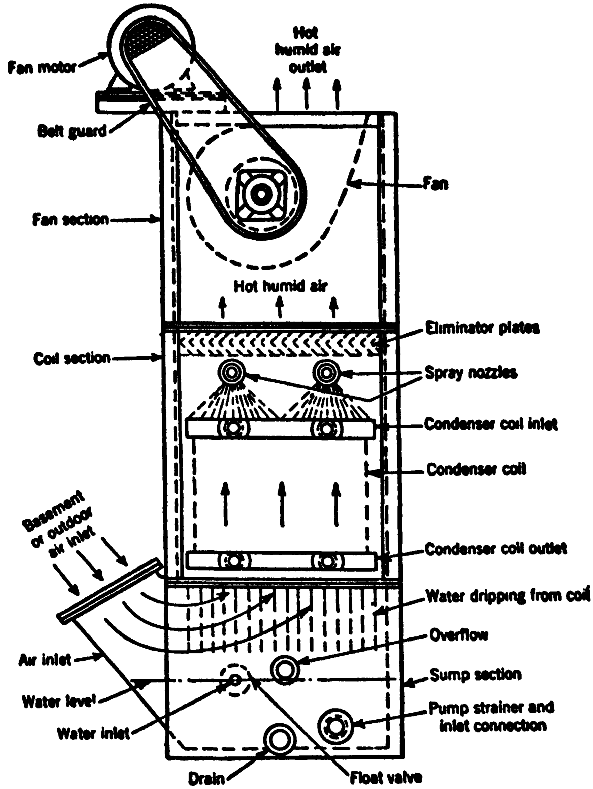


FIGURE 74

Schematic View of an Evaporative Condenser

374. Selection of Refrigerants for Air Conditioning

In Table 4 Chapter I, is given a partial list of refrigerants which might be used in mechanical refrigeration systems; however, in general air-conditioning practice, the choice is limited by the considerations of safety, chemical stability, availability, size, and first maintenance and service costs. Municipal codes are being gradually formulated to limit the tonnage capacity of fully automatic systems, thus requiring the full time service of an operating engineer. More reliable performance is to be expected under full-time operating supervision, which minimizes the cost of repairs and loss of refrigerant; moreover, the efficiency of the refrigerant selected need not be sacrificed for extreme safety characteristics. Inasmuch as industrial plants always have an operating engineering staff available, the refrigerant selected should have a high coefficient of performance, especially in units with a capacity of 100 tons or more. Operating costs per ton, theoretical and practical, are lower for ammonia than for any other refrigerant used in industrial compression systems. Its toxic and irritant properties however, are second on the list. Regardless of the fact that ammonia heads the list of explosive refrigerants, insurance experience has proved that the probability of an ammonia explosion is lower than the hazard of fire from conventional causes. Inasmuch as comfort-cooling air-conditioning systems, at present, outnumber the industrial type, this chapter will devote more details to Group I below, which is recognized by the code as comprising the safest refrigerants. Refrigerants for Safety Code purpose are divided into groups as follows:

Group 1	*Chemical Formula*
Carbon dioxide	CO_2
Dichlorodifluoromethane (Freon-12)	CCl_2F_2
Dichloromonofluoromethane (Freon-21)	$CHCl_2F$
Dichlorotetrafluoroethane (Freon-114)	$C_2Cl_2F_4$
Dichloromethane (Carrene No. 1) (Methylene chloride)	CH_2Cl_2
Trichloromonofluoromethane (Freon-11) (Carrene No. 2)	CCl_3F
Trichlorotrifluoromethane (Freon-113) under consideration	$C_2Cl_3F_3$
Group 2	
Ammonia	NH_3
Dichloroethylene	$C_2H_2Cl_2$
Ethyl chloride	C_2H_5Cl

Methyl chloride	CH_3Cl
Methyl formate	$HCOOCH_3$
Sulfur dioxide	SO_2
Group 3	
Butane	C_4H_{10}
Ethane	C_2H_6
Isobutane	$(CH_3)_3CH$
Propane	C_3H_8

375. The Freon Refrigerants

The most common of these is Freon-12, whose physical properties are best adapted to the economical reciprocating-compressor design used in the smaller type of air-conditioning and refrigeration installations. Freon-11 is used in large centrifugal units. Freon-21 and Freon-114 are used in rotary compressors. The displacement per ton of refrigeration is directly related to the boiling point of the refrigerant. Freon refrigerants are solvents for certain substances, including rubber, but are non-corrosive to the metals used in refrigeration machinery. Decomposition results on exposure to fire or hot metal surfaces and the decomposition products are irritating and toxic. (They include hydrochloric and hydrofluoric acids and traces of phosgene). The Freon refrigerants as such, are the least toxic refrigerants discovered to date, with the exception of air and water vapor. Air as a refrigerant is obsolete, but the water-vapor system has many points in its favor under certain conditions.

376. Leak Detection

Loss of refrigerant is dependent on density, rate of diffusion, capillarity, pressure and workmanship in lining up shaft and facing seal. However engineers look forward to greater developments in the application of hermetic sealing to the design of large units. The majority of service troubles arising in a refrigeration system is due to leakage of refrigerant or penetration of atmospheric air.

TABLE 174

PRESSURE DROP IN REFRIGERANT SUCTION LINES

Freon-12 Flowing in Class L Copper Tubing

Cooling Load Per Pipe		OUTSIDE DIAMETER OF PIPE, INCHES										
		5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8	4 1/8
Btu/Hr	Tons	PRESSURE DROP/100 FT OF PIPE, LB/SQ IN.										
12,000	1	7.3	2.8	....	....	....	....	....	....	....	....	....
24,000	2	....	9.6	4.4	....	....	....	....	....	....	....	....
36,000	3	....	19.4	9.2	2.5	....	....	....	....	....	....	....
48,000	4	....	....	15.4	4.2	1.6	....	....	....	....	....	....
60,000	5	....	....	....	6.4	2.3	....	....	....	....	....	....
72,000	6	....	....	....	8.9	3.2	....	....	....	....	....	....
84,000	7	....	....	....	11.6	4.3	1.8	....	....	....	....	....
96,000	8	....	....	....	....	5.5	2.3	....	....	....	....	....
120,000	10	....	....	....	....	8.1	3.4	0.9	....	....	....	....
180,000	15	....	....	....	....	....	7.2	1.9	....	....	....	....
240,000	20	....	....	....	....	....	12.3	3.3	1.1	....	....	....
360,000	30	....	....	....	....	....	....	6.4	2.3	....	....	....
480,000	40	....	....	....	....	....	....	10.2	4.0	1.7	....	....
600,000	50	....	....	....	....	....	....	14.1	6.0	2.5	1.2	....
720,000	60	....	....	....	....	....	....	....	8.2	3.5	1.7	0.8
840,000	70	....	....	....	....	....	....	....	....	4.5	2.3	1.0
960,000	80	....	....	....	....	....	....	....	....	5.6	2.9	1.3
1,200,000	100	....	....	....	....	....	....	....	....	8.8	4.3	2.3
1,800,000	150	....	....	....	....	....	....	....	....	....	8.9	4.7
2,400,000	200	....	....	....	....	....	....	....	....	....	14.8	7.9

Both suction and liquid line pressure drops are based on 95° F liquid entering the expansion valve, gas leaving the evaporator with 15° F superheat, and a 40° F temperature in the evaporator.

At other suction temperatures multiply the values in Table 174 by these factors: 20° F by 1.55; 30° F by 1.25; 40° F by 1.00; 50°F by 0.85.

Pressure drops include a 50% increase to allow for an average number of fittings.

For every foot elevation from liquid receiver to expansion valve, increase liquid line pressure drop by 0.55 lb per sq in.

The usual practice is to select pipe sizes on the basis of pressure drop, in a given length, of 5 lb per sq in. for liquid lines, and 2 lb per sq in. for suction lines.

TABLE 175

PRESSURE DROP IN REFRIGERANT LIQUID LINES
Freon-12 Flowing in Class L Copper Pipe

Cooling Load Per Pipe		OUTSIDE DIAMETER OF PIPE, INCHES									
		3/8	1/2	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8
Btu/Hr	Tons	PRESSURE DROP/100 FT OF PIPE, LB/SQ IN.									
12,000	1	2.8	0.6	....	....	....	....	....	....	....	....
24,000	2	10.0	1.9	....	....	....	....	....	....	....	....
36,000	3	....	4.7	1.4	....	....	....	....	....	....	....
48,000	4	....	7.9	2.4	....	....	....	....	....	....	....
60,000	5	....	11.6	3.7	1.4	....	....	....	....	....	....
72,000	6	....	....	5.2	1.8	....	....	....	....	....	....
84,000	7	....	....	6.8	2.4	1.1	....	....	....	....	....
96,000	8	....	....	8,6	3.1	1.4	....	....	....	....	....
120,000	10	....	....	12.6	4.9	2.2	....	....	....	....	....
180,000	15	....	....	....	10.3	4.7	1.2	....	....	....	....
240,000	20	....	....	....	....	7.8	2.1	....	....	....	....
360,000	30	....	....	....	....	14.2	4.4	1.5	....	....	....
480,000	40	....	....	....	....	....	7.5	2.6	....	....	....
600,000	50	....	....	....	....	....	11.3	4.1	1.8	....	....
720,000	60	....	....	....	....	....	....	5.7	2.5	....	....
840,000	70	....	....	....	....	....	....	7.6	3.3	....	....
960,000	80	....	....	....	....	....	....	9.6	4.2	....	....
1,200,000	100	....	....	....	....	....	....	14.4	6.3	1.7	0.6
1,800,000	150	....	....	....	....	....	....	....	12.5	3.4	1.2
2,400,000	200	....	....	....	....	....	....	....	....	5.7	2.0

TABLE 176

REFRIGERANT CHARGE FOR FREON-12 LIQUID LINES

Actual Outside Diameter of Liquid Line, Inches	Pounds of Refrigerant to Fill 10 Feet of Liquid Line: Soft Copper Tubing	Hard Drawn Copper Pipe: Standard (Class L)	Hard Drawn Copper Pipe: Extra Heavy (Class K)	Actual Outside Diameter of Liquid Line, Inches	Pounds of Refrigerant to Fill 10 Feet of Liquid Line: Soft Copper Tubing	Hard Drawn Copper Pipe: Standard (Class L)	Hard Drawn Copper Pipe: Extra Heavy (Class K)
3/8	0.41	0.44	0.41	2 1/8	....	17.2	16.7
1/2	0.81	0.81	0.71	2 5/8	....	26 5	25.9
5/8	1.34	1.30	1.21				
	2.02	1.94	1.86	3 1/8	....	37.8	36.8
7/8	....	2.69	2.42	3 5/8	....	51.1	50.0
1 1/8	....	4.58	4.32	4 1/8	....	66.5	64.8
1 3/8	....	7.0	6.8				
1 5/8	....	9.9	9.6				

Note—The weight of refrigerant to fill 10 ft of suction line is only 0.016 times the weight to fill a liquid line of the same diameter and length, and may therefore be neglected.

Table 176A

GAGE PRESSURE-TEMPERATURE RELATIONS OF REFRIGERANTS

Degrees F	Carbon Dioxide	Ammonia	Freon-12 (F-12)	Methyl Chloride	Sulfur Dioxide	Freon-114 (F-114)	Freon-21 (F-21)	Freon-11 (F-11)	Methyl Formate	Methylene Chloride	Freon-113 (F-113)	Dieline (Dichlorethylene)	Trichlorethylene	Water
—30	163.1	1.6	5.45	11.5	21.1	24.70	26.08	27.81			29.31	29.40		
—20	200.2	3.6	0.58	6.0	17.9	22.90	24.67	27.03			29.05	29.20		
—10	242.6	9.0	4.50	0.2	13.9	20.60	22.87	26.01		28.08	28.69	28.92		
—6	261.2	11.6	6.28	1.2		19.50		25.52		27.86				
0	290.8	15.7	9.17	4.2	8.8	17.80	20.59	24.72	26.87	27.48	28.21	28.51	29.61	
2	305.2	16.5	10.19	5.0	7.7		20.07			27.34	28.10			
4	311.8	18.8	11.26	5.9	6.5		19.53	24.11		27.20	27.99			
5	317.2	19.6	11.81	6.4	5.8	16.00	19.25	23.95		27.11	27.92			
6	322.7	20.4	12.35	6.9	5.2		18.96			27.03	27.86			
8	334.0	22.1	13.48	7.9	3.9		18.37	23.45		27.00	27.73			
10	345.5	23.8	14.65	8.9	2.6	14.30	17.75			26.69	27.60	28.00	29.53	
12	357.2	25.6	15.86	9.9	1.1		17.11	22.73		26.49	27.45			
14	369.2	27.5	17.10	11.0	0.1	12.74	16.43			26.00	27.30			
16	381.5	29.4	18.38	12.1	0.8	11.86	15.73	21.94		25.80	27.14			
18	394.2	31.4	19.70	13.3	1.6	10.98	15.01			25.70	26.97			
20	407.1	33.5	21.05	14.5	2.4	10.10	14.25	21.08	24.43	25.60	26.80	27.27	29.41	
22	419.3	35.7	22.45	15.7	3.3	9.08	13.46			25.39	26.61			
24	433.7	37.9	23.88	17.0	4.1	8.04	12.64	20.15		25.08	26.42			
26	444.5	40.2	25.37	18.2	5.1	7.00	11.79			24.84	26.22			
28	461.6	42.6	26.89	19.6	6.0	6.00	10.90	19.14		24.55	26.01			
30	476.1	45.0	28.46	20.9	7.0	5.00	9.98			24.27	25.79	26.44	29.20	

Table 176A (*continued*)

Degrees F	Carbon Dioxide	Ammonia	Freon-12 (F-12)	Methyl Chloride	Sulfur Dioxide	Freon-114 (F-114)	Freon-21 (F-21)	Freon-11 (F-11)	Methyl Formate	Methylene Chloride	Freon-113 (F-113)	Dieline (Dichlorethylene)	Trichlorethylene	Water
32	491.8	47.6	30.07	22.4	8.0	3.80	9.03	18.05		23.95	25.55			29.741
34	507.9	50.2	31.72	23.8	9.0	2.60	8.03			23.63	25.31			29.725
36	521.3	52.9	33.43	25.4	10.1	1.40	7.00	16.87		23.31	25.06			29.709
38	537.0	55.7	35.18	26.9	11.2	.80	5.94			22.97	24.79			29.692
40	563.1	58.6	36.98	28.6	12.4	0.50	4.84	15.61	20.45	22.63	24.52	25.36	28.92	29.673
42	584.3	61.6	38.81	30.2	13.5	1.18	3.70			22.27	24.23			29.653
44	586.4	64.7	40.70	31.9	14.8	1.86	2.52	14.24		21.91	23.93			29.632
46	603.5	67.9	42.65	33.6	16.1	2.54	1.28			21.51	23.61			29.609
48	621.0	71.1	44.65	35.4	17.4	3.22	0.01	12.78		21.09	23.29			29.585
50	638.9	74.5	46.69	37.3	18.7	3.90	0.63			20.67	22.94	24.08	28.55	29.558
52	657.2	78.0	48.79	39.2	20.1	4.74	1.30	11.20		20.21	22.59			29.530
54	675.9	81.5	50.93	41.1	21.6	5.58	1.99			19.75	22.22			29.501
56	694.8	85.2	53.14	43.1	23.1	6.40	2.70	9.53		19.23	21.83			29.469
58	714.1	89.0	55.40	45.2	24.6	7.20	3.44			18.71	21.43			29.435
60	733.9	92.0	57.71	47.3	26.2	8.00	4.20	7.73	14.44	18.19	21.02	22.27	28.09	29.399
62	754.3	96.9	60.07	49.4	27.8	8.92	4.99			17.61	20.59			29.361
64	774.7	101.0	62.50	51.7	29.5	9.84	5.80	5.80		17.03	20.14			29.320
66	795.6	105.3	64.97	53.9	31.3	10.76	6.64			16.40	19.67			29.277
68	816.9	109.6	67.54	56.3	33.0	11.68	7.50	3.72		15.76	19.18			29.231
70	838.7	114.1	70.12	58.7	34.9	12.60	8.38			15.09	18.68	20.19	27.48	29.182
72	861.1	118.7	72.80	61.1	36.8	13.64	9.30	1.53		14.43	18.16			29.130
74	883.5	123.4	75.30	63.6	38.7	14.68	10.25			13.73	17.62			29.075
76	906.6	128.3	78.30	66.2	40.7	15.74	11.22	0.39		13.05	17.06			29.016

Table 176A (*continued*)

Degrees F	Carbon Dioxide	Ammonia	Freon-12 (F-12)	Methyl Chloride	Sulfur Dioxide	Freon-114 (F-114)	Freon-21 (F-21)	Freon 11 (F-11)	Methyl Formate	Methylene Chloride	Freon-113 (F-113)	Dieline (Dichlorethylene)	Trichlorethylene	Water
78	930.1	133.2	81.15	68.8	42.8	16.82	12.23			12.29	16.47			28.954
80	954.0	138.3	84.06	71.5	44.9	17.90	13.26	1.61	5.37	11.54	15.87	17.64	26.75	28.889
82	978.3	143.6	87.00	74.3	47.1	19.14	14.32			10.30	15.25			28.819
84	1003.0	149.0	90.10	77.1	49.4	20.38	15.41	2.90		9.06	14.60			28.746
86	1028.0	154.5	93.20	80.0	51.7	21.60	16.53	3.58		8.22	13.93			28.668
88		160.1	96.40	82.9	54.1	22.80	17.69	4.27		7.78	13.24			28.586
90		165.9	99.60	85.9	56.5	24.00	18.88			7.32	12.53	14.57	25.89	28.499
92		171.9	103.00	89.0	59.0	25.40	20.10	5.73		6.40	11.79			28.408
94		178.0	106.30	92.1	61.6	26.80	21.36			5.48	11.03			28.311
96		184.2	109.80	95.3	64.3	28.20	22.65	7.27			10.24			28.209
98		190.6	113.30	98.6	67.0	29.60	23.98				9.42			28.102
100		197.2	116.90	102.0	69.8	31.00	25.34	8.90	3.56	2.40	8.59	10.84	24.94	27.988
110		232.3	136.00	120.7	85.0		32.70	13.40		1.62		6.57	23.51	27.325
120		271.7	157.10	139.5	106.2		41.05	18.50	12.54	4.75	0.70	1.23	21.80	26.475
140			205.50	184.9	143.9		61.02	30.80	23.71	12.40	7.23		17.10	24.040
160				238.8			85.91	46.34			15.74			20.270

Note: Above line —— inches vacuum. Below line —— pounds per square inch gage. Barometric pressure 29.5 in. Hg for Methylene Chloride. Allow 29.921 in. Hg otherwise.

TABLE 177

STANDARD RATING BASIS FOR SELF-CONTAINED AIR-CONDITIONING UNITS

Functions	Types of Units	Rating Condition		
		Item	Description	Value
All	All	a	Barometric Pressure	29.92 in. Hg
Cooling	Water-Cooled, Air-Cooled and Evaporatively Cooled Condensers	b	Unit Ambient and Air Entering Room—Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 80° F 67° F
		c	Ventilation Air	See Note
	Water-Cooled Condensers	d	Water Temperature Entering Unit	75° F
		e	Water Temperature Leaving Unit	95° F
	Air-Cooled and Evaporatively Cooled Condensers	f	Air Entering Outside Air Inlet (1) Dry-Bulb (2) Wet-Bulb	 95° F 75° F
Heating	All Types Provided with Heating Function	g	Unit Ambient and Total Air Entering Unit	70° F
		h	Heating Medium, Pressure or Temperature (1) Dry Saturated Steam (2) Water In (3) Water Out	 16.7 lb per sq in. 180° F 160° F
Humidifying	All Types Provided with Humidifying Function	i	Unit Ambient	70° F
		j	Total Air Entering Unit (1) Dry-Bulb (2) Wet-Bulb	 70° F 53° F
Air Circulation	All	k	Filters	New and Clean

Note: Rating shall be based on both ventilation and recirculated room air entering at 80° F dry-bulb and 67° F wet-bulb temperature.

Courtesy of A.S.H.V.E.

377. Compressor-Type Refrigeration Control

Refrigeration compressors may furnish refrigerant to direct expansion coils through which air is being passed, or to coils in cooling tanks through which is passed water which is then pumped to air washers or remote cooling coils.

In either case the compressor motor may be started or stopped in order to meet the demand for refrigeration, or a back-pressure automatic pressure valve may be used to regulate the low side or suction pressure of the compressor. When the latter method is used the flow of liquid refrigerant to the cooling coils may be regulated by the opening and closing of a solenoid refrigerant valve at the command of a temperature controller or thermostat. A high pressure cut-out as an individual unit or in combination with either a temperature or pressure controller provides a safety feature against excessive pressures on the high side of the compressor. Many compressors may be unloaded by instruments sensitive to room or duct conditions or by refrigerant pressures, thus reducing the frequency of starting and stopping. If two or more compressors are used for a single coil, step controllers are used to start them in sequence, at intervals of a few seconds, to avoid the large momentary electric current input that simultaneous starting would demand.

378. Ice Cooling Control

When ice is used for the cooling or dehumidification of air, it is usually placed in bunkers and water is sprayed over it. This water, after being cooled, may be used in air washers or surface cooling coils and is usually returned to the bunker for additional cooling after being used.

Control of the water temperature leaving the cold water tank may be maintained by a temperature controller, which measures the temperature of the water in the tank and modulates a control valve in a by-pass which permits a portion of the return water to return directly to the tank instead of passing through the sprays.

379. Vacuum Refrigeration Control

A vacuum refrigerating system consists of an evaporator, compressor, condenser and auxiliaries. The refrigerant used is water, and water vapor (steam) is the power medium.

Water which has been passed through an air washer or cooling coil is sprayed directly into the evaporator or water cooler where it is cooled by its own evaporation. A condenser is attached directly to the compressor discharge and its function is to recondense the water vapor drawn from the evaporator, plus the steam which supplies the energy for compression.

The temperature of the cold water leaving the flash chamber should be measured by a temperature controller which will in turn operate a two-position or positive-control valve installed in the steam line to the jet so as to permit steam to flow only when cooling is required. If city water is used in the condenser, the amount of water should be modulated according to the demand as measured at the condenser outlet by means of a temperature controller and control valve.

380. Cooling by Well Water

When well water is available in sufficient quantities at low temperatures during the cooling season, it may be pumped directly to air washers or cooling coils. Control is usually effected through control valves on the water supply to the cooling unit actuated by temperature or humidity controllers, or both, located either at the outlet of the conditioner or in the conditioned space.

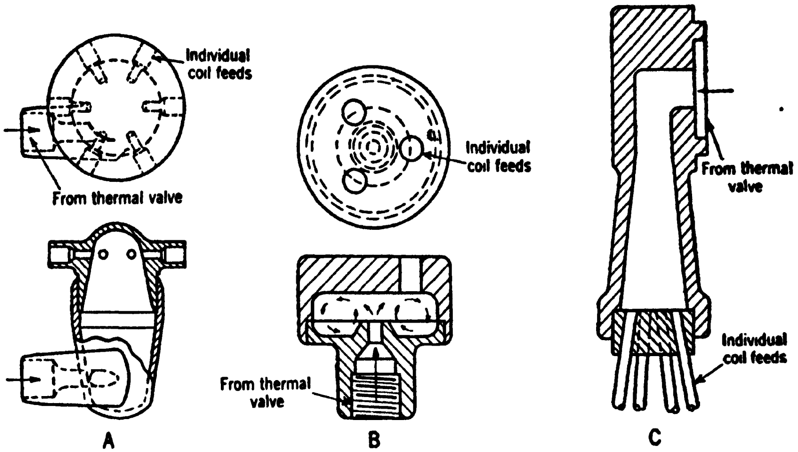


FIGURE 75

Types of Refrigerant Feed Distributing Heads

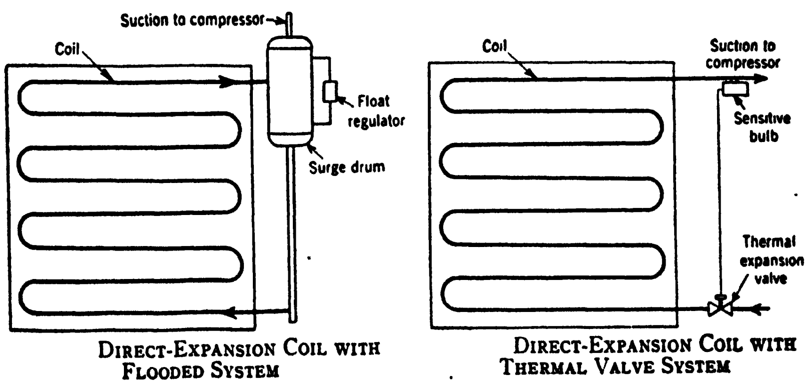


FIGURE 76

Expansion Systems

Chapter XIX

UNIT AIR CONDITIONERS

The unit air conditioner is a miniature air-conditioning plant in itself and may be located around the building in any position where the services required to operate the unit can be obtained or supplied. In cases where running a duct system involves too much difficulty or where there are only a few rooms to be conditioned, the air-conditioning unit is particularly suitable; it has even been used in some large installations, although there seems to be a point where the size of the installation will require a central system both on the basis of economy in first cost as well as on the servicing of a very large number of comparatively small units scattered throughout the building.

The installation of units, although it eliminates the necessity for large-scale duct work and for most small-scale duct work, introduces other difficulties: an outside air supply, electric services, water and waste piping, heating supply and return and occasionally refrigerant supply and return, depending on the type of unit employed.

In the central system these services all may be concentrated in the apparatus room where appearances do not count and large single connections may be used. In the unit system the services must be split up into many small connections, located around the building wherever the units happen to be, and where appearances must be considered after the piping and services are installed. Besides this, units occupy a certain amount of floor space and the larger the number of units the greater this floor space requirement becomes. There are certain jobs which are peculiarly suited to the use of units and other jobs which are peculiarly suited to the use of the central system. There are still other jobs which may be conditioned in either way, often at costs which are approximately the same.

While the use of units eliminates all of the large-capacity apparatus required in the apparatus room of a central system as well as the expensive item of duct work and its insulation, it follows that a similar and equal capacity must be supplied in the units resulting in a large number of small-capacity machines which in themselves are expensive; so expensive, in fact, as to eliminate the savings over the installation of a central system in many instances.

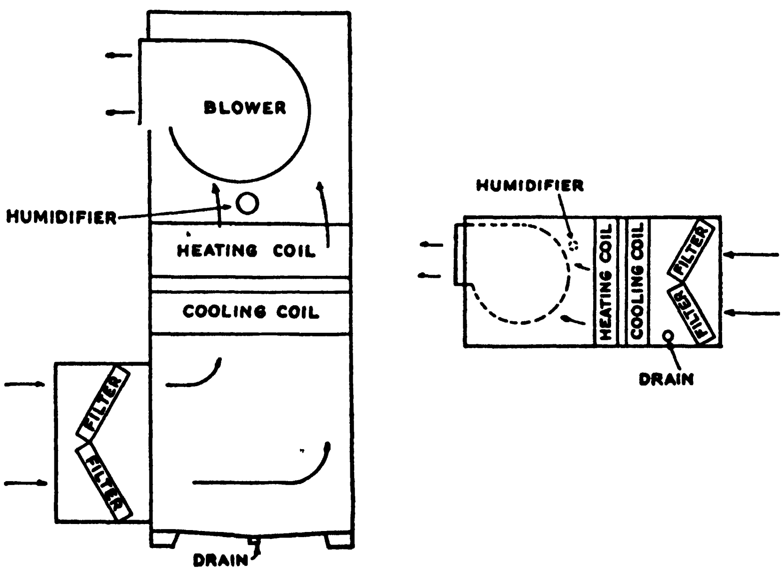


FIGURE 77

Floor and Ceiling Low Side Units for Remote Installations

381. Kinds of Units

There are many types of air-conditioning units on the market, some of which are not real air-conditioning units at all since they do not furnish all three essential features of air conditioning: heating, humidifying and air motion in the winter and cooling, dehumidifying and air motion in the summer, all requiring simultaneous control in each case. Thus, a unit supplying warmed air in the winter gives heating and air motion, but, unless it included humidification and had all of these automatically controlled, it should not be classified as a real winter air-conditioning unit. A cooling unit supplying air motion, cooling and dehumidifying in the summer could claim true summer air conditioning only if these three features were automatically controlled. A manually operated unit technically would not be a summer air-conditioning unit under this requirement, although this type of unit sometimes is installed.

382. Construction of Various Units

Air-conditioning units are made in many styles and forms. The designs usually include a heating element supplied with steam or hot water, a pair of fans, or, in some cases a single fan, and some type of humidifying equipment (most frequently of the spray or atomizing form). Summer-cooling types will include the self-contained unit with the compressor, condenser and compressor-motor all included in the unit along with the cooling element and fan or fans; some types have a drip pan and drain to remove the water condensed out of the air, and other types employ a small air discharge to remove this condensate as it is evaporated on the condenser coils. The ordinary summer unit, which is not self-contained, has simply the cooling element and fans with some means for removing condensate. Year-round units must either have a separate heating and cooling element or may have a combination element, especially when hot and cold water is used for operation. They must also have some means of humidifying in winter and for removing the condensate from dehumidifying in summer. Every manufacturer has his own ideas as to how this may be best accom-

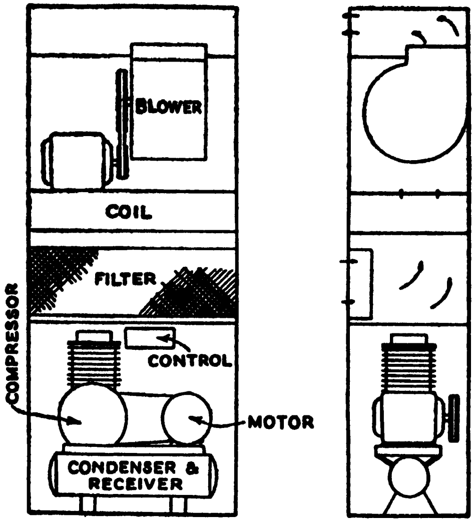


FIGURE 78

Complete Air-Conditioning Unit Designed for Stores and Offices

plished and the various designs represent these ideas in concrete form.

383. Standard Rating of Air-Conditioning Units

In the last few years air conditioning with unit equipment has gained in popularity, and unit conditioners represent a large portion of the industry. Since some equipment does not fulfill all the basic requirements of a true air-conditioning unit, the necessity arose for standardizing and defining functions; therefore a code was prepared by a joint committee of manufacturers and representative societies. This defines various types of unitary equipment as follows:

The code, Standard Method of Rating and Testing Air Conditioning Equipment, defines the various types of unitary equipment:

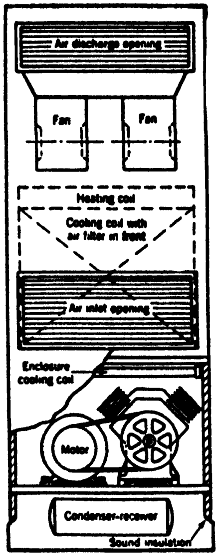


FIGURE 79

Self-Contained Water-Cooled Air Conditioner

1. A *Cooling Unit* is a specific air treating combination consisting of means for air circulation and cooling within prescribed temperature limits.

2. An *Air-Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for maintaining temperature and humidity within prescribed limits.

3. A *Cooling Air-Conditioning Unit* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for cooling and maintaining temperature and humidity within prescribed limits.

4. A *Self-Contained Air-Conditioning or Cooling Unit* is one in which a condensing unit is combined in the same cabinet with the other functional elements. Self-contained air-conditioning units are classified according to the method of rejecting condenser heat (water cooled, air cooled, and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type)

5. A *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

6. A *Pressure Type Unit* is for use with one or more external elements which impose air resistance.

7. A *Forced-Circulation Air Cooler* is a factory encased assembly of elements by which heat is transferred from air to evaporating refrigerant.

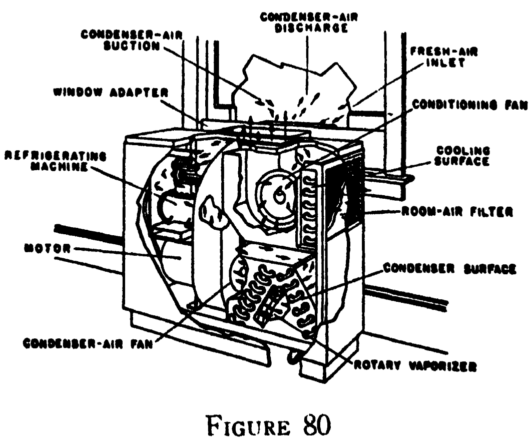


FIGURE 80

Self-Contained Air-Cooled Unit Air Conditioner

TABLE 178

TYPICAL RATINGS OF LOW SIDE UNITS

(Refrigerant at 40° F, entering air at 80° F dry-bulb and 67° F wet-bulb)

Unit Number	Air Volume, Cu Ft/Min	Coil Face Area, Square Feet	Number of Stacks	Capacity, Btu/Hr	S.H.R.	Leaving Temp., °F Dry-Bulb	Leaving Temp., °F Wet-Bulb
1	1200	3	2	27,200	0.68	66	60.0
			3	37,300	0.66	61	57.1
			4	46,000	0.65	57	54.5
			5	53,000	0.64	54	52.3
			6	59,000	0.63	52	50.4
2	2000	5	2	45,300	0.68	66	60.0
			3	62,400	0.66	61	57.1
			4	76,600	0.65	57	54.5
			5	88,500	0.64	54	52.3
			6	98,300	0.63	52	50.4
3	3000	7½	2	68,000	0.68	66	60.0
			3	93,400	0.66	61	57.1
			4	115,000	0.65	57	54.5
			5	132,500	0.64	54	52.3
			6	147,500	0.63	52	50.4
4	4000	10	2	90,500	0.68	66	60.0
			3	124,300	0.66	61	57.1
			4	153,000	0.65	57	54.5
			5	177,000	0.64	54	52.3
			6	196,500	0.63	52	50.4
5	6000	15	2	136,000	0.68	66	60.0
			3	187,000	0.66	61	57.1
			4	230,000	0.65	57	54.5
			5	265,000	0.64	54	52.3
			6	295,000	0.63	52	50.4

$$\text{S.H.R.} = \text{Sensible heat ratio} = \frac{\text{Sensible heat}}{\text{Total heat}}.$$

Courtesy of A.S.H.V.E.

384. Year-Round Conditioning with Water

Water makes an ideal medium with which to accomplish summer and winter air conditioning with the same unit. In the winter the water may be heated to any desired temperature, not above boiling point, and the temperature of the water may be easily controlled. In the summer the water may be cooled to any desired temperature not below freezing point and the cooling and dehumidifying regulated to suit the cooling load. Moreover there is no necessity for changing any valves in the operation since water is used for both winter and summer conditions; all that is necessary is a thermostat which will switch on the

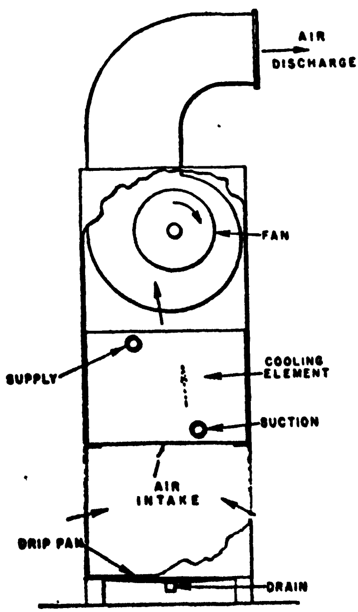


FIGURE 81
Surface Type Cooling Unit

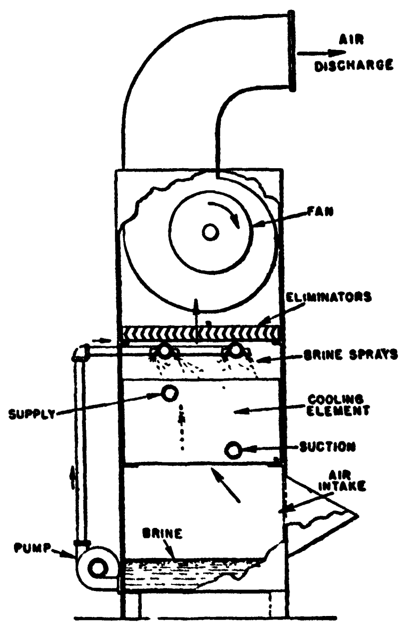


FIGURE 82
Brine Spray Type Cooling Unit

cold water circuit when the room gets too hot, or the hot water circuit when the room grows too cool. The same piping to and from the units is used in either case and in winter the water runs through a heater in the apparatus room, while in the summer it is shunted through a water cooler in which the refrigerant coils are placed and also thermostatically controlled. See Figure 85.

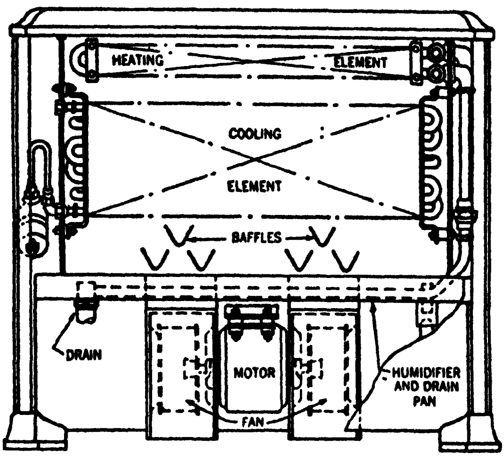


FIGURE 83

Remote Floor Type Room Unit Air Conditioner

For comfort cooling in summer water must be chilled to 50° F or lower and allowed to rise to a temperature not exceeding 55° F, to attain a satisfactory dew point.

Recent designs for contemplated hotels, apartments and office buildings feature this plan for the cooling and heating of individual guest rooms and offices.

385. Water Circuit Piping for Units

The installation of a circulating system of water piping for unit installations should be thoroughly understood by the maintenance man as he may be called on to make additions to such a system or to remove units already on the system and to substitute larger units. The water is circulated by a pump, usually of the centrifugal type, and as the system is a closed circuit, the pump circulates the water around the system and through the units, cooler or heater so as to maintain the flow in adequate quantity. As the degree of cooling usually exceeds the degree of heating necessary, it is customary to size the circulation piping on the basis of the summer requirements and then to use water just hot enough to meet the winter requirements; this arrangement permits the same pump and piping to be used without change. It will be noted that this is somewhat similar to the procedure in the case of central duct systems intended for both summer and winter use. A smaller pump may be used in winter.

386. Quantity of Water Required

In summer cooling it is customary to assume as large a drop in the temperature of the water going through the unit as can be permitted and still maintain an average temperature of water in the unit sufficiently low to accomplish the dehumidifying desired. Usually this temperature drop is not less than 10 degrees and not more than 20 degrees. The amount of heat that each pound of water circulated will pick up in rising 10 degrees is 10 Btu, but if it rises 20 degrees it will pick up 20 Btu. From this the rule may be formulated that, the greater the rise in temperature of the water passing through the unit, the smaller will be the number of pounds of water which will have to be circulated. It is necessary, however, to keep this rise in temperature within reasonable limits as the supply water cannot approach too close to freezing temperature without introducing

A: Danger of freezing the water in the cooler.
B: Producing a greater refrigeration load due to the low temperature required in the water.

As a ton of refrigeration is equivalent to the removal of 200 Btu per minute, it is evident that when the water rises 10 degrees in passing through the unit it will require

200 Btu/10 Btu or 20 lb of water per minute

per ton of refrigeration produced while with a 20 degree rise the amount of water to be circulated will reduce to

200 Btu/20 Btu or 10 lb of water

per ton of refrigeration produced. There are 8⅓ lb of water to a gallon so 20 lb of water approximately equals 2½ gal, and 10 lb of water approximately 1¼ gal. From this the following table may be developed.

TABLE 179

Tons of Refrigeration	Rise in Water Temperature Through Unit (Approx.) 5° F	10° F	12.5° F	20° F	25° F
	Gallons Per Minute to Be Circulated				
1	5.0	2.5	2.0	1.25	1.00
2	10.0	5.0	4.0	2.50	2.00
3	15.0	7.5	6.0	3.75	3.00
4	20.0	10.0	8.0	5.00	4.00
5	25.0	12.5	10.0	6.25	5.00
6	30.0	15.0	12.0	7.50	6.00
7	35.0	17.5	14.0	8.75	7.00
8	40.0	20.0	16.0	10.00	8.00
9	45.0	22.5	18.0	11.25	9.00
10	50.0	25.0	20.0	12.50	10.00

387. Water Circulating Piping (All Galvanized)

A water coil in the unit cooler should be made without air traps, so that it will fill and circulate water. It should be arranged with regulating and shut-off valves with long yokes to project through the pipe-covering. Vents and drains should be provided. The pipe length is determined by water velocity and permissible temperature rise in any parallel run of the circuit. The proper size of the water main is determined by permissible friction head losses. For positive circulation the three-pipe system is recommended (as shown in Figure 85) for distribution in buildings of more than three stories; where static pressure due to height is an important problem. In the three-pipe system the return riser reaches to the top of the building and is connected to a balancing tank, then drops clear through the building to the pump room. The balancing tank accomplishes the following:

(a) Equalizes head and reduces pump power
(b) Maintains practically constant water level
(c) Vents air and gases from return water
(d) Permits *breathing* of water, as it expands or contracts with temperature changes
(e) Visibly indicates leaks in a closed system, by its water level

Another advantage of the three-pipe system is that all the water in the system traverses the same lineal route, regardless of which unit coil it may enter. This prevents short circuiting and permits a fairly uniform temperature difference in all parts of a multistory building. In all cases, the third leg, or balance line equalizes the static pressure in the supply and return mains. Consequently, the pump becomes a circulator, for it must over-

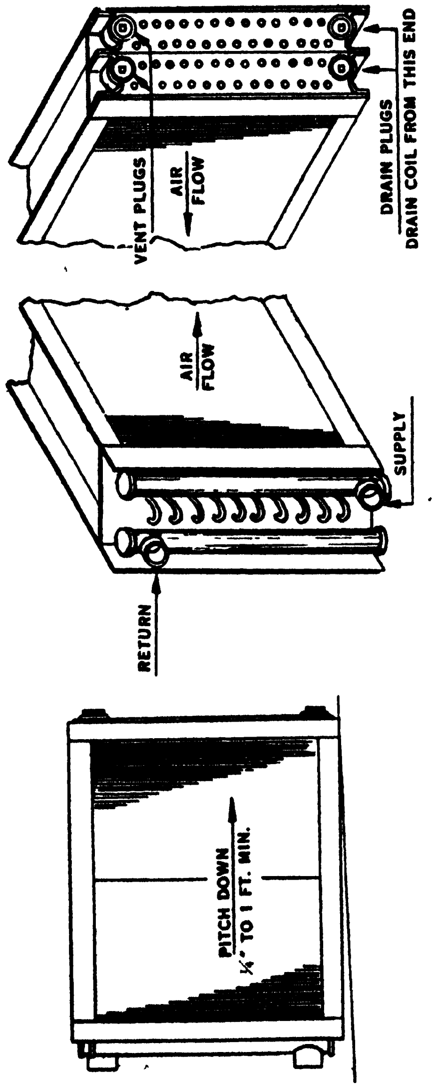
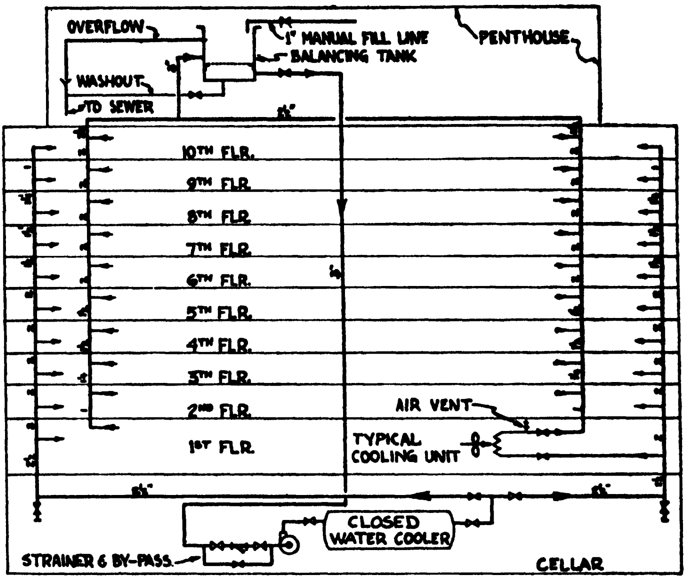


FIGURE 84
Drainage of Water Cooling Coils

THREE PIPE WATER CIRCULATION FOR REMOTE UNIT AIR CONDITIONERS



RISER DIAGRAM

Note: Allow 4 GPM for each cooling unit. (Equivalent to 2 tons refrigeration)

TABLE : PIPE SIZING OF WATER MAINS AND BRANCHES (Based on 2 GPM per ton of refrigeration and 12° rise to and from water cooler.)

NOMINAL PIPE SIZE	GPM	HEAD LOSS FT. PER 100	VELOCITY FT. PER SEC.	TONS REFRIGERATION
10	2400	5.04	9.76	1200
8	1200	4.90	8.33	600
6	750	4.46	6.66	375
5	360	4.24	3.77	180
4	220	5.12	5.55	110
3½	150	4.87	4.67	75
3	100	4.47	4.34	50
2½	55	4.24	3.69	22.5
2	36	4.37	3.35	18
1½	18	4.31	2.84	9
1¼	10	3.08	2.15	5
1	5	3.24	1.86	2.5

FIGURE 85

come only the friction head of the piping system regardless of height. The operating cost for the pump power of such a system is therefore greatly reduced; a single stage centrifugal pump is usually ample to produce circulation, but the pump casing must be of sufficient strength to withstand the full static head to the balancing tank. All pipe and fittings should be heavily galvanized to prevent corrosion.

388. Water Friction Table

In general, it is economical to design pipe sizes in a continuous recirculating system so that the water velocity will not exceed 2 fps in pipes smaller than 2 in. or up to 5 fps for 3 in. pipe. This gives a friction loss of less than 2 lb per 100 ft of run, or, expressed differently a loss of 4 ft of head per 100 ft of run.

The water friction table for straight pipe and the table for fittings are given in Chapter X.

The friction table shows the size of pipes needed to deliver various quantities of water. It also gives the friction loss in head per 100 ft of length of pipes. The tons of refrigeration to be delivered for various quantities of water are given in a preceding topic.

389. Piping Layout

After locating pipe stacks and pipe runs the next step is to determine the amount of friction which it is desired to have for the entire system; i.e., the head against which the pump must operate. Consideration must be given to the fact that small pipes produce high friction resistance while large pipes produce small friction resistance for equal quantities of water flowing. Therefore, it is desirable to have sufficiently large pipes so that the pump head will not become excessive, but at the same time small enough so as not to materially increase the cost of the work. Whatever head may be set for a maximum on the pump, it follows that the head or friction which can be used up in the pipe must be the pump head less the friction of the unit less the friction of the cooler. The friction of the units is very variable and should be obtained from the manufacturer for the quantity of water which will be forced through the unit desired.

The power consumed in the operation of a closed system is based on the difference in pressure between the discharge pressure and the suction pressure. When the pump is shut down the full static pressure head of a high building will crack the pump casing. A special extra heavy pump casing is required on closed systems for the above reasons.

In most cases, 10 ft of head will probably be sufficient, although this may be exceeded at times. Perhaps another 10 ft of head will be used up in passing the water through the cooling coils so that about 20 ft of head must be subtracted from the total head on the circulation pump in order to find the remaining head which may be allowed for the piping.

Let it be assumed, for example, that the total head on the pump is not to exceed 50 ft. Then the friction head which may be used up in the piping is

50 ft minus (10 ft plus 10 ft) or 30 ft

provided that the loss of head through the cooler and one unit does not exceed a total of 20 ft.

The next point to be determined is the equivalent length of run (elr) for the pipe circuit through which the water must travel. For this purpose the length of the supply pipe from the cooler to the farthest unit connection is taken and the length of the return pipe from the outlet of the same unit back to the cooler also is taken; the sum of these two lengths is the actual length of the longest travel through which the water will have to flow, but it is *not* the equivalent length of the piping. *Equivalent length* is that length of pipe in which the friction loss will be the same as in the shorter length plus the friction set up by the valves and fittings on the lines.

Usually it is close enough to add 50% to the actual length to allow for the fitting and valve loss. So if the actual length of the supply piping to the farthest unit measured 150 ft and the actual length of the return piping from the farthest unit measured 250 ft the total *actual* length of the circuit is

150 ft plus 250 ft or 400 ft

while the equivalent length would be

400 ft plus 50% of 400 ft or 600 ft

390. Equivalent Length

Since most friction tables for the flow of water in pipes are based on the friction per 100 ft of straight pipe, it is evident that the circuit must be reduced to terms of straight pipe in order to use such tables in pipe sizing and, therefore it is necessary to determine the length of straight pipe which would have the same friction as the actual pipe plus its valves and fittings. Since the circuit has a friction which is equal to 600 ft of straight pipe, it is evident that if the friction loss for the entire circuit is to be 30 ft as previously set up, then the friction per 100 ft of straight pipe must not exceed

$$\frac{30 \text{ ft of head}}{600 \text{ ft elr}} \times 100 \text{ ft elr or } 5 \text{ ft per } 100 \text{ ft of elr}$$

(for pipe sizes larger than 1¼ in.; allow 2 ft friction loss per 100 ft for smaller pipes) and the pipe may be sized from any set of friction tables or curves showing the friction for the flow of water in pipes. See table in Figure 85.

391. Shortening the Run

If the run is shortened and all other conditions remain the same, it is evident that a greater amount of friction may be allowed per foot of elr as is shown by the following:

Assuming that the distance to the farthest unit is 100 ft on the supply side and 100 ft on the return side, the total actual length of the circuit will be 200 ft and the elr

200 ft plus 50% of 200 ft or 300 ft elr

As 30 ft of head can be used up in pipe friction, the friction allowable per 100 ft of elr will be

$$\frac{30 \text{ ft of head}}{300 \text{ ft elr}} \times 100 \text{ ft elr or } 10 \text{ ft per } 100 \text{ ft elr}$$

and the pipe size for an equal amount of water flowing in gpm (gallons per minute), will possibly be found to be one pipe size smaller, owing to the fact that the allowable friction is twice as great because the length of run is only half as long.

Thus it will be seen that a long circuit should have a low friction per 100 ft elr to avoid building up a high head on the pump, while a short circuit may have a higher friction and use smaller pipes to carry the same amount of water, still maintaining the same head on the pump.

It should also be noted that *head on the pump is not the water pressure in the circuit,* but is the difference in the water pressure between the suction side of the pump and the discharge side; it is a factor of the sum of all the friction generated by circulating the water through the system, but the *water pressure* on the system may be furnished by a balancing tank on the roof or by street pressure and has nothing to do with the friction set up when the water is circulated through the line. The pressure on the system will exist whether the pump is running or not, but the head on the pump will exist only when the pump is operating.

Chapter XX

CENTRAL AIR CONDITIONING SYSTEMS

As the name implies, such systems centralize the means for air distribution and condition the air for a group of rooms, a zone or a complete building through one or more centrally located fans and suitable distribution ducts. Depending on the extent to which the air handled in such systems is conditioned, additional equipment may be included as part of such central fan systems. Such additional equipment consists of dampers used to regulate the fresh, return or recirculated air, to adjust the volume of air and to determine the path of the air through or around the heating or cooling coils. The positioning of all such dampers is provided for by motion imparted to them from damper motors regulated in turn by room thermostats and duct or outdoor temperature controllers.

392. Heating and Cooling Coils

The heating coils are divided into three types, considered in terms of temperature control

1. Preheating coils
2. Reheating coils
3. Steam coils for pan or grid humidifiers

Depending on the results desired, the admission of steam to these coils is regulated by either two-position or modulating steam-control valves under the command of room thermostats or duct controllers, or, in the case of the humidifiers, the steam valves are regulated from room or duct humidity controllers.

Cooling coils may be generally divided into two sections; one section producing sensible heat cooling and the second section producing dehumidification.

393. All Year Control *

With the increasing emphasis on air conditioning as a year-round program, there will be an increase in the design and installation of systems which properly treat the air at all seasons. While the modernization type of job with an already existing heating system to which the cooling cycle equipment will be added may be operated in two phases, one to cover the heating cycle and the other to cover the cooling cycle, the installation of automatic control will make it desirable that these two phases be interlocked so that there will be a minimum amount of, if not a complete elimination of the need for, manual attention in arranging the system to function on either cycle. There are, during the days of late spring and early fall, periods within which the complete system may be required to change from the heating to the cooling cycle and back again during a single day. For instance, during these seasons it is often necessary that heat be supplied during the night or in the early morning or evening hours, but, during the mid-day, temperatures and humidity conditions are such that cooling cycle operation is desirable. Where the control system or the design of the primary equipment is such that it is a burdensome task to change the system in accordance with these needs, the satisfaction to the user cannot be as complete as in the case of systems which can and will make such necessary changes as frequently as required and in a completely automatic manner.

From the standpoint of the automatic controls, such automatic change-over is entirely practical and available provided, of course, that the air-conditioning equipment itself has been selected and arranged with this possibility in mind.

* *Courtesy Minneapolis Honeywell Corp.*

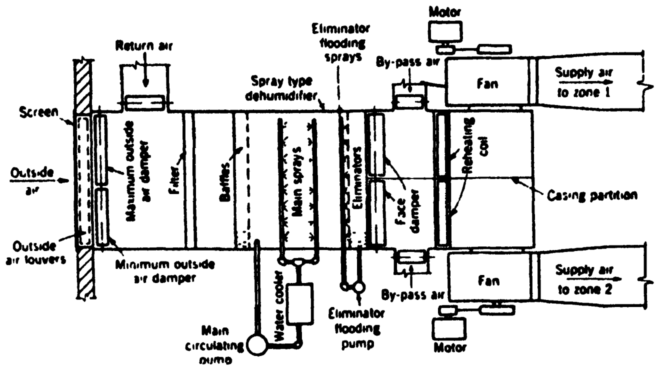


FIGURE 86

Central System Using Multiple Fans with By-Passes and Reheating for Zoning

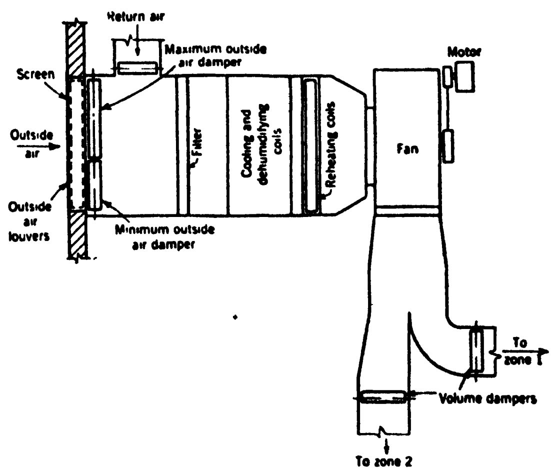


FIGURE 87

Central System with Zoning by Volume Control

394. Central Systems Versus Unit Coolers

The features that determine whether or not to specify a central system are:

1. Initial cost
2. Operating and maintenance costs
3. Comparative effectiveness
4. Space requirements
5. Characteristics of the load

Initial cost is too frequently given first consideration. A small increase in initial cost to provide better access to equipment, better airflow and distribution, and better zoning, usually will pay for itself.

Comparative operating costs should be compiled by the combined efforts of an air-conditioning engineer and cost accountant representing the purchaser. Cooling systems in general operate at maximum capacity less than 20% of the time and heating systems operate under design conditions only a few days of the year. Proper zoning and selection of equipment tend to reduce operating costs.

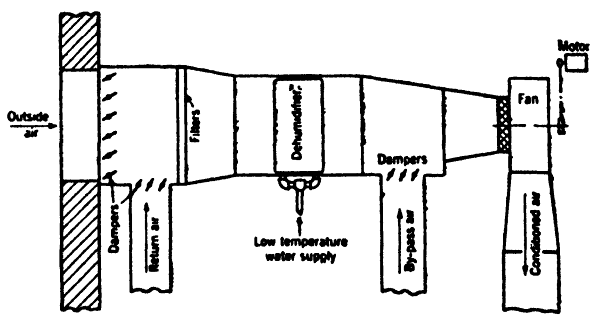


FIGURE 88

Summer Dehumidifying Equipment with By-Pass

One great advantage of the central system lies in its ability to diffuse odors and smoke which may occur in parts of the system, so that the outside air is determined by the average instead of the sum of the peak requirements.

The characteristics and requirements of the load are frequently the deciding factors. Wide non-simultaneous variations in load between spaces indicate the necessity of zoning. Isolated spaces having a short-time occupancy, as a directors' meeting room, or a one-meal cafeteria may be handled by units to advantage.

The maintenance of conditions during intermediate seasons, without use of refrigeration, must be considered.

The question of space requirements may rule out either the central system or the unit system.

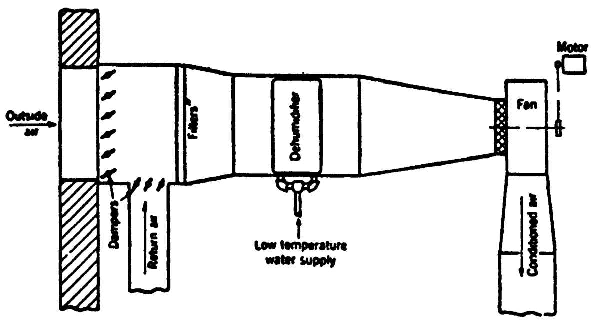


FIGURE 89

Simple Dehumidifying Apparatus

395. Design Conditions

In the discussion of comfort zones and comfort lines it was pointed out that the most ideal conditions for comfort in a room existed on the comfort line when 97 to 98% of all of the occupants would be satisfied with the air conditions but that practical considerations made it necessary under some outside conditions to keep the room only within the comfort zone and not necessarily on the comfort line.

Admitting that the comfort line is the place where the greatest percentage of occupants will be satisfied, or comfortable, what are the conditions which would prevent the room being held on the comfort line at all times?

396. Condensation on Window Glass

In winter air conditioning it has been found that when the outside temperature goes to low levels the air in contact with the window glass will be reduced below its dew point and will deposit condensate on the glass; this is commonly termed *sweating* of the window; while this does not usually occur in rooms not artificially humidified, the only reason it does not occur in such cases is that the humidity is so low in the room that the air in contact with the glass is not reduced below its dew point, and, hence, no condensation occurs. But this condition can be obtained only by a reduction of the humidity in the room to a point below that of the comfort line and, in fact, leaving the room without any added humidity at all.

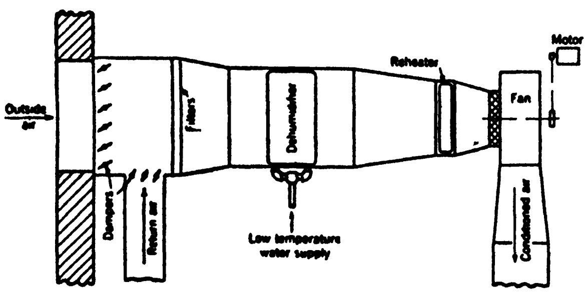


FIGURE 90

Dehumidifying Equipment with Reheater

When the outside temperatures go to extremely low levels (e.g., near zero) it will be found that the residence window, for example, will begin to form frost on the glass at about 5° F outside and this also is without the addition of any artificial humidity and with a relative humidity in the room of only about 20%; with lower outside temperatures the humidity in the room must be even less to prevent window frosting. Frosting on the windows is simply condensate deposited on the glass at a time when the outside temperature is sufficiently low to make the *inside surface of the glass below 32° F or freezing point.* When this condition obtains the condensate deposited on the glass is immediately frozen as it is deposited, and appears in the form of frost.

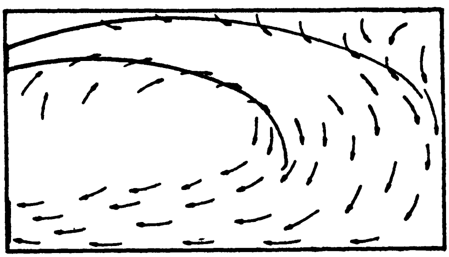
397. Objections to Condensation and Frosting

There are several very strong objections to condensation on windows and even more to frosting. In the first place, it destroys the transparency of the glass, which, in a shop window is a serious matter; in the second place, if artificial humidification is maintained so as to hold the humidity in the room on the comfort line, the condensate will continue to form, since it is re-supplied from the moisture constantly added to the air by the humidifying equipment. In cases of frosting the frost will continue to build up in the same manner as on a refrigeration coil and the windows will have to be defrosted; this can only be accomplished by a rise in the outside temperature or by increasing the room temperature so that the frost will melt off. The first method is uncertain, as no one knows when the outside temperature may rise, and the second method may be impossible of accomplishment if the heating system is not oversized. In either case a considerable quantity of water is produced in the defrosting process.

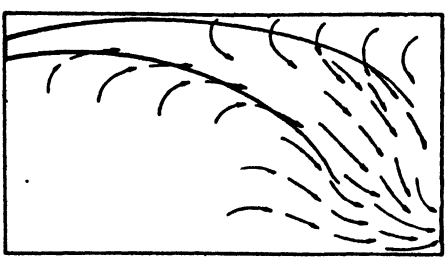
When defrosting finally occurs, by whatever process it may be accomplished, there is a lot of damage done to the window woodwork, the sills and the wall under the sill; sometimes the water will even run down onto the floor. This makes it evident that in winter air conditioning it is not practical to keep the room conditions on the comfort line unless double windows are used, in which case the humidity in the room could be kept at 50% with an outside temperature as low as 5° F without condensation or sweating, and as high as 40% even to —10° F. As the usual construction of buildings in this country is with single windows and not double windows, it follows that, in the ordinary case, the room humidity must be reduced as the outside temperature falls until at very low temperatures it must be cut off altogether.

398. Temperature Shock in Summer Cooling

In summer cooling also it is impossible to keep the room on the comfort line when extremely high outside temperatures are reached, owing to *temperature shock;* this means the possible deleterious effects on persons entering the cooled room from the high outside temperatures. A person perspiring profusely, on entering a cold room, may suffer a severe chill.



ELEVATION VIEW CORRECTLY LOCATED RETURN OUTLET



ELEVATION VIEW OF IMPROPERLY LOCATED RETURN OUTLET

FIGURE 91

Return Outlets

Air conditioning is developed for comfort and not to produce sickness, and as a result, the room temperature cannot be kept

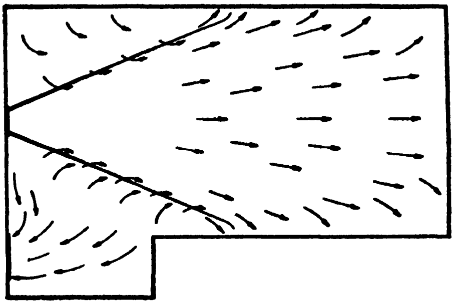


FIGURE 92

Plan View Correctly Located Return Outlet Eliminating Stagnant Space

on the comfort line when extremely high outside temperatures obtain. It has been found that a 15 degree temperature drop is the limit of safety in avoiding temperature shock, and this is why, in figuring the *heat gain* for a room in the summer, only a 15 degree difference between the inside and outside temperatures is used.

No way of combating this condition has been found so far except the use of *temperature locks* (consisting of rooms at a temperature half-way between the room, which is kept on the comfort line, and the outside temperature, which may be anything from 100° F down), but the period of time which the entering occupant must spend in such a room before becoming acclimated to the cooler temperature and in a condition to permit safe entrance to a still lower temperature is a serious objection; this probably would be at least one-half hour, and the same process would have to be repeated in reverse on leaving the cooled room, to avoid the danger of a heat stroke. From this it becomes apparent that places advertising "70 degrees the year round" are a real menace to health and air conditioning.

399. Room Conditions Which are Practical

It will be evident from the previous topic that it is practical to keep a room on the comfort line neither during extremely low outside temperatures nor during extremely high outside temperatures, and that room conditions must be modified to meet outside conditions of temperature in both summer and winter conditioning.

In winter the humidifying must be reduced as the temperature lowers with the dry-bulb temperature suitably raised to compensate, even if the relative humidity drops below that permitted by the comfort zone; in summer the temperature of the room must be not more than 15 degrees lower than the outside temperature if temperature shock is to be avoided. From these considerations, certain room conditions are set up as the basis of air-conditioning design; for instance, in the case of extremely low outside temperatures in winter, the room may have to be held at 73° F dry-bulb, and no humidity added, while in the case of extremely high summer conditions the room may have to be raised to 80° F, but the proper humidity percentage can be maintained without difficulty.

400. Room Conditions Assumed for Air-Conditioning Design

For the purposes of an air-conditioning design the winter condition of the room is assumed to be 70° F with a minimum outside temperature about 10 degrees above the lowest recorded outside temperature, because the lowest outside temperature seldom occurs. When it does occur, this extreme cold is usually of short duration, and the warmth stored in the building, which sometimes is called "temperature lag", is sufficient to carry the building over this short period without a serious drop in the inside temperature.

In cooling in summer, the outside temperature is assumed to be about 5 degrees below the highest recorded temperature and a relative humidity close to 50% is assumed to accompany this high temperature. It is usually found that high outside temperatures and high outside humidities do not go together; that is, you may have one, or the other, but not both, at the same time.

To reduce this to definite figures, in this locality (New York) the room temperature in winter is assumed to be 70° F and the outside temperature at 0° F; the additional heat required for humidifying will not be required at the lowest outside temperature and when the temperature is not at its lowest level, there will be sufficient excess heat available to take care of the humidifying. In summer, the room is assumed to be at 80° F and 50% relative humidity when the outside temperature is at 95° F and 50% relative humidity. On this basis, or something very close to it, all air-conditioning calculations are made and have proved adequate.

401. Inside Changes Made in Room Conditions

The conditions in the room may, and will, be modified by the occupants of the room and processes, if any, carried on in it. For instance, it is known that each person adds a certain amount of heat, some of which is in the form of sensible heat and the balance of which is in the form of latent heat; the sensible heat increases the air temperature in the room and the latent heat increases the relative humidity of the air after it has entered the room. As explained previously, no attention to items of heat addition in the room is made in winter air conditioning owing to the fact that it is necessary to heat the room before any occupants arrive, but, in summer cooling, the heat and moisture added by the occupants plus any heat and moisture received from any other source, must be removed by the air and absorbed into the refrigeration system if the room conditions are to be maintained and a heat balance secured.

402. Supply Air Conditions Required

In order to do this in summer, the air must enter at a temperature sufficiently below that of the room to cool the room so that it will have the desired temperature. It also should enter with a moisture content low enough so that, after picking up the moisture in the room, the relative humidity will be at the desired point. These two features are those upon which all air conditioning for cooling is based. On the other hand, however, the air entering the room should not be more than 15 degrees below the temperature being maintained in the room. Some people use a 20 degree difference (but great care in the introduction of the air is necessary in this case to avoid the possibility of cold drafts), while others use air only 10 degrees below room temperature to make absolutely certain that cold drafts will not occur. However, 15 degrees may be considered as a fair compromise between the two and will be used for the basis of design here.

403. Quantity of Room Air Required

If it is assumed that the room air does not enter the room at more than 15 degrees below the room temperature, then the quantity of sensible heat to be removed from the room determines the quantity of air to be circulated, and from this quantity the size of the equipment, ducts, etc., is determined. For instance, it will be found that roughly 4 cu ft of air will be required per hour for each Btu of sensible heat to be removed in one hour, but, if the air enters at only 10 degrees below the room temperature, the ratio will be increased to about 6 cu ft per hour for each Btu removed per hour. It is customary to add 10% to the theoretical quantity of air calculated to allow for emergencies so that it will be found with air introduced at 15 degrees below room temperature the amount of air required per Btu removed will be

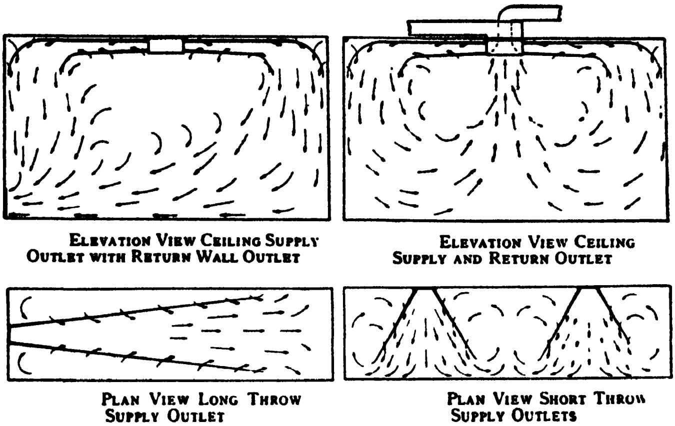


FIGURE 93

Typical Air Distribution Diagrams

$$\frac{1 \text{ Btu} \times 55 \text{ cu ft}}{15^\circ \text{ F}} \text{ or } 3.67 \text{ cu ft plus } 10\% \text{ or } 3.67 \text{ plus } 0.37 \text{ or } 4.04 \text{ cu ft}$$

while with air at 10 degrees below room temperature the requirement will be

$$\frac{1 \text{ Btu} \times 55 \text{ cu ft}}{10^\circ \text{ F}} \text{ or } 5.50 \text{ cu ft plus } 10\% \text{ or } 5.50 \text{ plus } 0.55 \text{ or } 6.05 \text{ cu ft}$$

404. Recirculation of Air From the Room

In air conditioning it has been found so expensive to throw away the exhaust air that this procedure is impractical; therefore, the exhaust from the room is carried back to the apparatus for reconditioning. It is not practical in most cases to use recirculated air for more than 75% of the total air supplied, because if all of the air is recirculated and no outside air is added, odors will build up and unpleasant room conditions result. Consequently, only a portion of the air is recirculated, this in most cases being about 75% of the total, the other 25% being outside air; by this means, a better air supply is maintained and noticeable odors are eliminated.

Supplying 100% of the air required and recirculating only 75% means that 25% of the air supply must find its way out through the doors, window-cracks and other crevices, resulting in the maintaining of a positive pressure in the room, or building; this positive pressure does much to prevent inward infiltration. As a result, the room conditions are not disturbed by opening doors or the leakage of hot air around the windows where it cannot be treated.

This arrangement makes it necessary for the air-conditioning equipment to handle about 25% of the total air supplied as outside air, and about 75% of the total air supplied as recirculated air. The saving made in the size and cost of operation of the refrigeration plant is considerable when recirculation is used, so much so that an air-conditioning installation without recirculation is almost always classified as a poorly designed system owing to its cost, both initial and operating. Recirculation is undesirable in high class restaurants and smoking rooms. Allow at least 40 cu ft of fresh air per occupant in the ordinary uncrowded smoking room.

405. Work Done on the Supply Air

The supply air constitutes 100% of all the air required to cool the room, but it is made up of 25% outside air and 75% recirculated air. Therefore, recirculation saves not on the fan power, but on the refrigeration load, since the only treatment required by the recirculated air is its cooling back to the original supply air temperature, i.e., removing the heat it has picked up in the room, and its dehumidifying to remove the moisture picked up in the room (which seldom exceeds 10 to 12 grains per pound) and ¾ of the total air has only this small amount of work performed on it.

The outside air, on the other hand, must be cooled as a maximum from 95° F outside to 65° F (the temperature of the supply air), and must be dehumidified to the extent of 95 degree air at 50 per cent relative humidity (or 128 gr per pound) down to 80 degree air at 50 per cent relative humidity (or 78 gr per pound), plus 10 to 12 gr added in the room, resulting in a dehumidification on the outside air of 60 to 62 gr per pound against only 10 to 12 gr with the recirculated air. It is plainly evident that much less work must be expended on the recirculated air per pound than on the outside air, but it usually will not be found practical to increase the amount of recirculated air over 75% of the total, regardless of how much added economy might result in recirculating a greater precentage of the air.

The cooled and dehumidified air should be supplied at as many points in the room as are necessary to secure good distribution of this air. In general, it might be stated that the greater the number of inlets used, the better will be the distribution, the more even will be the room conditions and the less probability there will be of complaint from cold drafts. Greater care must be exercised with air 20 degrees below the room temperature than with air 15 degrees below, and still less with air 10 degrees below, but in all cases the supplies should be so located that direct currents of cold air toward the floor are not introduced, and thorough mixture with the room air is effected before the incoming air reaches the occupants.

406. Outlets From the Room

Outlets for recirculated air are not nearly so important, as they produce no cold drafts and may be placed, in most cases, where conditions permit, provided that no dead pockets are developed. With properly designed fixtures it is even possible to sometimes place the outlets on the ceiling, especially if the room is of low ceiling height. The favorite location, however, is on the side walls near the floor at strategic positions, so as to induce a flow of air toward the far corners of the room which otherwise might develop into dead pockets. In theaters, mushroom vents are often placed under the seats, riser grilles in the balcony steps, large grilles on the side walls and a small amount of air taken off at the orchestra pit, or in the place where the orchestra pit used to be. The prevailing direction of air flow must be against the faces of the occupants and *not against the backs of their necks;* this is highly important, as the face is accustomed to facing air movement, whereas the back of the neck is very susceptible to cold drafts. From this it follows that the major portion of the air should be introduced in a theater *toward the front of the house or balcony* and that the principal recirculation outlets should be toward the rear of the house, or balcony, as the case may be.

407. Deflectors and Plaques

Where large quantities of air are to be introduced into a room such as an auditorium or theater, the air is often discharged downward from the ceiling opening onto a deflector or plaque which turns the air sideways and discharges the air radially in all directions on a horizontal plane. This causes an excellent intermixture between the entering air and the room air so that no cold drafts are likely to occur, regardless of the quantity of air introduced through a single opening. It is customary to keep these quantities down to between 2,000 and 3,000 cfm, however, in order to maintain a sufficient number of openings to insure good distribution.

408. Cove and Cornice Inlets

Slots in the coves in the ceiling, or in cornices, also are used to distribute the air when the ceiling design is such as to facilitate this method. In this scheme a long slot is used with an air velocity of between 200 and 300 ft per minute and a long box, or plenum chamber, is built back of the slot. The supply duct is connected into the plenum chamber and such boxes are often divided into sections, with a duct connection to each section to insure that the entire length of the slot will be served; each duct connection has a volume damper to control the air flow to the particular portion of the slot served.

All methods work toward the same objectives of introducing the air so as to feed the whole room equitably as far as possible, removing the air in such a manner as to produce an air movement against the face and preventing dead pockets in the room where circulation of the air will be poor or non-existent.

TABLE 180

ROOM HEAT LOAD RATIOS FOR TYPICAL SUMMER COMFORT CONDITIONING

Room Heat Load Ratios [a]	Typical Classes of Room Service or Load: No Occupants or Sources of Vapor	Private Office or Residence	Restaurant or Crowded Office	Auditorium at Capacity or Crowded Restaurant	Ballroom at Capacity
$\frac{\text{Sensible Heat}}{\text{Total Heat}}$	1.00	0.90	0.80	0.70	0.60
$\frac{\text{Total Heat}}{\text{Sensible Heat}}$	1.00	1.11	1.25	1.43	1.67
$\frac{\text{Latent Heat}}{\text{Total Heat}}$	0	0.10	0.20	0.30	0.40
$\frac{\text{Total Heat}}{\text{Latent Heat}}$	...	10.00	5.00	3.33	2.50
$\frac{\text{Sensible Heat}}{\text{Latent Heat}}$	...	9.00	4.00	2.33	1.50
$\frac{\text{Latent Heat}}{\text{Sensible Heat}}$	0	0.11	0.25	0.43	0.67

[a] The over-all heat load ratio for the dehumidifier will be different from the heat load ratio for the room. The extent of the difference will depend on the quantity and condition of the outside air used, upon the magnitude of the duct losses, and upon whether or not reheat or by-pass are used.

TABLE 181

DRY-BULB TEMPERATURE OF AIR AT ROOM INLETS

To Maintain Typical Room Conditions of 80° F Dry-Bulb, 50 per cent Relative Humidity

Sensible Heat / Total Heat	1.00	0.90	0.80	0.70	0.60
Air entering saturated[a]	60.0	58.6	56.5	53.0	35.0
Air entering with 4° F wet-bulb depression	66.5	65.4	64.1	61.8	56.0
Air entering with 8° F wet-bulb depression	72.6	72.1	71.6	70.5	68.0

[a] Typical air conditions leaving the central conditioner are: With spray dehumidifier, 0 to 2° F wet-bulb depression. With surface-type dehumidifier, 1 to 6° F wet-bulb depression. With by-pass or reheat, 4 to 10° F wet-bulb depression.
Courtesy of A.S.H.V.E.

409. Cooling Load

Computation of the cooling load is outlined in Chapter XIV. Many items of heat gain are variable and do not attain their maximum values simultaneously. Maximum outdoor wet-bulb temperature is seldom coincident with maximum solar heat gain, especially on an eastern exposure. A large difference in the timing of peaks indicates the advantage of zoning.

410. Heating Load

Computation of the heating load is outlined in Chapter XIII. Many of the factors outlined under this subject of cooling load, such as zoning, diversity and non-simultaneous peaks, apply in reverse here due to the heating requirement instead of the cooling requirement. Where heating is concerned, it is necessary not only to heat a building to its design condition, when there is a small fraction of its normal occupancy, but it is necessary also to provide the capacity for heating the building quickly after a shut-down due to sudden warm weather after a cold snap. In many cases it requires less fuel to keep a building at a temperature of 50° to 65° F for some time than to shut down the system and bring the temperature up again.

411. Apparatus Dew Point

A method of locating the apparatus dew point will be outlined in the next chapter. Briefly the method consists of estimating the net energy gain in cooling (or loss in heating) per hour and the net moisture gain (or loss) per hour from data on exposure, construction, appliances, occupants, infiltration, ventilation requirements and outside design conditions. In computing the quantities of energy and moisture introduced and displaced by the ventilating air, only that portion of the ventilating air admitted directly to the conditioned space is considered. In actual practice with commercial apparatus, it is rarely possible to obtain complete saturation and there may be a temperature difference of several degrees between the dry-bulb and wet-bulb temperatures of the air returned to the conditioned space. This causes no difficulty. The only special significance of the apparatus dew point is that it provides a convenient control point from which to regulate the operation of the apparatus.

In winter a degree of saturation in excess of 30% is seldom required and a low saturation efficiency may be necessary where the full summer fan capacity is maintained.

When it becomes necessary to increase the saturation efficiency of the sprays in the humidifier, the spray water may be heated. Present day practice for spray type dehumidifiers assumes that the air leaves the dehumidifier at a temperature 1 to 2 degrees higher than the temperature of the spray water leaving the dehumidifier.

412. Designing the System

The general procedure for the design of a central system is as follows:

1. Calculate the sensible heat and latent heat gains for each room or space to be conditioned.
2. Establish the temperature of the air leaving the supply inlets.
3. Calculate the quantity of air to be circulated.
4. Estimate the temperature rise in the duct system.
5. Determine the volume of outside air to be introduced.
6. Calculate the heat to be removed by cooling and dehumidifying apparatus, and the type and arrangement of apparatus to be used.
7. Calculate the size of the reheating equipment, if any.
8. Select cooling and dehumidifying equipment, and refrigerating and reheating equipment, from manufacturers' data and performance curves.
9. Design the air-filtering and distribution system, the air outlets and inlets.
10. Calculate the total static pressure of the system.
11. Select the fan, motor and drive.
12. Select the pump and motor.
13. Design the control system.

Chapter XXI

AUDITORIUM COOLING AND THE BY-PASS SYSTEM

An essential element, after cooling and saturating the air passing through a dehumidifier, is the plan adopted to reheat the air to the desired temperature, whether the load is a maximum or minimum. The temperature to which the conditioned air must be heated depends on the outside air temperature and upon the temperature difference to be maintained between the conditioned space and the dew point of the air supply, as this difference is required by the humidity control.

The by-pass consists of an arrangement of ducts, controls and dampers which will permit air to pass around the dehumidifier or conditioner without being heated. This method provides three important features which may be employed separately or simultaneously. The first is temperature control at a constant volume of air supply, by automatically adjusting the by-pass damper. The second feature is the maintenance of a lower effective temperature difference between the air supplied to the room and the desired room temperature, than could be obtained if the air at the apparatus dew point were supplied by a fixed by-pass. The third feature inherent in this method is a decrease in refrigeration requirements directly proportional to the decrease in load.

413. Room Cooling and Auditorium Cooling

Comparing the problem of conditioning air for processes in a factory with conditioning air in an auditorium or living room for human comfort the following differences will be noted.

(a) Dew point in a factory or for an industrial process varies with the process but dew point in a theater or living room is determined by experience, research and general opinion as to what condition is most comfortable for human beings in a particular state of rest or labor. It is the opinion of heating and ventilating engineers that an average inside condition of 80° F dry-bulb and 50% relative humidity in summer, corresponding to a 60° F dew point, with an air motion obtained by good distribution gives a fair degree of comfort when outside wet-bulb varies from 75° F to 78° F. With greater air turbulence an inside dry-bulb temperature of 85° F has been tried with fair success.

(b) The maximum moisture load is determined by the number of persons in the conditioned space. At 80° F dry-bulb a man having 19.5 sq ft of body surface will produce perspiration and expiration, approximately, of 1,000 gr of moisture per hour, absorbing 150 Btu per hour in the form of latent heat, which raises the inside dew point. His body will also deliver 250 Btu per hour in sensible heat, which heats the circulating air.

(c) The amount of circulating air is determined by local ordinances and varies from 25 cfm to 30 cfm per person. A certain proportion must be taken from the outside, the balance being recirculated. If less air is circulated the air motion will be less and lower dry-bulb temperatures must be assumed. A compromise must be decided upon between proper air motion and corresponding temperatures. A variable by-pass is usually sized to handle 50% of fan capacity, irrespective of the calculated proportioning desired. In winter the by-pass damper is kept closed, but in all heating calculations allowance must be provided for 20% leakage through all closed louver dampers.

(d) In midseasons 100% outside air may be desired, therefore the fresh air intake is designed with an area to handle full fan capacity. There also arises a situation in very cold weather when it is desirable to close the fresh air damper and circulate

100% return air, therefore the return air inlet to the dehumidifier should provide for a minimum of 75% full fan capacity thus allowing for an estimated leakage of 20% in the fresh air damper when it is closed. A constant proportion of two volumes of outside air to one volume of return air, reduced according to the number of seats occupied, generally proves satisfactory. When all seats are occupied, many theaters operate with the following proportions

Summer—full load
30% outside air
10% return air
60% by-pass air
Summer—half load
15% outside air
5% return air
80% by-pass air

these proportions allow 9 cu ft of fresh air per person.

414. Auditorium Problem Using Spray-Type Dehumidifier

Cool 30,000 cfm air of which
12,000 cfm is fresh air and
18,000 cfm is recirculated air

to supply 500 men, 400 women, 100 children, and maintain comfortable conditions in the theater.

Note: Heat content values are based on average computations of various authorities.

Outside Air		Inside Air (At Exhaust Grilles)
95° F	Dry-bulb	80° F
75° F	Wet-bulb	67° F
37%	Relative humidity approximately	50%
66° F	Dew point (approximately)	60° F
7.1	Grains per cubic foot dry air	5.8

Outside Air		Inside Air (At Exhaust Grilles)
0.0136	Pounds moisture per pound dry air	0.011
14.30	Cubic feet per pound mixture	13.85
13.98	Cubic feet per pound dry air	13.6
23.8	Sensible heat per pound dry air	19.79
14.5	Latent heat per pound dry air	11.8
38.3	Total heat per pound dry air	31.59

415. Determination of Dew Point of Air Entering Auditorium

The heat input due to body heat of a mixed population is assumed to be 10% less than that due to the body heat of male adults.

90% of 150 = 135 Btu per person latent heat per hour

90% of 1,000 = 900 gr moisture per hour per average person

900 ÷ 60 = 15 gr per minute

15 × 1,000 = 15,000 gr total moisture per minute

15,000 ÷ 7,000 = 2.14 lb moisture per minute

30,000 cfm ÷ 13.6 cu ft per pound = 2,210 lb dry air per minute

2,210 × 0.011 = 24.5 lb moisture per minute in air leaving auditorium

24.5 − 2.14 = 22.36 lb moisture per minute in air entering

22.36 ÷ 2,210 = 0.0101 lb moisture per minute per pound dry air entering

which corresponds to a dew point of 57½° F. Therefore in crowded auditoriums it is safe to expect a 2½ degree rise in dew point due to body moisture. Air must enter the theater, therefore, at a dew point 2½° lower than it has when leaving, i.e., 60° F − 2½° F = 57½° F, to maintain a condition of 80° F dry-bulb, 67° F wet-bulb, 50% relative humidity and 60° F dew point. Latent heat per pound air entering is 10.8 Btu.

416. Determination of Dry-Bulb Temperature of Air Inlet

Local ordinances demand 30,000 cfm or 2,210 lb dry air per minute for 1,000 people. The sensible heat of a male adult is 250 Btu per hour; for a mixed population allow 10% less sensible body heat or 225 Btu per hour per person or 3.75 Btu per minute.

1,000 × 3.75 = 3,750 Btu per minute sensible body heat. Heat input due to lights, wall losses etc. may safely be assumed to be 25% of total sensible heat input or about one Btu per minute per seat, of which ½ Btu is assumed to be due to heat from lights.

3,750 ÷ 0.75 = 5,000 Btu total heat input per minute
5,000 ÷ 2,210 = 2.26 Btu per pound air (sensible heat)

Sensible heat per pound air at 80° F	19.79 Btu
Deduct sensible heat input	2.26 Btu
Sensible heat per pound air entering	17.53 Btu

From psychrometric table 17.53 Btu sensible heat corresponds to a dry-bulb of 70° F, which is the exact dry-bulb temperature of the air entering the theater. The latent heat having already been determined as 10.8 Btu and the sensible heat as 17.53 Btu determines a total heat of 28.33 Btu which corresponds to a wet-bulb of 62° F. The diffusion temperature difference is noted to be 10 degrees between inlet grilles and outlet grilles. The dry-bulb may be checked by the following formula:

$$2{,}210 \text{ lb air} \times 0.24 \text{ Btu sp ht} \times T_d = 5{,}000 \text{ Btu}$$
$$T_d = 9.8^\circ \text{ F}$$
$$80^\circ - 9.8^\circ = 70.2^\circ \text{ F}$$

417. Useful Data for Auditorium Cooling

Dew point usually specified, 60° F at exhaust grilles
Dew-point rise due to moisture from people, 2½° F
Dew point of air entering theater, 57½° F
Sensible heat per average person, 3.75 Btu per minute

Sensible heat input, lights and wall transmission, 1.00 Btu per minute per seat ($\frac{1}{2}$ Btu for lights alone)

Total sensible heat input per seat, 4.75 to 5.00 Btu per minute

Grains moisture per minute per seat, 15 grains

Pounds moisture per minute per seat, 0.00214 lb

Allow 10 watts per seat total lighting load; this will include lobby and stage lights.

Latent heat per seat, 2.24 Btu

418. Conditioning of Air Between Air Washer and Inlet Grilles

It has been determined that in summer the condition of the air at inlet grilles desired is 70° F dry-bulb, 57½° F dew point, 62° F wet-bulb. Now air from the air washer is discharged saturated and must therefore be heated by some external source to 70° F. If the ducts are uninsulated an average rise of 4° F may be expected, especially at grilles 100 ft distant from the washer. A heat exchanger may be placed in the path of the cooled air to cool the incoming fresh air and at the same time reheat the saturated air. In the following problem it is assumed that the reheating is done by means of 60% by-passed recirculated air, in this case 18,000 cfm. In this problem it is assumed that the ducts are insulated and there is no rise in temperature due to heat transmission through the ducts.

419. Determination of the Dew Point of the Air Leaving the Air Washer in Order to Obtain the Specified Mixture of 40% Fresh Air Through Washer and 60% By-Passed

It has been specified that 12,000 cfm fresh air are to be cooled from an outside condition of 95° F dry-bulb and 75° F wet-bulb. The air must leave the air washer saturated and at a dew point *D* unknown, to mix with 18,000 cfm recirculated air from the auditorium at 80° F dry-bulb, 67° F wet-bulb, 60° F dew point, the final mixture to enter the auditorium at

70° F dry-bulb, 57½° F dew point and 62° F wet-bulb.

12,000 ÷ 13.98 = 860 lb dry air per minute (fresh air)

18,000 ÷ 13.6 = 1,340 lb dry air per minute (recirculated)

Total heat content per pound fresh air = 38.3 Btu per pound dry air plus moisture

Total heat content recirculated air = 31.59 Btu per pound dry air plus moisture

Total heat content at 62° F wet-bulb mixture = 28.33 Btu entering auditorium

Note: 28.33 Btu is the sum of the sensible heat at 70° F dry-bulb plus latent heat at 57½° F dew point

Let H = Btu per pound air leaving air washer

$$860\,H + 1340 \times 31.59 = 2{,}200 \times 28.33$$

H = 23.0 Btu per pound dry air plus moisture

which corresponds to a 54° F dew point, wet-bulb and dry-bulb or saturation leaving air washer.

420. Refrigerating Effect Required

The dew point must be determined before the refrigerating effect is calculated. This has been done above, and the dew point is 54° F.

Heat content air entering cooler	= 38.3	Btu per pound dry air plus moisture
Heat content air leaving cooler	= 23.00	Btu at 54° F dew point
Heat abstracted per pound in cooler	15.30	Btu

$$\frac{800 \text{ lb air through cooler} \times 15.3}{200 \text{ Btu per minute per ton}} = 66 \text{ refrigeration tons}$$

equivalent to 2.2 tons per 1,000 cfm total circulation.

421. Second Method of Calculating Tons of Refrigeration

In a recirculating by-pass air cycle, refrigeration is required to reduce the moisture content of the air mixture entering the auditorium to the desired dew point, in this case 57½° F, by reducing fresh air to a dew point lower than 57½° F, to counteract the moisture regain from the people in the auditorium at a 60° F dew point; also to counteract sensible heat input from

(1) fresh air
(2) body heat of people
(3) wall and roof transmission losses
(4) lights
(5) fan horsepower
(6) pump horsepower

To condense moisture in 860 lb fresh air

0.0136 lb moisture per pound air at 95° F dry-bulb, 75° F wet-bulb
0.0091 lb moisture per pound air at 55° F dew point
0.0045 lb condensed per pound dry air

$860 \times 0.0045 = 3.88$ lb moisture per minute total

$3.88 \times 1{,}060$ Btu latent heat at 55° F $= 4{,}140$ Btu per minute to condense moisture

The sensible heat reduction of fresh air is

$860 \times (95-55) \times 0.24 = 8{,}256$ Btu per minute

Total Refrigerating Load

Moisture load	4,140 Btu per minute
Sensible heat	8,256 Btu per minute
	12,396 Btu per minute

$12{,}396 \div 200 = 62$ tons, this checks the previous calculation within 5%. To this add 2% or 1.2 tons to counteract pump horsepower and ½ hp per 1,000 cfm for fan horsepower, in this case 15 hp absorbing 3 tons refrigeration.

422. Pump Horsepower and Fan Horsepower

For rough estimating add 2% to the total refrigerating load to counteract the heating of water by the air-washer pump.

At 1½ lb static pressure a fan will consume about ½ hp per 1,000 cfm. One horsepower is equivalent to 2,545 Btu per hour or 42.4 Btu per minute. Allow one ton refrigeration for fans per 10,000 cfm total circulation.

423. Mixture of 40% Fresh Air and 30% Recirculated Air Passing Through Air Washer and 30% By-Passed

In the previous problem 60% was recirculated and by-passed so that it did not pass through the air washer. In this problem is considered the condition resulting when 860 lb fresh air plus 670 lb recirculated air are passed through the washer, cooled to a certain dew point, and then mixed with 670 lb of air by-passed around the washer, to be conditioned to certain specifications before entering the auditorium of the theater. Inasmuch as the heat to be extracted is the same as in the earlier problems, the quantity of refrigeration is constant, irrespective of the percentage by-passed. However, the dew point of the air leaving the washer will vary, depending upon the percentage by-passed, which affects the operating suction pressure in the evaporator of the refrigerating system.

Problem

Total heat of the fresh air entering the washer equals $860 \times 38.3 = 32{,}938$ Btu per minute

Total heat of the recirculating air entering the washer equals $670 \times 31.59 = 21{,}165$ Btu per minute

$\dfrac{32{,}938 + 21{,}165}{860 + 670} = 35.4$ Btu per pound mixture

13,158 Btu per minute extracted by refrigeration or 66 tons (fan horsepower and pump horsepower not considered.)

$\dfrac{13{,}158}{1{,}530} = 9.6$ Btu extracted per pound mixture passing through air washer

$35.4 - 9.6 = 25.8$ Btu total heat per pound of air leaving washer, saturated equivalent to 59° F dew point

40,945 Btu total heat leaving air washer

21,165 Btu total heat by-passing air washer

$\dfrac{40{,}945 + 21{,}165}{1{,}530 + 670} = 28.2$ Btu heat content per pound mixture entering auditorium

Sensible heat per pound at 59° F dew point = 14.43 Btu

Latent heat per pound at 59° F dew point = 11.4 Btu
Sensible heat per pound by-pass air = 19.79 Btu
Latent heat per pound by-pass air = 11.8 Btu

$$\frac{(860 + 670) \times (14.43) [+ 670] \times 19.79}{2{,}200} = 16.3 \text{ Btu per pound}$$

air entering auditorium (sensible) which corresponds to a dry-bulb temperature of 68° F. This is a little cool, showing that slightly more air should be by-passed.

28.2 Btu − 16.3 Btu = 11.9 Btu latent heat per pound air entering theater

equivalent to a dew point of 60° F. This is not quite satisfactory, showing that less recirculated air should pass through the washer.

424. Mixture of 40% Fresh Air and 60% Recirculated Air, All Passing Through Washer

860 × 38.3 = 32,938 Btu total heat in fresh air
1,340 × 31.59 = 42,330 Btu total heat in recirculated air

$$\frac{32{,}938 + 42{,}330}{2{,}200} = \frac{75{,}268}{2{,}200} = 34.0 \text{ Btu per pound air entering washer}$$

$$\text{Refrigeration per pound air} = \frac{13{,}158}{2{,}200} = 6 \text{ Btu}$$

34.0 − 6 = 28.0 Btu heat content per pound air leaving air washer saturated, equivalent to 61½° F dew point, which is an undesirable condition.

425. Determination of Dew Point for Various Fresh Air and Recirculated Air Mixtures

The condition of the air entering a room to maintain a comfortable condition in summer is to be 70° F dry-bulb, 62° F wet-bulb, 57½° F dew point at inlet grilles. The outside air is at a condition of 95° F dry-bulb and 75° F wet-bulb. The air leaving the room at exhaust grilles is at 80° F dry-bulb, 67° F wet-bulb, 60° F dew point and 50% relative humidity; this air returns to the air washer and part of it is mixed with air after leaving the washer or dehumidifier. The following problems consist of determining the resultant dew point of the cooled air leaving the air washer prior to mixing. The dew point naturally varies with the percentage of by-passed air and the smaller the amount of air passing through the cooler the lower the dew-point temperature of the air leaving the cooler, requiring a lower suction pressure of the refrigerant, and extra compressor displacement. Assume the following variations in the amounts of by-passed air and fresh air

(a) When 25% by weight fresh air passes through washer and 75% is by-passed
(b) When 33⅓% by weight fresh air passes through washer and 66.7% is by-passed
(c) When 40% by weight fresh air passes through washer and 60% is by-passed
(d) When 50% by weight fresh air passes through washer and 50% is by-passed
(e) When 60% by weight fresh air passes through washer and 40% is by-passed
(f) When 75% by weight fresh air passes through washer and 25% is by-passed
(g) When 100% by weight fresh air passes through washer and 0% is by-passed

Assume a total circulation of 1,000 cfm which at 95° F dry-bulb, 37% relative humidity and 75° F wet-bulb weighs 71.5 lb.

Problem (a)

25% of 71.5 = 17.9 lb fresh air
75% of 71.5 = 53.6 lb recirculated air
Heat content return air at 67° F wet-bulb = 32.10 Btu per pound
Heat content air entering room at 62° F wet-bulb = 28.27 Btu per pound

Heat content air leaving washer or cooler = H Btu per pound

$17.9\ H + 53.6 \times 32.10 = 71.5 \times 28.27 = 2{,}030$

$H = 17.5$ Btu per pound air

which corresponds to a dew point of approximately 45° F.

Problem (b)

$33\frac{1}{3}\%$ of $71.5 = 23.83$ lb fresh air

66.7% of $71.5 = 47.67$ lb recirculated air

$23.83\ H + 47.67 \times 32.10 = 2{,}030$

$H = 21$ Btu per pound air

which corresponds to a dew point of approximately 51° F.

Problem (c)

40% of $71.5 = 28.6$ lb fresh air

60% of $71.5 = 42.9$ lb recirculated air

$28.6\ H + 42.9 \times 32.10 = 2{,}030$

$H = 22.4$ Btu per pound

which corresponds to a dew point of approximately 54° F.

Problem (d)

50% of $71.5 = 35.75$ lb each fresh air and recirculated air

$35.75\ H + 35.75 \times 32.10 = 2{,}030$

$H = 24.2$ Btu per pound air

which corresponds to a dew point of approximately 57° F.

Problem (e)

60% of $71.5 = 42.9$ lb fresh air

40% of $71.5 = 28.6$ lb recirculated air

$42.9\ H + 28.6 \times 32.1 = 2{,}030$

$H = 26.0$ Btu per pound air

which corresponds to a dew point of approximately 59° F.

Problem (f)

75% of $71.5 = 53.6$ lb fresh air

25% of $71.5 = 17.9$ lb recirculated air

$53.6\ H + 17.9 \times 32.1 = 2{,}030$

$H = 27$ Btu per lb

which corresponds to a dew point of approximately 61° F.

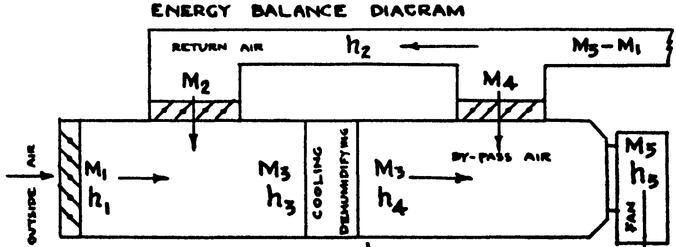
TABLE 182

ADIABATIC MIXING OF AIR STREAMS

% OUTSIDE AIR THROUGH COOLER HEAT CONTENT 38.3 BTU PER LB	% BY-PASS AIR AROUND COOLER HEAT CONTENT 32.1 BTU PER LB	% TOTAL FAN CIRCULATION HEAT CONTENT 28.27 BTU PER LB ENTERING ROOM	HEAT CONTENT AIR MIXTURE LEAVING COOLER BTU PER LB. SATURATED	MINIMUM DEW POINT LEAVING COOLER — ALL RETURN AIR BY-PASSED
M_1 lbs. h_1 BTU	$M_5 - M_1$ lbs h_2 BTU	M_5 lbs. h_5 BTU	M_3 lbs. h_4 BTU	t_{h4} DEGREES
25 %	75	100	17.5	45
26	74	100	18.0	46
30 NOTE	70	100	19.4	48 1/2
33.3	66.7	100	20.7	51
35	65	100	21	51 1/2
38	62	100	21.8	53
40	60	100	22.4	54
42	58	100	22.7	54 1/2
45	55	100	23.3	55 1/2
48	52	100	23.4	56 1/2
50	50	100	24.2	57
52	48	100	24.5	57 1/2
55	45	100	24.8	58
57	43	100	25.1	58 1/2
60	40	100	26	59
65	35	100	26.1	60
70	30	100	26.4	60 1/2
75	25	100	27	61
80	20	100	27.1	61 1/2
85	15	100	27.4	62
90	10	100	27.6	62 1/4
100	0	100	28	62 1/2

NOTE :- 10 % RETURN AIR THROUGH COOLER RAISES DEW POINT 5°F. APPR.

ENERGY BALANCE DIAGRAM



M = LBS. OF AIR h = TOTAL HEAT

$$\frac{h_2 - h_3}{h_3 - h_1} = \frac{M_1}{M_2} \qquad M_1 + M_2 = M_3 \qquad \frac{h_2 - h_5}{h_5 - h_4} = \frac{M_3}{M_4}$$

$$M_3 + M_4 = M_5$$

$$h_3 = \frac{M_2 h_2 + M_1 h_1}{M_1 + M_2} \qquad \text{D.P.} \propto h_4 = \frac{M_3 h_5 + M_4 h_5 - M_4 h_2}{M_3}$$

GOOD PRACTICE

$$\frac{M_1}{M_2} = 3 \qquad M_1 = 0.30\,M_5 \qquad M_5 - M_1 = M_2 + M_4 = 0.70\,M_5$$

Problem (g)

$$100\% \text{ of } 71.5 = 71.5 \text{ lb fresh air, no recirculated air}$$

$$71.5\ H = 2{,}030$$

$$H = 28.0 \text{ Btu per lb}$$

which corresponds to a dew point of approximately 62.5° F

CHAPTER XXII

INDUSTRIAL APPLICATIONS

In comfort air conditioning the ratio of sensible heat and latent heat (of moisture) removed is practically constant in places of assembly. Although it is true that outdoor conditions may have some effect on the ratio, in most cases it is a small percentage of the heat generated by the occupants. However, in industry the sensible heat load and the moisture load bear no simple relation to each other as in comfort cooling. Where large quantities of air at small temperature differentials are required for proper circulation in comfort cooling, lending economy to the by-pass method, in industrial applications only refrigerated and dehumidified air may be necessary. A complete knowledge of all the factors involved is necessary before industrial air-conditioning problems can be solved correctly. A detailed survey of the project must be made, as explained in Chapter XXIV.

426. Industries Requiring Air Conditioning *

A few of the industries in which air conditioning plays an important part (and its 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

* *Courtesy of A.S.H.V.E.*

electrical apparatus, storage battery plates and the rubber containers.

Bakery. Flour storage, yeast and ingredient storage, mixers, fermentation rooms, make-up room, proof boxes, load 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 gelatins.

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 manufacture 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, enzymatic 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 finishing 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 weighting 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.

427. Atmospheric Conditions in Industry

It is generally recognized that relative humidities of 50% or less are on the dry side and therefore conducive to drying out, increased brittleness of fibrous materials, low regains in hygroscopic materials, tendencies toward dust liberation and prevalence of increased static electricity. Relative humidities higher than 50% are considered on the damp side and are conducive to softness and pliability, high moisture regain, decreased static and tendencies toward reduced product dust, preventing a loss in weight of material in process. Desired conditions vary in many processes, according to the stage of the processing cycle, from the raw material to the finished product. Cotton textiles begin with a low relative humidity in the carding and picking rooms, and, after passing through various intermediate steps with a gradual increase in relative humidity, the product is subjected to 75 to 85% in the final wearing stage. Other processes are encountered that require reverse treatment e.g., gelatin and glue products.

In Table 184 the temperatures listed in some cases have no direct bearing on the process itself, except as they affect the efficiency of the employees and, through them, the quality of workmanship and uniformity of product. An automobile assembly line may be included in this category. Temperatures and relative humidities listed in Table 184 should be analyzed by an experienced engineer for scientific adaptation to the problem in question. Conditions generally accepted for industrial processing and for general storage are listed in these tables. Many processes require refrigeration only.

428. Drying

Many phases of drying may be discussed under the subject of industrial air conditioning, especially where temperature and humidity changes bring about some definite change in the physical characteristics of the product, as in the coating of candy, pills and chewing gum with sugar syrup.

429. Air-Conditioning Problems

The five fundamental problems may be classified as follows:

1. Control of regain
2. Control of rate of chemical reaction
3. Control of rate of biochemical reaction
4. Control of rate of crystallization
5. Elimination of static electricity

TABLE 183

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 10	20	30	40	50	60	70	80	90	Authority
Natural Textile Fibers	Cotton	Sea island—roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino—skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes—skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun—yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal—rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa'
Rayons	Viscose, Nitrocellulose, Cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fiber	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood pulp—24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood pulp—3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% rag—1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.

REGAIN OF HYGROSCOPIC MATERIALS (*continued*)

Classification	Material	Description	Relative Humidity—Per Cent 10	20	30	40	50	60	70	80	90	Authority
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 Fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel .		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulfuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

Courtesy of A.S.H.V.E.

430. Regain

The temperature and relative humidity of the air have a marked influence upon the rate of production, the weight, strength, appearance and general quality of such hygroscopic materials as textiles, paper, wood, leather, tobacco and foodstuffs. This influence is due to the fact that the moisture content of substances of vegetable or animal origin, and, to a lesser extent, of certain mineral origin, comes to equilibrium with the moisture of the surrounding air. In the processing of hygroscopic materials it is necessary to secure a final moisture content suitable for shipping, with standards of regain fixed by the trade in question. Where goods are sold by weight it is proper that they contain a standard moisture content.

431. Definitions

The terms *moisture content* and *regain* refer to the quantity of moisture in hygroscopic materials. Moisture content is the more general term and refers to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). Regain is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the dry weight of the material. The moisture content of a hygroscopic material, at any time, depends upon the nature of the material, the temperature, and especially the relative humidity. Table 183 shows the regain or hygroscopic moisture content of substances when in equilibrium at a dry-bulb temperature of 75° F and various relative humidities. All hygroscopic materials, when absorbing moisture from the surrounding air, produce a sensible heat rise in the air equivalent to the latent heat released by the air to the material.

432. Conditioning and Drying

The exposure of materials to desirable conditions for treatment may coincide with the processing of the materials, or it may occur separately in special rooms. The latter treatment is known as drying or conditioning. Its purpose is to establish properties and moisture content. When the final moisture content is lower than the initial the term drying is applied, as outlined in Chapter XVII.

433. Control of the Rate of Chemical Reactions

A typical example occurs in the manufacture of rayon. The pulp sheets are conditioned, cut to size and then mercerized. During this operation close control of both humidity and temperature are maintained. The temperature controls the rate of reaction and the relative humidity maintains a constant rate of evaporation from the surface of the solution.

434. Control of the Rate of Biochemical Reactions

In this field, industrial air conditioning has been applied to many well-known products. All problems involving fermentation come under this heading. As biochemistry is a branch of chemistry, subject to the same laws, the rate of reaction may be controlled by temperature. An example of this is the production of pencillin.

Yeast develops best at a temperature of 80° F and the surface of the dough is held open, allowing carbon dioxide gas to pass through, if it is processed at 65% relative humidity.

The curing of bananas and lemons also belongs in this group.

Tobacco, from the field to the finished cigar or cigarette, offers another example, of air conditioning to control color, texture and flavor.

435. Control of the Rate of Crystallization

The rate of cooling of a saturated solution determines the size of the crystals formed. Both temperature and humidity control are vital, as one controls the rate of cooling while the other, through evaporation, controls changes in the density of the solution.

In the coating of pills, gum, nuts and other confections, a heavy sugar solution is added to the tumbling mass. A smooth opaque coating is obtained only at the proper temperature and humidity.

436. Elimination of Static Electricity

Static electricity is troublesome in the handling of light materials such as textile fibers, paper etc. It is also dangerous, especially where explosives are concerned. In order to eliminate static electricity a relative humidity greater than 50% must be maintained.

TABLE 184

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
	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45

* *Courtesy of A.S.H.V.E.*

TABLE 184 (*continued*)

DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
Confectionery	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
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
Matches	Manufacturing	72 to 74	50
	Storage of matches	60	
Munitions	Fuse Loading	70	55
Paint	Air drying lacquers	70 to 90	25 to 50
	Baking lacquers	180 to 300	
	Air drying of oil paints	60 to 90	25 to 50

TABLE 184 (*continued*)

DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
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

437. Calculations

The routines for determining the heating and cooling loads for various industrial processes are similar to those outlined in Chapters XIII and XIV. The solution of a most familiar prob-

APPARATUS DEW POINT AND INDUSTRIAL AIR CONDITIONING ESTIMATING FORM

JOB NAME ————LOCATION ————ESTIMATOR ————DATE ————
ROOM ————LENGTH ————WIDTH ————HEIGHT ————FLOOR ————NAME ————
Outside Conditions: Dry-Bulb ————Wet-Bulb ————Dew Point ————R.H. ————
Outside Conditions per Lb Dry Air: Water (W) ————Sensible ————Latent ————Total ————V Cu Ft ————
Entering Dehumidifier: Dry-Bulb (t_1) ————Wet-Bulb————Dew Point ————R.H. ————
Entering Dehumidifier per Lb Dry Air: Water (W'_1)——Sensible (h)——Latent ($h_1 - h$)——Total (h_1)——V_1 Cu Ft——
Exit Dehumidifier: Dry-Bulb (t_{ss})——Wet-Bulb (t_{sz})——Dew Point (t_{sz})——R.H.——For Surface Coolers: D.B.——W.B.——
Exit Dehumidifier per Lb Dry Air: Water (W'_{sz})——Sensible (h_{as})——Latent ($h_{sz} - h_{as}$)——Total (h_{sz})——V_{sx} Cu Ft——
Entering Room: Dry-Bulb (t_e)————Wet-Bulb————Dew Point (t_{sz})————R.H.————
Entering Room per Lb Dry Air: Water (W'_{sz})——Sensible (h_e)—Latent ($h_{sz} - h_{as}$)—Total ($h_{sz} - h_{as} + h_e$)—V_e Cu Ft—

Item No.	(Refer to Cooling Load Estimate Sheet)		Heat Load Btu Per Hour Sensible	Latent
9	Transmission Heat Gain			....
17	Solar Heat Gain (Sun Load)			....
20	Duct Heat Gain			....
21	Sensible Body Heat Gain	No. People ———— × 225		....
22	Latent Body Heat Gain	No. People ———— × 175 (Quiet)	....	
23	Latent Body Heat Gain	No. People ———— × 450 (Active)	....	
30	Equipment Heat Gain			
31	Infiltration Heat Gain			
32 (a)	Product and Miscellaneous			
32 (b)	Re-heat (Steam, Gas, Electric, Etc.) By-Passing not to be Included			....
41	Ventilating Air			
33	Total Sensible Heat Gain			....
34	Total Latent Heat Gain			←
35	Total Heat Gain (Cooling Load)			....

FIGURE 94

APPARATUS DEW POINT AND INDUSTRIAL AIR CONDITIONING ESTIMATING FORM

Item No.	
36	Sensible Heat Differential Factor F_d = Item 33 ÷ Item 35 = ________________
44	Total Heat per Lb Dry Air Leaving Room or Entering Dehumidifier h_1 = ________________
45	Dew Point (t_{as}) = [F_d (Item 44 + 11.2) — 0.24 Item 46] ÷ [0.625 F_d — 0.24] = ________________
46	Dry-Bulb (t_l) Leaving Room and Entering Dehumidifier = ________ Degrees
47	Drop in Air Temperature Through Dehumidifier = Item 46 — Item 45 = ________________
48	Volume Air Through Dehumidifier = Item 33 ÷ [1.08 × Item 47] = ________________ CFM
49	Weight of Air Through Dehumidifier = Item 33 ÷ [0.24 × Item 47] = ________ Lb Dry Air per Hour
50	Dry-Bulb Temperature Entering Room (t_e) = ________ Degrees
51	Volume Air Through Room = Item 33 ÷ [1.08 (Item 46 — Item 50)] = ________ CFM
52	Weight Air Through Room = Item 33 ÷ [0.24 (Item 46 Item 50)] = ________ Lb Dry Air per Hr
53	Tons Refrigeration = Item 35 ÷ 12000 = ________ Tons at Suction Temp. t_{as} — 20 = ________ Degrees
56	CFM By-Pass Air = Item 51 — Item 48 = ________ CFM

FIGURE 94 (*continued*)

lem in industrial air-conditioning work is the determination of the apparatus dew point, outlined in Chapter IX. The apparatus dew-point equation chart is given in Figure 12, Chapter IX.

The apparatus dew-point estimating form, Figure 94 in this chapter, offers a systematized suggestion that will prove a source of convenience. Because of the large number of motors and heat producing units prevalent in industrial applications, it is particularly important that allowances be made for these in order to determine the sensible heat and moisture loads. The estimating form attempts to list a few of the items of heat gain. Appliance heat sources are given in Chapter XIV as are also heat losses from bare and insulated pipe.

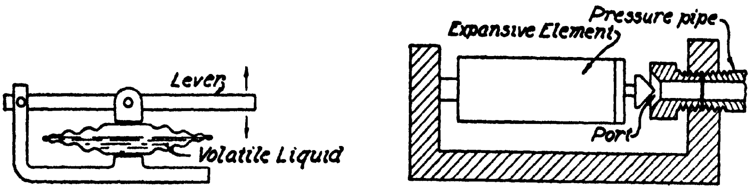
Chapter XXIII

AUTOMATIC CONTROLS

In the operation of automatic heating and air conditioning systems automatic controls play a most important part. It must be remembered that air conditioning by its very definition requires simultaneous control of all the essential functions of the system.

In classifying controls the two general classifications may be said to consist of *safety controls* and *operating controls,* the safety controls being installed to insure safety from dangerous conditions while the operating controls are used to obtain certain features of operation. These two general classes may be divided into *limit controls* and *ordinary controls.* The limit control is one which functions only when conditions approach or reach the limit, beyond which they should not go, while an ordinary control operates as closely to the condition for which it is set as is possible, with such types of mechanism.

An example of a safety control is embodied in the pressure control used on a steam boiler where the oil or gas burner is closed off when the boiler reaches the maximum safe operating pressure. This is also a limit control, as it operates only when such a maximum pressure is reached, and at other times does not control the boiler in any way. A safety control which is not of the limit type is that, embodied in the "Stackstat," which shuts off the oil burner if the stack gases do not rise in temperature within a sufficient interval after ignition has presumably taken place. An operating control of the ordinary type is embodied in the room thermostat which starts and stops the burner in accordance with the room temperature and an operating control of the limit type is embodied in the aquastat used on a steam boiler to prevent the boiler water temperature from going below 160° F when the boiler is doing summer hot water heating for the domestic water supply system.

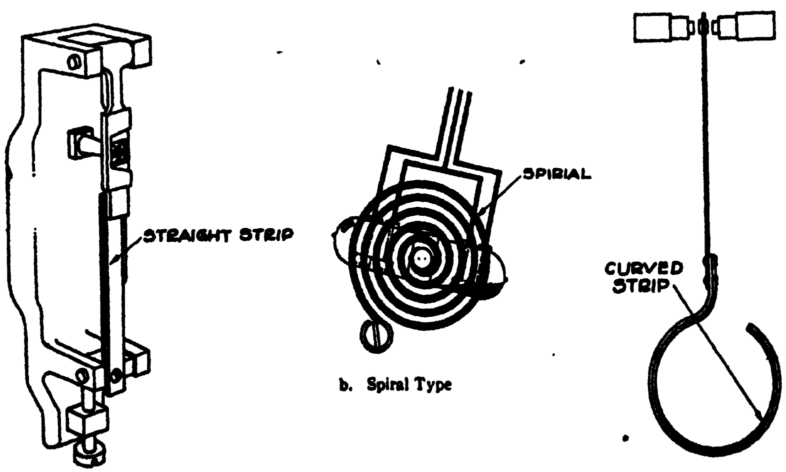


DIAPHRAGM TYPE THERMOSTAT

DIRECT EXPANSION TYPE THERMOSTAT

FIGURE 95

Diagrammatic Representation of Thermostats



a. Straight Strip Type

b. Spiral Type

c. Curved Strip Type

FIGURE 95

Three Types of Bi-Metallic Thermostats

438. Air Conditioning Controls

Air conditioning system controls run into considerable complexity, especially when refrigeration is used in combination with the conditioning system. The controls vary with the type of system, whether 100% outside air is provided for, whether the system is for summer use, winter use or year-round use, and with the type of equipment used. One of the most important controls is that which operates the cooling coils or the temperature of the air washer when one is used, as this is the control which governs the amount of moisture in the air going to the room. Another very important control is the room thermostat which controls the temperature or amount of air delivered in accordance with the room temperature as this thermostat controls the temperature obtained in the room. Still another is the thermostat in the supply air duct which prevents air at a lower temperature than 65° F from being delivered at any time.

Besides these, are the thermostats in the outside air intakes which turn on 100% outside air when the temperature of the outside air is sufficiently low to permit cooling without the use of refrigeration and other thermostats in the outside air intake to turn steam on the first row of heaters whenever the outside temperature reaches 35° F and the danger of freezing the coils becomes imminent. Dampers in the by-pass must be controlled, dampers under or around the coils, if any, must be controlled, and various other features made automatic.

On the refrigeration system the machines are usually started and stopped by a thermostat on the water cooler; limit pressures on the high pressure side are used, together with various other controls on the motors for their protection and safety, such as low voltage, the fuse in one phase of the current blowing, current failure, and so on.

Fans often are provided with remote controls, circulation pumps with controls such that the system cannot be started until these pumps are first put into service, and, sometimes, so arranged that the stopping of the system automatically will shut down the operation of auxiliary features such as the refrigeration, circulation pumps, cooling water and so forth.

As a usual thing, each air conditioning system is worked out by itself as far as controls are concerned, because, while there is a great similarity between air conditioning systems in general, the operating features often vary considerably. The type of operator has something to do with how complex an automatic system can be used; the size of the system also has a bearing on the controls, running toward simplicity on the small systems and more complication for the larger ones.

439. Unit Air Conditioner Controls

The controls on unit air conditioners vary accordingly if the unit is a summer cooling unit, a winter unit or a year-round unit. As a general thing, most units are supplied by the manufacturer with any controls necessary or desired; while the most frequent type of control is a thermostat regulating the temperature of the room, in humidifying units a humidistat may also be installed to control the humidity in the room. Self-contained units generally have all the controls required for the successful operation of the refrigeration portion and the other controls may be added as outlined above.

440. Combination Heating and Cooling Units

The cabinet-type room cooler is constructed in various combinations involving both heating and cooling equipment, and, in addition, ventilating and humidifying equipment. Some combination units arranged for cooling also include built-in refrigeration compressors. While the application of controls to such combination units depends upon the various functions which are to be performed by the unit, the general considerations from the control standpoint have already been covered in the preceding topics describing the unit heater, the unit cooler and the unit ventilator, and while such combination units often permit desirable coordination or interlocking of the control functions, the wide number of possible combinations prohibits a detailed discussion.

441. Principles of Control Systems

Controls may be broadly classified by considering them in relation to the primary operating media which they employ. Such a classification indicates that there are five major groups into which they may fall:

1. Electric Control Systems.
2. Pneumatic Control Systems.
3. Combination Systems.
4. Self-Contained Systems.
5. Hydraulic Control Systems.

A brief description of each of these systems is listed below.

Electric Control Systems

In such control systems the primary medium utilized to provide operation is electricity. The basic function of these controls consists of switching or otherwise adjusting electric circuits to govern electric motors, relays or solenoids. The individual units of this type of system are inter-connected by line voltage or low voltage wiring.

Pneumatic Control Systems

In the pneumatic control systems, compressed air is used as the primary source of operation. The air pressure is varied by the controlling devices. In these systems one or more centrally located air compressors furnish a supply of compressed air which is distributed in special piping to the various controlling and controlled devices. By means of leak ports or orifices, the pressure of the air is varied in the branch lines and the changing pressures are utilized to provide the movement necessary for the operation of valves and dampers.

Air Conditioning and Refrigeration Controls

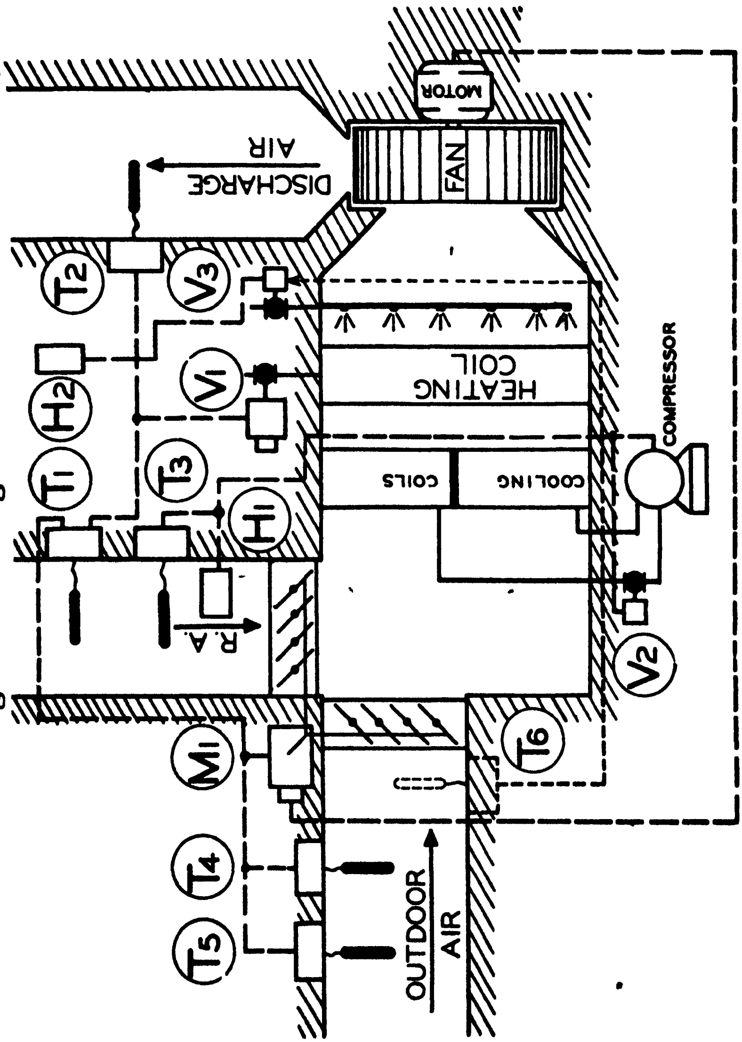


FIGURE 97

Figure 97 illustrates a year-round central fan air conditioning system using direct expansion refrigeration for cooling and operating as a warm air system in the winter.

OPERATION

It provides for control of temperature and humidity on the heating cycle and for a variation of the sensible and latent heat ratio when cooling.

HEATING CYCLE

T1—Thermostat positions as required
V1—Modulating Motorized Valve
T2—Thermostat positions V1 on discharge air
M1—Damper motor shifts from T1 to
T4—Thermostat to take in outdoor air
H2—Humidity controller operates
V3—Solenoid valve

COOLING CYCLE

T5—Thermostat positions outdoor air
M1—Damper Motor
T3—Thermostat starts compressor—both coils
H1—Humidity control at high humidity closes
V2—Solenoid valve, operating compressor on one coil, decreasing sensible latent ratio
T3—Opens both coils at higher temperature

Combined Systems

Experience has indicated that both electric and pneumatic control systems have inherent advantages which indicate their respective use for specific types of applications. It is only natural, therefore, that the two media should be combined to provide a system of control sharing the advantages of each. Combined systems are usually best adapted to large air conditioning and public building installations.

Self-Contained Systems

Self-contained control systems have, in general, been restricted to such operations as could be effectively handled by a power unit with an integrally mounted or directly connected controller. Such applications consist of valves utilized to admit steam or other media into coils to regulate the temperature of tanks or to regulate the admission of steam into heating coils as determined by the controller element. Such systems do not ordinarily have wide separation of the controlling mechanism and the power unit.

Hydraulic Systems

Liquid under pressure provides another easily controllable source of power and is occasionally used as a control medium.

442. Control Operation

Control systems may be further classified into three general types as regards the characteristics of the motion imparted by the controls to the controlled equipment. These three classifications will be described in the following paragraphs. It must be remembered that very often heating, ventilating or air conditioning systems under automatic control may make use of more than one and sometimes all of the three types of control in various phases or functions of the system. Following this broad discussion will be found pages outlining the actual basic circuits used in obtaining these actions.

Two-Position Control

Two-position control is also referred to as *on and off* control or as *positive-acting* control. As an example, a simple thermostat which starts and stops an oil burner or a ventilating fan, or which opens and closes a solenoid valve, can merely select between starting and stopping of the fan or between opening and closing of the valve. There are no intermediate positions nor degrees of motion between the two extremes of operation. Similarly, a two-position motorized damper would move between two fixed limits such as full-closed and full-open.

Two-position control is, in general, the simplest method of providing regulation; however it has definite shortcomings in certain circumstances.

Two-position systems are most generally used where the controlled device is one of the following:

1. Relay
2. Solenoid Valve
3. Exhaust Damper
4. Small Volume Damper

443. Floating Control

Floating control is a designation applied to a control system in which a control valve or damper motor will operate when the controller makes contact at either extreme of its differential, coming to rest only when the controlled medium has stabilized between these limits.

Floating control may be used to advantage on applications where the movement of a motor valve or damper will produce a change in the controlled medium which will be reflected immediately at the controller unit.

Applications of this type are:

1. Static pressure regulation
2. Tank level control

444. Modulating Control

The modulating control system is also designated as *gradual or graduated acting control* or *proportioning control.* These names are synonymous, as applied to automatic control, and are used to designate the type of system in which a control valve or damper motor modulates or proportions the flow of air, steam or water in response to a change of conditions at the controller. Modulating control causes motion in the controlled device in proportion to the motion produced in the controller by fractional degree variations in the medium to which the controller is responsive. After this fractional change has been measured at the controller and translated into terms of a new position of the valve or damper, the modulating system stands by, awaiting further change at the controller. The extent of the motion is limited only by the limits of the controller and by the intensity of the change of condition as measured, hence the valve or damper is repositioned as frequently as changes at the controller occur, but always in direct proportion to the degree of change.

Modulating control systems may be used on all installations where it is desirable to establish and maintain a large number of intermediate positions or stages between the two extremes of operation.

TABLE 185
FREON-12, CONDENSER PRESSURE 136 LB GAGE
(110° F Saturation Temperature)

Btu Per Hour Per Horsepower	Suction Temperature ° F	Suction Gage Pressure, lb
12000	44	40
10000	32	30
8000	26	25
6000	11	15
4000	—13	3
3000	—21	0

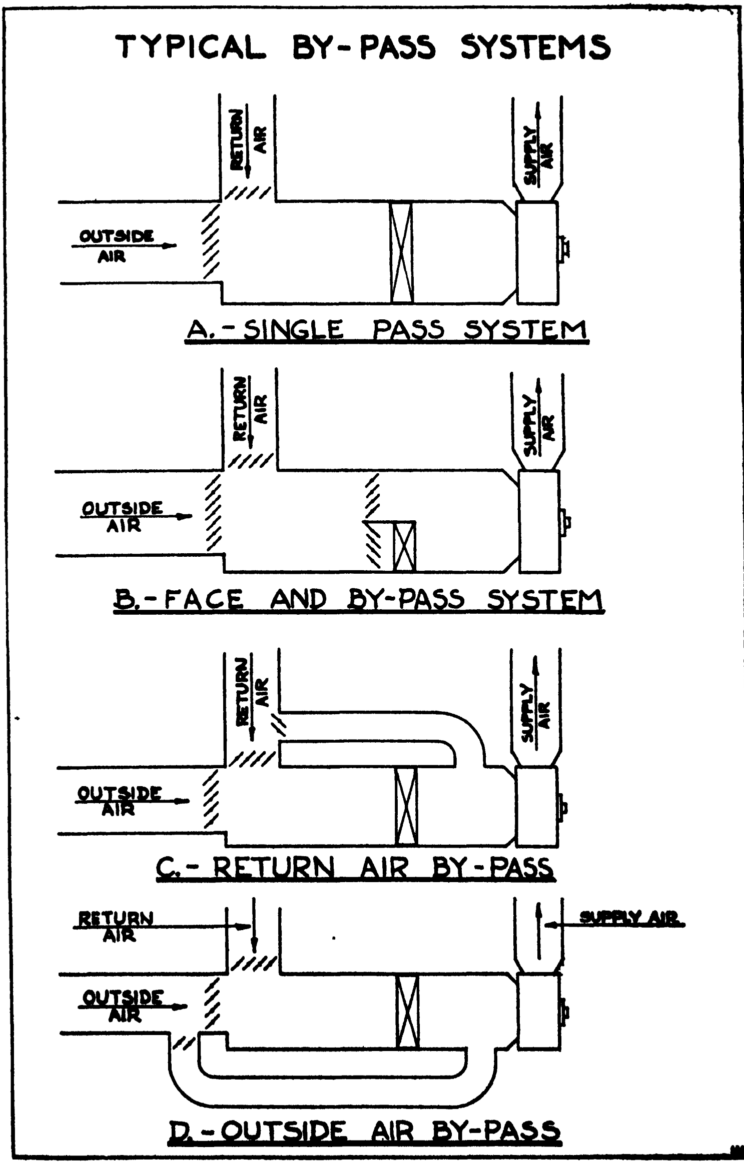

TYPICAL BY-PASS SYSTEMS
RETURN AIR
SUPPLY AIR
OUTSIDE AIR
A.-SINGLE PASS SYSTEM
RETURN AIR
SUPPLY AIR
OUTSIDE AIR
B.-FACE AND BY-PASS SYSTEM
RETURN AIR
SUPPLY AIR
OUTSIDE AIR
C.-RETURN AIR BY-PASS
RETURN AIR
SUPPLY AIR
OUTSIDE AIR
D.-OUTSIDE AIR BY-PASS

FIGURE 98

445. Control of Refrigeration Load

From Table 185 it will be seen that definite control over coil temperatures should be maintained to avoid waste of power, and at the same time insure sufficient dehumidification when necessary.

While Table 185 refers to Freon-12 similar comparisons may be made for other refrigerants and compressor capacities.

The basic characteristics of a compression refrigerating system permit control over coil temperatures for the reasons listed below:

1. The coil temperature bears a direct relation to the suction pressure.
2. The suction pressure is determined by the relation of the load on the low side (the coil) as balanced against the capacity of the high side (refrigerating machine).
3. A decrease in the volume of air flowing over the coil decreases the load on the coil and therefore causes the suction pressure to drop, giving a lower coil temperature. Conversely, an increase in the air volume over the coil gives a higher coil temperature.
4. A decrease in the total amount of coil area on the system causes a decrease in load and a drop in suction pressure resulting in lower coil temperature. An increase in the total amount of coil surface results in a higher coil temperature. This principle is utilized in the Modutron System.
5. A decrease in the capacity of the refrigerating machine results in an increase in suction pressure and a higher coil temperature. An increase in machine capacity results in a lower coil temperature.

446. Control of Central Fan Cooling Systems

The summer air conditioning system must control the temperature and moisture content of the air as well as provide ventilation, circulation, and cleaning.

The first types of systems to be considered are those wherein moisture removal, or dehumidification, is accomplished incidental to the cooling of air.

In most air conditioning systems air is cooled to a temperature below the dew point of air in the space being conditioned and under these circumstances the air conditioning plant is said to have a capacity for absorbing latent heat or moisture. The extent of this capacity for dehumidification will depend upon the type of refrigerant utilized and the cooling surface over which the air passes.

From the standpoint of heat exchange surface used, central fan cooling systems fall generally into these classifications:

1. Systems providing cooling through the use of finned coils; cold water or brine being used as a refrigerant.
2. Direct expansion systems. Cooling is provided by the expanding of liquid refrigerant into a finned coil, cooling being accomplished because of evaporation of the refrigerant in the coil.
3. Washer systems. In this type of system there usually is no cooling surface per se, but the air is cooled in passing through a chamber in which cold water is sprayed in a finely atomized condition. In this type of system the air is brought down to a nearly saturated condition at a dew point determined by the water temperature used.

A further classification of cooling systems may be made with reference to the arrangement of the plenum chamber. Most of the systems illustrated in this section are shown with the fan drawing air through a conditioning chamber and discharging it to the duct system. Actually this arrangement is frequently reversed so that the fan will force the mixture of return and outside air through the cooling surface and thence to the duct system.

The plenum chamber or conditioning unit generally takes one of the following forms:

Single Pass System

In such a system all of the conditioned air will pass through the cooling coil or washer. (Figure 98, A)

Face and By-Pass System

On systems of this type dampers are usually placed across the surface of the cooling coil or washer chamber, and a by-pass duct with damper permits part of the air to pass completely around the cooling coil or washer. These dampers are generally operated together so that as one closes the other opens. On this type of system all of the air does not pass through the coil since some of it is by-passed and the air that is by-passed may be a mixture of outside and return air. (Figure 98, B)

Return Air By-Pass

This type of system is similar to the one just described. Instead of the by-pass handling a mixture of outside and return air, the by-passed air comes directly from the return air duct and dampers are installed at that point. The air that is by-passed and mixed with conditioned air is return air from the rooms being served. (Figure 98, C)

Outside Air By-Pass

This type of arrangement is occasionally used and again is similar to those just described with the exception that the air by-passed comes from the outside air duct. (Figure 98, D)

The design of both the cooling surface and the plenum or conditioner chamber will depend largely upon the conditions of load peculiar to individual installations. Controls may be applied to any combination of cooling surface and plenum design.

Some of the most common ways of effecting changes in suction pressure and thereby regulating the dehumidifying capacity of a direct expansion system are:

Variable Coil Area Control

Perhaps the most satisfactory means of capacity reduction is obtained with the *Modutron* control system, manufactured by the Minneapolis-Honeywell Co. The Modutron provides for the variation of liquid flow to the direct expansion coil and in effect changes the superheat at the outlet of the coil in response to changes in room conditions. This means that the effective coil area is cut down as the cooling requirements drop off, resulting in a desirable reduction in coil temperature under light load which in turn produces better relative humidity conditions.

Two-Speed Compressors

Operating suction pressure can be changed by switching a compressor from one speed to the other.

Multiple Compressor Installations

Again the capacity, and hence the suction pressure, can be changed by controlling the number of units operating at a given time.

Compressor By-Pass Valves

Frequently a refrigerating machine may be equipped with by-pass valve or clearance pocket control that will provide a means of automatically reducing the capacity of the machine.

Variable Speed Compressors

Compressors with an electric drive of the variable speed type or compressors driven by gas engines generally provide a means of changing the speed, and hence the capacity of the machine, with modulating action.

Double Coil or Multiple Coil Systems

In such installations the direct expansion coil which is located in the duct is made in two sections independently controlled so that either one section or both can be made effective. It is obvious that if we operate on only one-half of the coil the load on the machine in terms of Btu/min will be materially less than the load where the entire coil surface is used. This means that lower suction pressures will prevail when operating on half coil area.

Variable Air Volume

An installation which, through the adjustment of dampers, provides a means of changing the volume of air passing over the coil, does to some extent change the load on the coil and therefore the coil temperature. This principle is involved in by-pass systems.

Figure 99, A, illustrates a simple type of direct expansion cooling installation utilizing a single pass system with a single coil section. The flow of refrigerant to the coil is controlled by a solenoid valve. The solenoid is opened and closed upon demand of a thermostat.

With this type of system the suction pressure is allowed to assume any value determined by the load at a given instant. It can be seen that coil temperatures will automatically assume a higher value as load conditions rise. The only variation in instantaneous load on this type of system is that which comes from a reduction in the entering wet-bulb value.

Although the control sequence as illustrated shows the solenoid valve alone under thermostatic control, it is common practice to de-energize the compressor circuit simultaneously. If this is not provided for, the compressor is operated by a combination high pressure cutout and suction pressure controller. The low pressure control is generally set to cut out at a point which will prevent frost from forming on the coil, and to cut in at a much higher point.

Figure 99, B, illustrates a face and by-pass damper installation wherein the temperatures are maintained by positioning dampers to regulate the amount of air flow through the coil. This system provides for gradually changing the temperature of the air discharged and is therefore a considerable improvement over the simple system shown previously. Discharge-air temperatures will remain constant at the proper point for meeting the load requirements whereas the on-off system results in alternate discharge of first cold and then warm air.

It is also true that under light sensible load conditions, the load on the coil will be reduced, causing a reduction in coil temperature, and therefore the system should permit lower relative humidities than the simple single coil system. The minimum coil temperature will be limited by the point at which frost will form on the coil and therefore it is necessary to cut off the coil on the refrigerating machine as the face damper approaches the closed position. This is frequently done by the use of an auxiliary switch associated with the motor which will act to close a solenoid valve. A suction pressure control on the refrigerating machine will cut off the compressor when the valve closes. This control is set so as to stop the machine whenever the suction pressure reaches a point which would be indicative of a frosting coil. It should be noted that the reduction in air volume on the coil also tends to limit the point at which frost will form on the coil.

With this arrangement of equipment, substantially constant temperatures are maintained in the discharge from the system and the noticeable ups and downs of on-off control are eliminated. It is also true that this system provides for more latent cooling under light sensible load conditions by reason of the slight reduction in coil temperature that takes place due to the restricted air volume.

The Modutron (Minneapolis-Honeywell)

Figure 99, C, illustrates what is known as the Modutron system for direct expansion refrigerant control. The Modutron

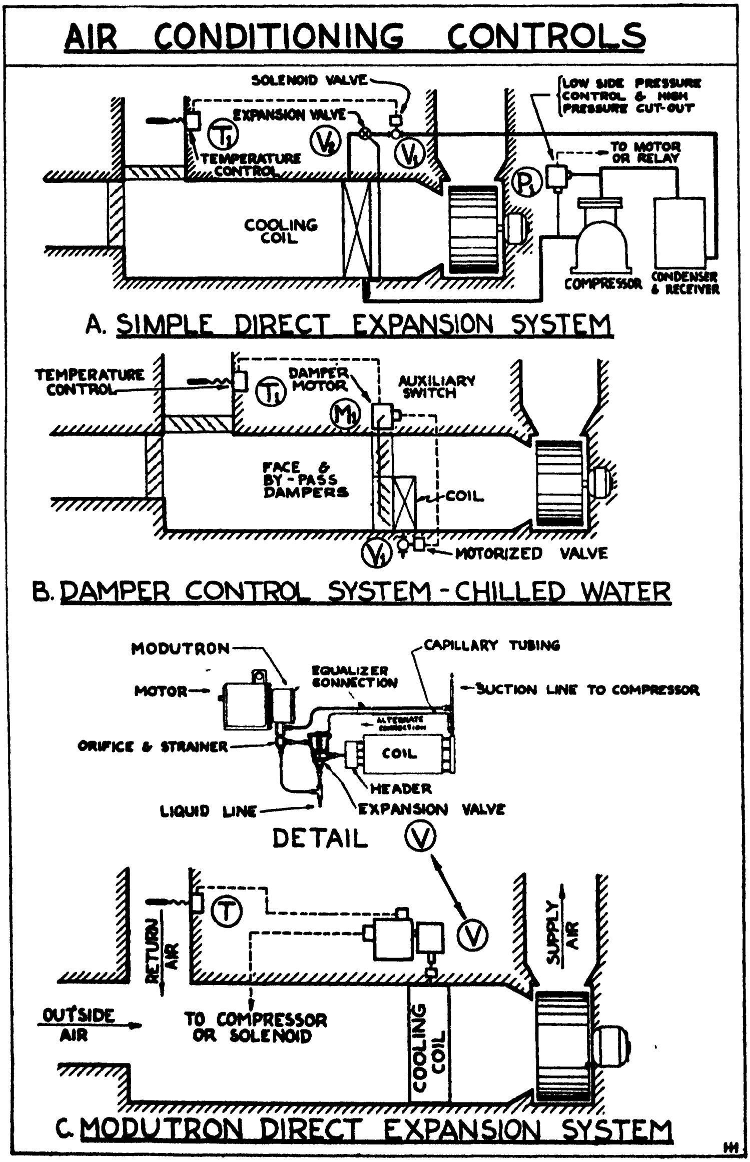

AIR CONDITIONING CONTROLS
SOLENOID VALVE
EXPANSION VALVE
LOW SIDE PRESSURE CONTROL & HIGH PRESSURE CUT-OUT
TO MOTOR OR RELAY
TEMPERATURE CONTROL
COOLING COIL
COMPRESSOR
CONDENSER & RECEIVER
A. SIMPLE DIRECT EXPANSION SYSTEM
TEMPERATURE CONTROL
DAMPER MOTOR
AUXILIARY SWITCH
FACE & BY-PASS DAMPERS
COIL
MOTORIZED VALVE
B. DAMPER CONTROL SYSTEM - CHILLED WATER
MODUTRON
CAPILLARY TUBING
EQUALIZER CONNECTION
MOTOR
SUCTION LINE TO COMPRESSOR
ORIFICE & STRAINER
COIL
HEADER
LIQUID LINE
EXPANSION VALVE
DETAIL
RETURN AIR
SUPPLY AIR
OUTSIDE AIR
TO COMPRESSOR OR SOLENOID
COOLING COIL
C. MODUTRON DIRECT EXPANSION SYSTEM

FIGURE 99

system may be applied to any refrigeration system employing expansion valves with the usual external equalizer. A small control valve is installed in the equalizer line of the thermostatic expansion valve. Figure 99, C, shows the details of the connections made to a typical coil installation.

The motor which operates the valve is modulated by temperature, humidity, pressure, or any other condition to be controlled and opens the valve wide when maximum cooling is required. Under these conditions, the refrigeration system operates in a perfectly normal manner at maximum capacity as determined by the expansion valve.

If less refrigerating effect is required, the motor moves the valve toward the closed position, reducing the effect of the external equalizer on the expansion valve. Since the inlet pressure to the coil will be higher than the outlet pressure because of pressure drop through the distributor and coil, the expansion valve is then affected by a higher pressure than when the Modutron is wide open. As a result the expansion valve operates at a higher super-heat.

When the Modutron is completely closed, the expansion valve is affected only by the pressure of the refrigerant leaving the valve, and on most systems, the capacity will then be reduced to about 60 or 70% of full load capacity.

Because the cooling requirements will usually be more than 60 to 75% of full capacity, a further reduction is often not necessary. On single coil jobs the compressor can be controlled at lighter loads by a switch which operates in sequence with the Modutron valve. (Figure 99, C)

On jobs requiring further reduction of capacity, an orifice assembly may be added. This orifice bleeds, from the liquid line, a small constant quantity of liquid into the pressure chamber of the expansion valve. When the Modutron is wide open, the quantity of liquid has no effect on the expansion valve. When the Modutron is tightly closed, the liquid is sufficient to build up a closing pressure in the expansion valve and prac-

AUTOMATIC CONTROL OF CENTRAL SYSTEMS

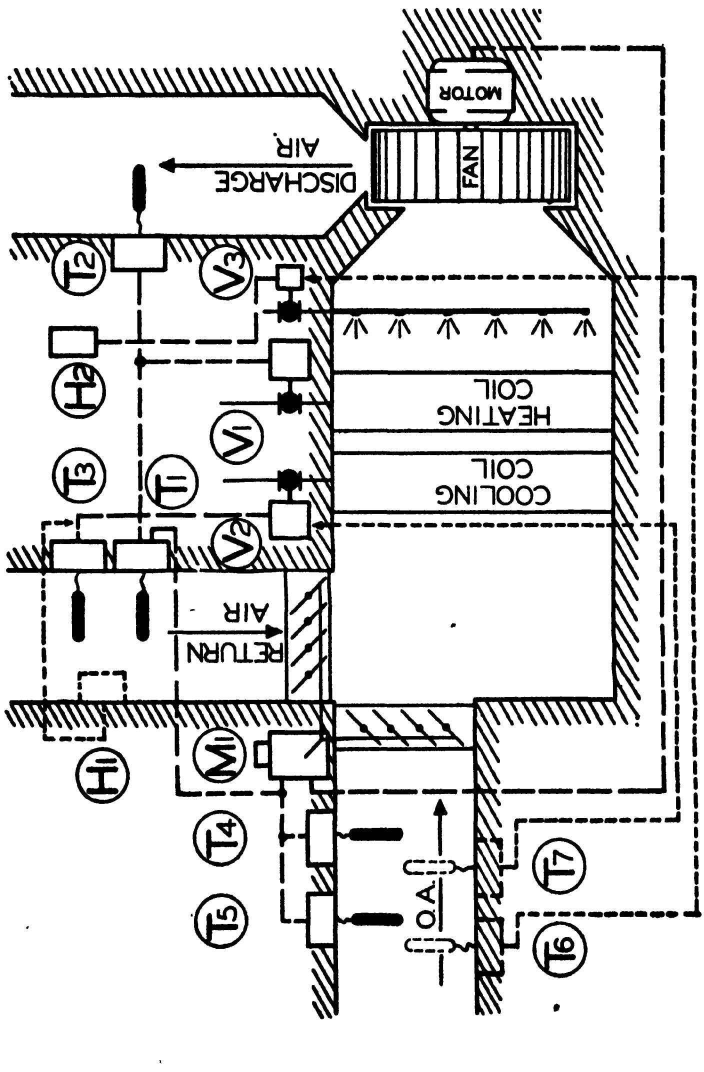


FIGURE 100

Figure 100 Illustrates a year round central fan air conditioning system. This system provides temperature control on a cold water coil, and temperature and humidity control on heating. Heating—cooling change-over is automatic.

HEATING CYCLE

T1—Thermostat control return air temperature by
V1—Modulating motorized valve positioning
T2—Thermostat positions V1 on discharge air
M1—Damper motor shifts from T1 to
T4—Thermostat to take in outdoor air as per T2
H2—Humidity controller operates
V3—Solenoid valve to add moisture

COOLING CYCLE

T3—Thermostat controls return air temperature by
V2—Motorized valve (2 pos. or mod.) positioning
M1—Outdoor air damper motor controlled by
T5—Thermostat on mild days
Control panel devices adjust minimum outdoor air damper setting, closing damper and valve V2 on fan shut down.

ALTERNATES

H1—Modulating humidity controller
T7—Thermostat with compensating control,

tically close it. Thus, the addition of the orifice to the Modutron gives modulation from full capacity down to between zero and 20%, depending upon the characteristics of the expansion valve. It should be noted that capacity reduction is secured without changing the air velocity over the coil, and that it is possible to modulate to a lower suction pressure without danger of frosting the evaporator, even though only one coil may be used. It is possible for one Modutron to control as many as three evaporators by connecting together the equalizer openings and allowing the Modutron to control all three simultaneously.

447. All Year Systems

All year systems combine the features described for heating and cooling cycles, and have provisions for spring and fall conditions. Complete automatic control of all year systems incorporates an automatic change-over between the cooling and heating cycles. If the installation necessitates operation of a manual switch or other change-over device between the heating and cooling cycles, then the control system is semi-automatic. The full automatic change-over between cycles becomes particularly desirable in the early and late portions of the cooling and heating seasons, when heating and cooling may be required alternately.

In applying controls, attention to a few basic principles is important for best results. A control instrument can react only to the air immediately surrounding it. Where loads are dissimilar in various parts of the building, separate zones should be set up. This may be caused either by unequal exposure or unequal internal loads. For example, see Figure 101. In this L-shaped building, zones 1 and 2 are never exposed to the sun, while zones 3 and 4 are exposed at different periods of the day. However, on the top floors of the building, zone 1 may have sun for a period early in the morning. This would call for at least three zones in the lower floors and four zones in the upper floors of the building. Corner rooms such as 5 may often be served partly from each adjoining zone.

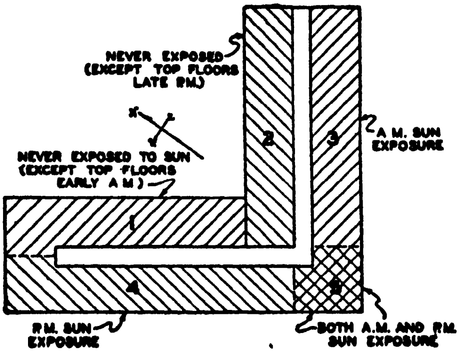


FIGURE 101

Zoning of Buildings

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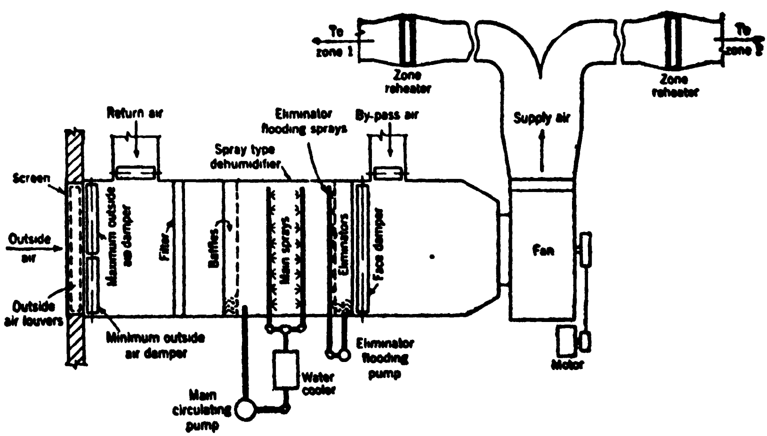

FIGURE 102

Central System with Zoning by Reheating

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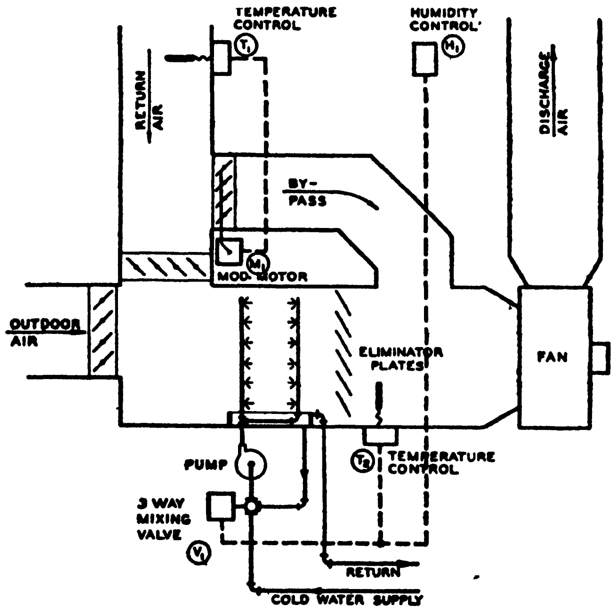

FIGURE 103

Application of Dewpoint Control

It is apparent that additional economies can be obtained if the dewpoint temperature is raised during periods when the latent heat load is low. This function can be accomplished by using a humidity control to reset the control point of the dewpoint thermostat in such a manner that

1. Low dewpoint temperature will be maintained when relative humidities are high.
2. A higher dewpoint temperature will be maintained when relative humidities are low.

This additional feature prevents the unnecessary removal of latent heat.

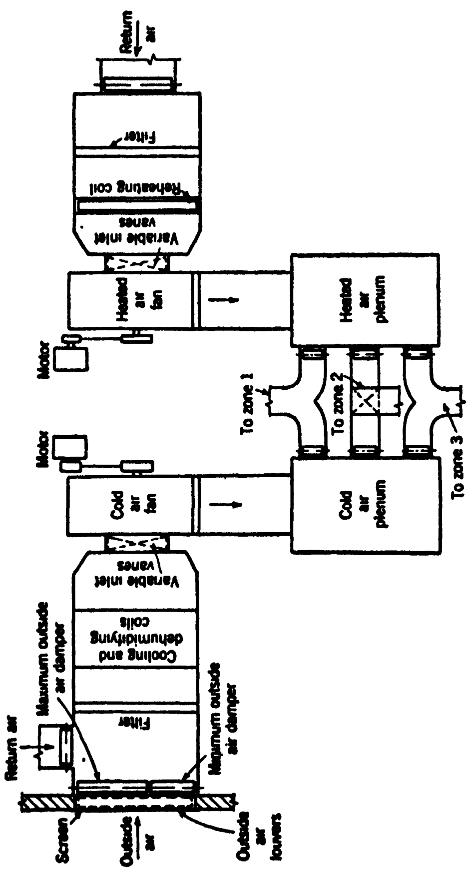

FIGURE 104

Central System Using Dual Duct Method of Zoning

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CHAPTER XXIV

ESTIMATING COSTS

Estimating costs is one of the important phases of an air conditioning project, as important as estimating engineering capacities.

Too much emphasis cannot be placed upon the necessity of a survey, to obtain a complete knowledge of all existing conditions and requirements. Without careful planning, unnecessary work and changes will be required after the work is done, and more often will result in unsatisfactory operation. These things all add to the cost of the installation and reflect unfavorably on the reputations of all concerned.

448. Engineering Survey and Report

The report forms the basis of the whole transaction and should therefore be full and complete. The calculation of accurate heating and cooling load conditions hinges on obtaining the necessary data on the project itself. Facts derived from a survey form the basis of the entire subsequent estimate (both load and cost); therefore, without an accurate survey there can be no accurate engineering or cost prediction. A thorough study of load conditions, then eliminates the need for a *factor of safety.* The following outline is a guide to the items necessary for a complete survey of an air conditioning system. Sketches must be made of all spaces to be conditioned, showing wall, floor, partition, ceiling construction and dimensions. In addition the available information should be tabulated on a standardized form which minimizes the possibility of omission of items. From the data thus accumulated the two important air conditioning calculations are made:

1. Heat Gain Estimate for Cooling and Dehumidifying in Summer.

2. Heat Loss Estimate for Heating and Humidifying in Winter.

ENGINEERING SURVEY AND REPORT

Date ______________________

1. Name of Firm ______________________
   Address ______________________
   Manager ______________________ Supt. ______________________
   Plant Engineer ______________________
2. What temperature and humidity are desired? ______________________
   (a) Summer ______________________
   (b) Winter ______________________
   (c) Shall estimate include humidifying and heating to room temperature in winter? ______________________
   (d) Shall estimate include cooling and humidifying or dehumidifying in summer? ______________________
   (e) State Purpose of Installation (Comfort or Industrial) ______________________
3. Number of Employees in room ______________________
4. Description of goods brought into room ______________________
   (a) Quantity ______________________
   (b) Temperature ______________________
5. Are there awnings, blinds or shades on South side? ____ West? ____ East? Will awnings, shades or translucent fabric on all skylights of the conditioned room be installed? ______________________
6. What processes are carried on in room to be conditioned? ______________________
   (a) If process is chemical, give heat of formation of batch or heat of fermentation ______________________
   (b) Weight of wet batch ______________________
   (c) Weight of water contained ______________________
   (d) Weight of finished batch ______________________
   (e) Number of batches per working day ______________________
   (f) Number of working hours per day ______________________
7. A. What processes are carried on in adjoining rooms? ______________________
   B. What temperatures are maintained in floors below and above? ______________________
8. A. What processes are carried on floors below and above? ______________________
   B. What temperatures are maintained in floors below and above? ______________________
9. What system of heating in room? ______________________
   (a) Give heating surface __________ Give pipe covering surface __________
   (b) Give temperature and pressure, if steam ______________________
   (c) Give volume of air per minute if indirect ______________________
   (d) Give type of return system-vacuum or gravity ______________________

Engineering Survey and Report (*continued*)

10. Will the conditioning apparatus displace the present heating system or will it act as a booster?__________
11. Electricity in room__________
    (a) Processing: __________Kw per hour? __________
    (b) Batteries: __________Kw hour rating? No & size__________
    (c) Heaters: __________Give current consumption per hr_____
    (d) Irons: __________Give current consumption per hr_____
    (e) Lights: __________Give current consumption per hr_____
    (f) Motors: __________Give number and hp__________
    (g) Generators: __________Give number and kw capacity ______
    (h) Motor Generators: _________Give number and kw capacity________
    (i) Miscellaneous: __________
12. Electric current characteristics __________
    (a) Alternating or direct current __________
    (b) Volts __________
    (c) Phase __________
    (d) Cycle __________
    (e) Will estimate include power wiring from main switchboard already installed? __________
    (f) Will estimate include additional electric lighting?__________
13. Is floor equipped with sprinkler system? __________
14. Are there any engines, pumps, presses or line shafting in room?________
15. Give list of all tanks and vats in room__________
    (a) Contents in gallons __________
    (b) Temperature of water __________
    (c) Surface area in sq ft __________
    (d) Heated by coils or by steam injector __________
    (e) Give heat supply and pressure__________
    (f) Is vat refilled?__________
    (g) Is same insulated?__________
16. Give list of all kettles and pots in room__________
    (a) Jacketed or open__________
    (b) Contents in gallons__________
    (c) Working temperature__________
    (d) Water surface area, in sq ft__________
    (e) Give coil surface, size of injector or jacket area__________
    (f) Are same insulated?__________
    (g) Are same hooded?__________
17. Are there any steam mains in the room?__________
    (a) Size and length, insulated or not__________
    (b) Any traps?__________Number and size__________
    (c) Are they in use in winter or summer?__________
    (d) Insulated or not__________
18. Any hot water piping in room?__________
    (a) Size and length__________
    (b) Insulated or not__________
    (c) Are they in use winter or summer?__________

ENGINEERING SURVEY AND REPORT (*continued*)

19. Is there any steaming or hot water spraying apparatus? Give all data__
____
20. Are there any hot plates or steam tables?____________________________
    (a) Give number and size in sq ft_____________________________
    (b) Lineal feet of coils_____________________________________
    (c) Size of pipe____
    (d) Temperature carried_____________________________________
21. Are there any gas-fired boilers in room?____________________________
    (a) hp ____
    (b) Insulated? ____
    (c) Smoke pipe connected to outdoors?________________________
22. Are there any gas burners, stoves or any other gas burning apparatus?___
    Give number and size in cubic feet per hour_______________________
23. What kind of gas is used?____
    (Burning gas gives off considerable water vapor when burned)
24. Is there well-water available?_____City water?_____Cooling tower?_____
    (a) Quantity ____
    (b) Temperature ____
25. If brine system, give quantity and temperature________________________
    If direct expansion, give suction and condenser pressure_______________
26. Will purchaser bring refrigerating lines and returns to apparatus?______
    Will he insulate them?____
27. A. Is ample space provided for apparatus? Build separate room if necessary? Insulate?____
    B. Will estimate include all foundations?__________________________
    C. Will estimate include cutting and patching where ducts pass through the walls, ceilings, floors and partitions?_________________________
    D. Will estimate include cartage____________________________________
    E. Will estimate include the common labor?__________________________
28. If air supply is taken from a window
    (a) Note whether from a court_______________________________
    (b) Note whether exhaust or vent from floor below can interfere______
29. What will be the hand of the unit? (Determined as follows: Facing front of unit if fan is to the left, it is a left hand machine and vice versa.)
____
30. Clear height or working space at proposed Air Conditioner Location____
____
31. Drain Pan Type desired:
    (a) Floor type with drain_____________________________________
    (b) Elevated type____
    (c) Floor type without drain___________________________________
32. Kind of floor: Kind of Roof:
    (a) Concrete ___________________(a) Concrete _________________
    (b) Wood ______________________(b) Barrett Specifications___________
    (c) Tile ________________________(c) Wood with tar and gravel_______

ENGINEERING SURVEY AND REPORT (*continued*)

(d) Special ____________________ (d) Slate ____________________________
(e) Special __________________________

33. Make sketch showing all dimensions of rooms and partitions; dimensions, thickness and directional exposures of doors and windows. If windows single glass, will estimate include changing to double glass? Give dimensions of skylights and note if estimate will include awnings.
34. Tabulate all data, if estimate will include insulation of walls, floors, ceilings, partitions or equipment.
35. Where holes or openings are to be cut in supporting walls, partitions, or roof, give sketch and record of local building code ordinances. Will estimate include this work?
36. Make sketches for location of refrigeration machinery if required.
37. Source of condenser water supply____________________________________
Maximum summer temperature____
Water pressure________________Size water supply line_________________
Gallons per hour available_______________________.

449. Predetermined Cost Data

In compiling estimating and cost data that may be used in all parts of the world, it is not feasible to quote material prices and labor costs that will apply in every locality, without the use of some judgment and experience on the part of the estimator. Prediction of labor costs are only of value if determined by past records of similar units on a basis of straight time man hours.

After the capacity of the air conditioning system is estimated and tentative specifications drawn up, material is taken off and priced according to the existing market. Local wage scales and labor skill are to be considered in order to obtain a dependable estimate.

Regardless of the many conflicting opinions, the author hesitates to state that labor at present is less efficient than it was in the "good old days"; production records have been made by labor under proper supervision and planned production during the war. Estimating will never be an exact science. Therefore if the job after completion proves unprofitable, it will be just as difficult to estimate whether labor, supervision or the estimator is to blame.

450. Detailed Estimates and Unit Costs

There is no "short cut" accurate enough for competitive bidding at a profit. Short cuts are necessary for submitting preliminary estimates for budgets; otherwise the management would have no sound basis for the appropriation of funds and the project would not go ahead. Accurate measurements from detailed plans and the breakdown of these measurements into as many logical units as possible offer the only firm foundation of a dependable estimate. After the job is completed records should be tabulated to be used on future jobs. The author hesitated before offering the following unit costs to the industry, fearing that misuse of them would cause someone a loss. However, if used with judgment, unit costs offer a check on a detailed estimate. The unit costs are based on American Federation of Labor average wages and a record of conditions during World War II, and may prove convenient to contractors in estimating budgets and checking detailed estimates. The labor costs are high compared with average building costs for the same branch of labor, because air conditioning subcontracts seldom attain the volume of a building contract. In fact it is difficult to find a sub-contractor willing to take these small jobs under any consideration, during prosperous building conditions when labor is at a premium.

451. Listing Quantities on an Estimate

All costs for material and labor should be estimated separately, because there is more variation in labor costs than in material costs. The various items that may appear on a summary sheet of an industrial air conditioning contractor are tabulated on the form *Engineering Survey and Report* in the second topic of this chapter.

452. Classification of Work

A list of sub-contractors that may be associated with the installation of an air conditioning system is tabulated below and if time permits, it is always common sense and good business to obtain authentic bids, in preference to the unit prices given.

1. *Carpentry*—Including installation of insulating board, sheathing, scaffolding, templates, shoring
2. *Concrete*—Including foundations, floors, trenches
3. *Electrical*—Including installation of electrical machinery, controls, power and light wiring
4. *Excavation and Backfilling*—Foundations, trenches
5. *Hauling and cartage*—Machinery from freight car to foundation, general hauling, dirt removal, miscellaneous cartage
6. *Iron or Steel erection*—Supports, tanks, general
7. *Lathing*—Metal lath covering insulation, or patched ceilings
8. *Machinists*—Setting of refrigeration machinery and other equipment
9. *Painting*—Machinery, equipment, damaged construction
10. *Plastering*—Insulation, patching, damaged construction
11. *Pipe covering*—Insulation of hot and cold pipes
12. *Plumbing*—Drains from air washer, refrigerators and other equipment—water supply
13. *Roofing*—Vaporproofing of floors and walls, for low relative humidities
14. *Rigging or machinery movers*—Delivery of machinery and equipment on foundation
15. *Sheet metal work*—Installation of air ducts, air washers up to and including 10 gage galvanized steel (or iron), fans, belts, air filters, miscellaneous
16. *Steamfitting*—Heating hot water and refrigeration piping including equipment
17. *Tile*—Partitions
18. *Waterproofing*—Other than roofing
19. *Welders*—Piping, hangers, supports

20. *Miscellaneous*—Watchman, rubbish removing, awnings, shades, etc.

453. Concrete and Excavation for Foundations

Excavation is measured by the cubic yard, and will vary with the kind of soil. With hand shovel and pick one man can excavate 2.5 cu yd in an eight-hour day when the ground is frozen. Allow one hour per cubic yard for removal and loading truck plus truck rental assumed at $30.00 a day. Allow $0.50 per square foot of 2 in. × 8 in. wood forms, sheet piling or bracing; this includes labor at $1.00 per hour for removal of forms. Cost of cement, sand gravel for a 1:3:5 mix and the pouring of the cement, including labor at $1.00 per hour amounts to $40.00 per cubic yard for small foundations up to 5 cu yd. Large foundations, complete including excavation, run as low as $30.00 per cubic yard.

454. Electric Motors; Setting and Final Connections

Roughing in includes labor and material brought within 15 ft of motor foundations. The following table gives an approximate estimate of *final connections* which includes the setting of motor on its foundation, supports for the motor starter and controller, all intermediate wiring, accessories and connection to the feeder. Motor and starter are furnished by others.

TABLE 186

FINAL ELECTRICAL CONNECTIONS

Horsepower (220 Volts)	Labor and Material per Motor (Labor $2.00 per hr)
½ to 1	$75.00
5	90.00
10	95.00
15	105.00
20	110.00

TABLE 186 (*continued*)

Horsepower (220 Volts)	Labor and Material per Motor (Labor $2.00 per Hr)
25	115.00
30	120.00
40	125.00
50	130.00
60	135.00
75	140.00

455. Electric Power Wiring; Main Feeders

To prepare an accurate estimate, the entire job should be laid out, specifying the exact quantity of cable, hangers, receptacles, connection boxes, distributing panels and miscellaneous accessories. For a preliminary estimate, approximate unit costs are given below.

TABLE 187

ROUGHING IN ELECTRIC CABLE

Amperes (50° C Ambient Temperature)	Horsepower (230 Volts)	Cost per Foot, Material and Labor (Labor, $2.00 per Hour)
4½ to 20	½–5	$1.45
38	10	1.60
56	15	1.75
74	20	2.30
92	25	2.50
110	30	2 70
146	40	3.25
180	50	3.65
215	60	3.95
268	75	4.60

Allow $35.00 per outlet for lights, automatic controls, thermostats etc., this includes wiring and labor for installation only.

456. Hauling, Riggers, Machinery Movers

Tanks, boilers and heavy machinery have been hauled five miles from freight car to site at a rate of \$50.00 per ton, allowing \$150.00 per day for a 5 to 7½ ton truck and up to \$200.00 per day for a ten ton truck including driver. Riggers and machinery movers are an additional expense that may bring the total cost up as high as \$100.00 per ton, e.g., hauling down sidewalk lift and up an elevator shaft, or hauling heavy machinery, fans etc. to upper stories or pent-house on roof.

457. Insulation and Rough Carpentry

The unit of measurement is usually 1,000 board feet. One board foot (BM) is equivalent to 144 cubic inches of lumber. An allowance of 8 to 10 cents per board foot is a reasonable cost for the material used in insulation sheathings, studs and ceiling beams, and should include waste. On insulation work 2 carpenters should erect

200 BM of 2 × 6 studs in an 8 hour day
300 BM of 2 × 10 joists in an 8 hour day
500 BM 2 × 6 sheathing (one layer) in an 8 hour day
1000 BM of heavy timbers in an 8 hour day
1000 BM of flooring in an 8 hour day
400 sq ft of deadening quilt per hour

The above applies to jobs completely installed in one week or less. If job runs longer greater production is obtained.

458. Insulation

The following table gives unit costs on structural insulation for all branches of labor. Data are based on close work, crowded conditions, and a work-period of one week or less.

TABLE 188

INSULATION—STRUCTURAL—COLD STORAGE

	Material Dollars per 100 Sq Ft	Man Hours per 100 Sq Ft
Material and labor values based on 100 sq ft surface		
One board foot = 1 BM = 144 cubic inches		
SHEATHING AND SUPPORTS		
50 nails—¾ lb—16d on studding at 12″ centers	.05	
300 nails—3 lb—8d on wood sheathing	.15	
100 nails—2 lb—16d on wood joists	.10	
200 nails—2 lb—8d on flooring	.12	
Self supporting wall or partition 2″ × 4″ studs—12″ ctr—75 BM	7.50	8
2″ × ¼″ T-irons to support insulated ceiling—180 lb	9.00	4
Ceiling joists 2″ × 8″—12″ center—146 BM	14.60	8
Refrigerator door, frame and hardware—including hanging	150.00	16
Wood sheathing—one (1) layer 1″ × 6″—119 BM	11.90	4
Extra layer of wood sheathing (T & G)	11.90	4
Waterproof paper—one (1) layer	0.75	1
Metal lath and fasteners	2.00	20
One coat Portland Cement plaster	0.80	3
One coat hard plaster	4.00	5
Galv. iron or steel sheathing—18 gage	12.00	35

INSULATION—STRUCTURAL—COLD STORAGE (*continued*)

	Material Dollars per 100 Sq Ft	Man Hours per 100 Sq Ft
FLOORS		
Wood flooring 7/8" × 2 1/4" with 2" × 4" sleepers	25.00	12
Concrete floor 4 1/4" thick of 1:3:5—top—1:2	12.00	12
Floor grating and sleepers—one (1) in. spacing including installation	42.00	30
Mastic covering	20.00	50
Lead flashing—6 lb sq ft	50.00	40
Galv. iron or steel flashing	12.00	35
CORKBOARD		
2" corkboard—against existing sides or self-supporting—one (1) course	31.00	12
3" corkboard—against existing sides or self-supporting—one (1) course	47.00	12
Two (2) courses 2" corkboard—against existing sides or self-supporting	62.00	20
Two (2) courses 3" corkboard—against existing sides or self-supporting	94.00	20
Floor—one (1) course 2" corkboard	31.00	4
Floor—two (2) course 2" corkboard	62.00	8
Floor—one (1) course 3" corkboard	47.00	4
Floor—two (2) course 3" corkboard	94.00	8
Overhead—one (1) course 2" corkboard	31.00	22
Overhead—two (2) course 2" corkboard	62.00	31
Overhead—one (1) course 3" corkboard	47.00	22
Overhead—two (2) course 3" corkboard	94.00	31

INSULATION—STRUCTURAL—COLD STORAGE (*continued*)

	Material Dollars per 100 Sq Ft	Man Hours per 100 Sq Ft
RIGID INSULATION		
2" Rigid board—NC Fiberglas—or mineral wool—175 weld nails and accessories	32.00	
3" Rigid board—NC Fiberglas—or mineral wool—175 weld nails and accessories	49.00	
4" Rigid board—NC Fiberglas—or mineral wool—175 weld nails and accessories	64.00	
Add $16.00 extra on material for each ½" additional thickness. Labor price same as cork		
SEMI-RIGID INSULATION		
One (1) course 1½" TW-MC Fiberglas, mineral wool, etc. on sides	4.15	10
One (1) course 2" TW-MC Fiberglas, mineral wool, etc. on sides	8.30	10
One (1) course 3" TW-MC Fiberglas, mineral wool, etc. on sides	12.50	11
One (1) course 4" TW-MC Fiberglas, mineral wool, etc. on sides	16.62	12
Add 9 hr per 100 sq ft for each additional course on sides		
One (1) course overhead—1½"—Fiberglas, mineral wool or equivalent	4.15	22
One (1) course overhead—2"—Fiberglas, mineral wool or equivalent	8.30	22
One (1) course overhead—3"—Fiberglas, mineral wool or equivalent	12.50	22
One (1) course overhead—4"—Fiberglas, mineral wool or equivalent	16.62	24

INSULATION—STRUCTURAL—COLD STORAGE (*continued*)

	Material Dollars per 100 Sq Ft	Man Hours per 100 Sq Ft

Note: When installing around beams or projections, measure additional distance for figuring surface. For additional courses, increase estimate accordingly. Material includes asphalt, cement, mortar, weld nails or other types of fasteners.

Add 9 hr per 100 sq ft for each additional course overhead

INSULATION—LOOSE FILL, WATERPROOFING, PAINTING
(Per One-Hundred (100) Sq Ft)

	Material Dollars per 100 Sq Ft	Man Hours per 100 Sq Ft
Six (6) inch rock wool or equivalent (Increase estimate proportional for greater thickness) Values include labor only for filling. Add extra for sheathing Standard bulk loose fill as rock wool, etc. weighs approximately ten (10) lb per cu ft and is usually packed in bags weighing 35–40 lb each containing approximately 3½ cu ft	14.00	6
WATERPROOFING		
One (1) special plastered coat	3.00	6
Membrane—3 ply—4 moppings	4.00	9
For each additional ply of felt and hot bitumen mopping add	1.00	2
PAINTING		
Wood sheathing—shellac—2 coats (prime coat before installation) ¾ gal	3.00	2

		Material Dollars per 100 Sq Ft	Man Hours per 100 Sq Ft
PAINTING (*continued*)			
Metal sheathing—red lead or zinc chromate—one (1) coat	⅓ gal	1.00	1
Plaster on metal lath sheathing—two (2) coats	½ gal	1.50	2
Color—for one (1) extra coat of color add	½ gal	1.00	1
GYPSUM TILE PARTITIONS			
3″ Partition gypsum or 4″ clay, mortar		4.00	8
½″ Plastering both sides—cement plaster—3 coats		2.50	8

459. General Painting

Class of Work	*Labor Sq Ft Per Hr*
Wire brush clean	37 to 63
First coat red lead—or zinc chromate	90 to 141
Sec. coat red lead—or zinc chromate	110 to 170
First Color (interior)	90 to 150
Sec. Color (interior)	110 to 180
Cork Paint (Spray)	61 to 109
Putty & Varnish	95 to 143
Shellac (Linoleum)	130 to 195
Varnish	150 to 172
Gloss	40 to 93
Furniture	50 to 90
Graining	32 to 56

Average cost material interior paint $2.12 per gal.
Average cost painting steel shapes $7.00 per ton for two coats includes material and labor.
Allow coverage of 300 sq ft per gallon for interior work.

460. Piping

Estimating Pipe and Fittings. To prepare an accurate estimate on the cost of piping, the entire job should be laid out using the quantity of pipe of each size required; connections such as elbows, tees, nipples, reducers, valves, accessories, etc. should all be figured separately, which requires considerable labor.

In order to avoid this detail work, the following unit costs will assist in the preparation of a rapid and fairly accurate estimate. This type of estimate is divided into three sections;

Table (1) Removals and Replacements
Table (2) *Roughing in*
Table (3) *Final Connections*

Roughing in consists of installing piping mains up to a point within about 15 feet of radiator, fixture, tank, etc. See Figure 105, Table 2.

Table 1: Hours Per Linear Foot for "Roughing In"
Minimum of 3 Lengths of Pipe, Joints and Hangers
Material Costs Based on 1945 O.P.A.

Nominal Pipe Size Inches	Std. Blk. Pipe and Joints Material Only	Std. Galv. Pipe and Joints Material Only	Ex. Hy. Blk. Pipe and Joints Material Only	Man Hours Per Foot (Close Work)
½	$0.06	$0.08	$0.11	0.30
¾	0.08	0.10	0.14	0.35
1	0.12	0.14	0.19	0.40
1¼	0.16	0.19	0.25	0.45
1½	0.18	0.22	0.31	0.50
2	0.25	0.35	0.43	0.65
2½	0.39	0.53	0.72	0.80
3	0.57	0.69	0.96	1.00
4	0.77	0.94	1.30	1.40
5	1.10	1.27	1.80	1.90
6	1.35	1.66	2.50	2.50
7	1.80	2.20	3.50	3.60
8	2.10	2.32	4.00	4.25
9	2.65	3.30	4.70	4.95
10	3.15	3.68	5.20	5.50
11	3.60	4.50	5.90	6.75
12	4.00	4.90	6.30	7.35

Screwed joints up to 2 in.—above flanged. Add for weld joints ½ hr per diameter inch for each joint.

Table 2: Man Hours Removal and Replacements

Nominal Pipe Size Inches	Remove and Replace Fittings Per Fitting	Install Fittings Only Per Fitting	Remove and Replace Pipe Per Foot	Remove Pipe and Fittings Per Foot
½	0.90	0.75	0.35	0.25
¾	1.05	0.90	0.40	0.30
1	1.20	1.00	0.45	0.33
1¼	1.35	1.15	0.55	0.35
1½	1.50	1.25	0.60	0.40
2	1.95	1.60	0.75	0.50
2½	2.40	2.00	0.95	0.65
3	3.00	2.50	1.20	0.80
4	4.20	3.50	1.70	1.10
5	5.70	4.80	2.30	1.40
6	7.50	6.30	3.00	1.60
7	9.90	8.30	4.00	2.00
8	10.50	8.80	4.20	2.50
9	14.85	12.50	6.00	3.00
10	16.50	14.00	6.60	3.50
11	20.40	17.00	8.20	4.00
12	22.05	18.50	8.90	4.40

Note: Man hours included for roughing in only and does not include final connections to fixtures. Increase estimate for valves, traps, controls.

FIGURE 105

Pipe and Labor Costs

Final Connections consist of connecting fixture to opening left in main, approximately 15 feet away measured along pipe, and allowing a unit cost for connecting up each fixture e.g. radiator, tank, pump, spray heads, drains etc. as per Table 188A.

461. Final Connections to Fixtures

A fair estimate of the labor required to set and connect up fixtures is herewith given. It is here assumed that mains, risers, stacks, drains, etc. are in place and it is only required to install branches to the following fixtures and these branches not over 15 feet long. Pipe-covering labor is not included.

TABLE 188A

LABOR REQUIREMENTS FOR FINAL CONNECTIONS

	Connection	Men	Days
(1)	4 Wall or floor radiator	2	2
(2)	1-Floor drain 4 in.—(50 ft cast iron pipe)	2	2
(3)	1-2 to 4 in. pump	4	2
(4)	1-1¼ in. pump	2	2
(5)	1-Tank (hot water)	2	3
(6)	1-Air washer water connections	2	1
(7)	1-Preheater (per coil)	2	1
(8)	1-Reheater (per coil)	2	1
(9)	1-Sink	2	1
(10)	Drinking fountain	2	1

462. Heating Fixtures

Cost of cast iron radiation $0.45 to $0.51 per square foot; material only.

Cost of 1¼ in. pipe coils $0.40 per linear foot; extra for galvanizing.

TABLE 189

PIPE COVERING AND INSULATION LABOR AND MATERIAL

85% MAGNESIA—STANDARD THICKNESS—AND CANVAS

Nominal Pipe Size	Material Per Foot	Man Hours Per Foot
½–1½ in.	$0.32	0.17
2 –3½ in.	0.40	0.20
4 –5 in.	0.48	0.24
6 –7 in.	0.58	0.33

1 IN. THICKNESS MAGNESIA AND CANVAS

½–1½ in.	$0.40	0.20
2 –3½ in.	0.48	0.24
4 –5 in.	0.56	0.28
6 –7 in.	0.66	0.32

1 IN. HAIR AND CANVAS

½–1½ in.	$0.27	0.14
2 –3½ in.	0.31	0.16
4 –5 in.	0.38	0.19
6 –7 in.	0.44	0.22

MAGNESIA BLOCK HEXAGONAL FLAT AND WIRE

1 in.	$0.20/sq ft	0.12/sq ft
1½ in.	0.25/sq ft	0.14/sq ft
2 in.	0.30/sq ft	0.15/sq ft

ARMSTRONG CORK, BRINE THICKNESS—STANDARD FINISH

	Straight Run		Fittings	
Nominal Pipe Size	Material Per Foot 50% Off List	Man Hours Per Foot	Material Per Fitting (T)	Man Hours Per Fitting
¼–⅜–½	$0 65	0.45	$0 63	0.43
¾	0.68	0 48	0.78	0.54
1	0.78	0.54	0.95	0.66
1¼	1.00	0.70	1 33	0.9
1½	1.14	0.79	1 65	1.15
2	1.18	0.82	1 93	1.34
2½	1.35	0.94	2.40	1.67
3	1 65	1.15	3.55	2.46
3½	1.86	1.29	4.03	2.80
4	1 98	1.37	4.90	3.40
5	2.40	1.66	6.08	4.20
6	2.65	1.85	8.10	5.60
8	3.40	2.40	34.22	7.20

Painting Steam Pipe Covering Two Coats Labor and Material Per Foot

¼–¾	$0.10
1	0.10
1¼	0.13
1½	0.15
2	0.18
2½	0.20
3	0.23
4	0.25
5	0.28
6	0.30
7	0.35

INSULATION

Specifications	Material	Man Hours
Ventilation ducts 1 in. semi-rigid	$0.30/sq ft	0.20/sq ft
3 in. semi-rigid—no sheathing	0.15/sq ft	0.15/sq ft
Metal lath and plaster sheathing	0.10/sq ft	0.10/sq ft
Sheathing 2 layers D & M & Paper	0.30/sq ft	0.10/sq ft

Table 189A

CLOSE WORK—MAN HOURS PER FOOT GALVANIZED IRON DUCTS, FITTINGS, FASTENERS, STIFFENERS

Width Inches / Depth Inches	4	5	6	7	10	14	18	20	22	26	30	34	38	42	46	50	60	70	84	96	Depth Inches
4	0.54	0.60	0.67	0.74	0.93	1.18	1 44	1.58	1.70	1.96	2.23	2.56	2.76	3.02	3.28	3.54	4.20				4
8	0.80	0.87	0.93	1.00	1.18	1.44	1 70	1.84	1.96	2.23	2.50	2.75	3.02	3.28	3.54	3.80	4.45				8
12	1.07	1.13	1 20	1.27	1.44	1.70	1.96	2.10	2.23	2.49	2.76	3.01	3.28	3.54	3.86	4.06	4.70				12
16	1.33	1.4	1.46	1.53	1.70	1.96	2.23	2.36	2.49	2.75	3.02	3.28	3.55	3.80	4 09	4.33	5.00				16
20	1.60	1.67	1.73	1.80	1.96	2.22	2 50	2 62	2.74	3.02	3.28	3 54	3.80	4.06	4 34	4 60	5.25				20
24	1.87	1.93	2 00	2.06	2.23	2.49	2.75	2.88	3.00	3.28	3.54	3.80	4.06	4.34	4.60	4.85	5.50				24
28	2.13	2.20	2.26	2.32	2.49	2.75	3 01	3.14	3.29	3.54	3.80	4.06	4.35	4.60	4.85	5.12	5.75	6.44			28
32	2.4	2.48	2.54	2.60	2.75	3.01	3.28	3.40	3.54	3 80	4.06	4.34	4.60	4.85	5.10	5.37	6.05	6.82			32
36	2.66	2.70	2.80	2.86	3.01	3.28	3 54	3.68	3.80	4.06	4 34	4.60	4.85	5.12	5.36	5.65	6.30	7.10			36
40	2.92	3.00	3.08	3.12	3.28	3.53	3.80	3.94	4.05	4.33	4.60	4.85	5 13	5.36	5.65	5.90	6.55	7.35			40
44	3.20	3.28	3.33	3.39	3.54	3.80	4.05	4.20	4.33	4.60	4.85	5.13	5.36	5.65	5 90	6.15	6.95	7.60			44
48	3.48	3.52	3.60	3.66	3 80	4.05	4.33	4.45	4.60	4.85	5 10	5.36	5.65	5.90	6.15	6.42	7.20	7.90			48
52	3.75	3.80	3.86	3.92	4.05	4.34	4.60	4.70	4.85	5.10	5.36	5.64	5.90	6.25	6.44	6.80	7.49	8.10	9.10	9.90	52
56	4.00	4.05	4.10	4.19	4.34	4.60	4.85	5.00	5.10	5.36	5.65	5.90	6 15	6.43	6.80	7.10	7.75	8.40	9.35	10.20	56
64	4.55	4.60	4.65	4.73	4.85	5.10	5.37	5.50	5.63	5.90	6.16	6.44	6.80	7.05	7.30	7.60	8.30	8.90	9.80	10.70	64
66	4.67	4.70	4.80	4.85	5.05	5.34	5.60	5.73	5.85	6.04	6.30	6.55	6.95	7.20	7.49	7.75	8.40	9.16	10.00	10.80	66
72																			10.50	11.20	72
76																			10.70	11.50	76
78																			10.80	11.60	78
88																			11.50	12.20	88

Measurements taken along center line of mains, fittings and branches. Gage of metal 4 in. to 12 in., 22 gage; 12½ in. to 23½ in., 20 gage; 24 in. up, 18 gage. Table also represents approximate dollar value material—1945.

Railing, Iron Pipe

Galvanized pipe—three rows (1 in. top row—¾ in. balance) sockets—stanchions—railing fittings.

One new stanchion and socket	$7.00
27 ft pipe railing (9 ft section) and fittings	10.75
Labor installation ($2.00 per hour) a section	20.00

463. Ventilation Ducts

The unit prices in Table 189A are based on ceiling prices of material during the war period and the billing price of sheet metal contractors for union labor at $2.75 per hour. Materials included in these units are galvanized sheet iron, drills, tools, screws, bolts, nuts, rivets, screens, expanded metal, structural steel stiffeners, diffusers, dampers, grilles, turnvanes, foundations, joints and all the miscellaneous material starting from fan inlet and outlet; included, also, is labor for fabricating, installing and testing the unit. Fans, motors, pulleys and belts are furnished by others but labor for setting, installing and testing these are included in unit prices. Items not included are, e.g., heating coils, piping, accessories and electric wiring. In case of labor rates lower than $2.75 per hour allow 75% of unit cost for labor and 25% for material.

Gage of Metal

	Gage	Lb/Sq Ft
Ducts 24 in. and larger	18	2.156
Ducts 12½ in. to 23½ in.	20	1.656
Ducts up to 12 in.	22	1.406

464. Sheet Metal Casings; Sheathing Flashing; Housings

Cost varies with price of metal, location of work, and complexity or simplicity of shapes. Ventilation ducts in Table 189A are an example of complicated shapes and cost about $1.00 per lb installed. Flat surfaces as above described may be estimated at 50¢ per lb installed.

Weight is based on actual area of galvanized sheets installed. The following table will prove convenient in reducing to cost per square foot.

TABLE 190

COST PER SQ FT INSTALLED
Material and Labor ($2.00 per hr)

16 gage	$1.35 per sq ft
18 gage	1.00 per sq ft
20 gage	0.80 per sq ft
22 gage	0.70 per sq ft
10 gage	2.90 per sq ft
12 gage	2.26 per sq ft

Base price of galvanized metal assumed at 5¢ per pound includes miscellaneous material, rivets or welding, gaskets, hardware, etc.

TABLE 191

APPROXIMATE PRICES OF VENTILATING FANS WITH V-BELT DRIVES AND 220 VOLT, D-C MOTORS

Cubic Feet Per Minute 2 in. Static	Area of Discharge, Square Feet	Size of Motor	Net Price of Unit
1,900	0.79	1½	$248.00
2,980	1.24	2	345.00
4,630	1.78	3	381.00
7,260	2.42	5	468.00
10,130	3.17	7½	572.00
12,830	4.01	10	658.00
17,810	4.95	15	794.00
24,000	6.00	20	910.00
28,500	7.13	25	1020.00

TABLE 191 (*continued*)

Cubic Feet Per Minute 2 in. Static	Area of Discharge, Square Feet	Size of Motor	Net Price of Unit
34,760	8.69	30	1160.00
42,520	10.63	35	1338.00
49,250	12.96	40	1610.00
57,100	15.86	45	1920.00
65,890	19.38	50	2370.00
80,720	23.74	60	2950.00
98,600	29.00	75	3800.00
134,940	35.51	100	4940.00
147,220	43.30	125	7200.00
190,981	53.05	150	8800.00
232,920	64.70	175	11200.00

TABLE 192

APPROXIMATE COSTS
Mechanical Draft Towers
Steel

Gallons Per Minute	Horsepower	Space	Price, Erected
200	2	9 x 13 ft	$3,600
300	3	11 x 16 ft	4,700
450	4	13 x 19 ft	6,050
600	6	16 x 21 ft	8,200
900	8	19 x 25 ft	10,500
1050	10	22 x 25 ft	12,200
1400	12	25 x 27 ft	15,100
1600	14	27 x 30 ft	18,000

465. Budget Estimates for Room Coolers

Costs are different in every locality, but as a guide, the estimator may find the following unit costs convenient to use until more specific information is available. The costs given are from records of jobs actually installed, and include final electric wiring, and final sheet-metal work.

Table 1: Fillet Arc Welding Decimal Hours Per Foot

Plate Thickness Inches	Flat	Vertical	Horizontal	Over Head
1/8	0.11	0.15	0.16	0.20
3/16	0.15	0.20	0.22	0.24
1/4	0.24	0.28	0.26	0.30
5/16	0.28	0.39	0.45	0.53
3/8	0.40	0.53	0.60	0.64
7/16	0.44	0.55	0.62	0.66
1/2	0.53	0.64	0.70	0.77
9/16	0.63	0.75	0.83	0.90
5/8	0.75	0.88	0.96	1.06
3/4	0.84	1.06	1.18	1.28
7/8	0.97	1.18	1.28	1.38
1	1.06	1.28	1.39	1.48

Add 0.22 hr per ft for welding on galv. metal.

Table 2: Butt Arc Welding Decimal Hours Per Foot

Plate Thickness Inches	Flat	Vertical	Horzontal	Over Head
1/4	0.34	0.42	0.50	0.53
5/16	0.43	0.53	0.60	0.64
3/8–7/16	0.64	0.85	1.07	1.28
1/2	0.74	0.97	1.18	1.40
5/8	0.96	1.40	1.78	2.00
3/4	1.28	1.60	1.94	2.12
7/8	1.40	1.82	2.04	2.38
1	1.50	1.94	2.12	2.46
1 1/8	1.83	2.38	2.54	3.20
1 1/4	2.04	2.56	2.78	3.30

Note: Above based on 25% estimated arc time. Actual production records have shown 50% improvement.

FIGURE 106

Welding Production

Approximate cost data, small room coolers.
One ton—1 horsepower—air cooled condenser installed $500
One ton—1 horsepower—water cooled condenser installed $600

TABLE 193

COMMERCIAL SELF-CONTAINED ROOM COOLERS

Tons	Horse Power	Number Persons	Floor Space Cooled Square Feet	Air Cubic Feet per Minute	Approximate Cost Installed
1½	1.5	6	150	650	$ 700.
2	2	10	240	650	1,000.
3	3	15	360	1,275	1,500.
5	5	24	600	2,000	2,000.
7.5	7.5	36	900	2,800	2,500.
10	10	50	1,200	2,800	3,000.

Electric wiring is an extra cost. The capacities above are based on the full occupancy of a metropolitan restaurant and lunch counter with a high moisture load, hooded.

466. Central Systems

Theaters

Capacities, in terms of the refrigeration tonnage found acceptable, vary from one ton per 15 people in southern sections to one ton per 19 people in northern sections. Operating costs amount to approximately 2½ cents per hour or approximately 5 cents per two-hour performance per seat at full capacity. Costs of equipment and installation in theaters vary from $400 to $475 per ton of refrigeration or about $25.00 to $30.00 per seat for 1500 to 3000 seat theater capacity.

Department Stores

There are three types of occupancy in department store classifications.

Basement: 1 person per 20 sq ft floor area
First floor: 1 person per 35 sq ft floor area
Upper floors: 1 person per 50 sq ft floor area

This space includes sales area and service area. Capacities vary from 1 ton per 8 persons to 1 ton per 10 persons, since the lighting load is considerable. The cost of equipment installed varies from $2.00 to $2.50 per square foot of floor space cooled (from 50,000 sq ft up). The yearly operating costs per square foot vary from 4½ to 6 cents.

Multi-story office buildings

An allowance of one ton of refrigeration per 350 to 500 sq ft of floor area has proved acceptable. Cost of equipment installed for summer cooling only varies from $2.00 to $2.50 per square foot of floor area. Operating costs per year per square foot of floor area vary from 4½ to 6 cents.

467. Yearly Operating Hours

Cooling systems are usually made ready for operation May first and closed down for the season September first. During the warm season the system operates about 1200 hours. The operating costs above are based on this period of operation, at latitude 40°; the maintenance costs in Table 194 are distributed over the year.

Column (1) gives tons of refrigeration capacity when conditioning air at a suction temperature of 40° F for the saturated refrigerant. It also gives connected horsepower to drive compressor.

Column (2) gives budget allowance recommended for recharging with refrigerant, lubricating oil, shaft seal packing, belts, servicing, general repairs and replacement of worn-out, broken or corroded parts. Add 10% to above charges for repairs of heating coils and controls, if winter air conditioning is considered.

Air filters should be replaced at least twice a year and vary

TABLE 194

REFRIGERATION MAINTENANCE AND REPAIRS FOR COMPRESSION AIR-CONDITIONING SYSTEMS

Tons of Refrigeration or Horsepower (1)	Yearly Maintenance Budget (2)
2	$150.
5	250.
10	330.
15	400.
20	450.
25	500.
30	575.
35	625.
40	675.
45	700.
50	775.
60	860.
75	1,000.
100	1,250.

in cost from $1.00 to $1.50 per square foot or $1.25 to $1.75 per refrigeration ton. Allow 300 cu ft of air per minute per ton for filter capacity.

468. Depreciation

The machine age has brought about a condition where depreciation is a factor of cost which must receive recognition. Practices for determining and accounting for depreciation are far from being standardized. Estimates of life expectancy or service life are primarily a matter of engineering and not a function of the cost accountant. Accountants have not reached conclusions recommending any particular method. This situation leads to numerous problems in estimating the life and abilities of equipment, and in methods of accounting for decrease in value. Engineers and cost accountants must cooperate in submitting reports on depreciation charges. From all the various definitions of depreciation it is evident that depreciation refers to the expiration in use of the usefulness, cost, or value of a unit of physical property. Therefore it does not include ordinary repairs, and maintenance, but the rate of depreciation is decidedly influenced by repair and maintenance policies. The act of 1909 levied a 1% tax on corporate incomes. Congress (in subsequent acts) and the Treasury Department have consistently held to the view that in determining income, the taxpayer is entitled to deduct a reasonable allowance for depreciation. Obsolescence was not explicitly recognized as a factor until the act of 1918.

469. Computing Annual Renewals of Air Conditioning Units

When a group of units is placed in service simultaneously, a few go out of service almost immediately, some remain in service a few years longer, others stay in service up to and around the expected average life, while approximately half of the original units give service beyond the expected life of the average unit; in fact some may stay in service beyond 200% of average life. When the number that go out each year are plotted against age reached at retirement, the well known distribution or frequency curve results.

470. Depreciation Rates

There are two principal sources for information on depreciation rates

1. Industrial and accounting authorities
2. Reports of the Commissioner of Internal Revenue.

From these sources the following data have been abstracted; these particularly refer to air conditioning, heating and refrigerating equipment and materials.

TABLE 195

DEPRECIATION RATE IN PER CENT

Item	Bureau of Internal Revenue	Other Authorities
Blowers and fans	6⅔	10
Boilers and furnaces	5	5
Hot blast coils	5	5
Radiators	4	5
Heating & ventilating systems	5	5
Copper piping		2
Cast iron piping		1⅓
Sewer & drainage		6
Iron (cold water black)	3½	
Iron (hot water or steam)	4	
Wrought iron	3	
Brine pipe (black steel)		15 (author)
Brine pipe (galvanized steel)		10 "
Brine pump		15 "
Brine tank		10 "
Sheet iron		3–7
Humidifiers	10	5
Ice boxes	6⅔	
Refrigeration machinery	6⅔	
Air Conditioners	6⅔	
Spray pond	6⅔	
Ovens	6⅔	
Dryers	10	
Electrical apparatus		7½–10
Steam Engines	5	
Feed heaters	4	
Compressors		4–5
Condensers (surface)	5	
Condensers (atmospheric)	4	
Electric controllers	6⅔	

APPENDIX *

TERMINOLOGY

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

Absolute Zero: The zero from which absolute temperature is reckoned. Approximately —273.2° C or —459.8° F.

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

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

Acceleration: The time rate of change of velocity, *i.e.*, the derivative of velocity with respect to time. In the cgs system the unit of acceleration is the centimeter per (second) (second) in the fps system the unit is the foot per (second) (second) $a = \frac{dv}{dt}$.

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

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

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

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

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

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* Reprinted from Heating, Ventilating, Air Conditioning Guide, 1946.

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

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

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

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

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

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

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

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

In rectangular ducts the ratio of the width to the depth.

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

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

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

Blow (throw): In air distribution, the distance an air stream travels from an outlet to a position at which air motion along the axis reduces to a velocity of 50 fpm.

For unit heaters, the distance an air stream travels from a heater without a perceptible rise due to temperature difference and loss of velocity.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929.)

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

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

For most accurate work the *International Table* (I.T.) Btu is usually used. This is defined by the relation: 1 (I.T.) Btu per (pound) (Fahrenheit degree) = 252 (I.T.) calorie per (gram) (Centigrade degree). This value corresponds to the amount of heat required to raise the temperature of 1 lb of water 1 Fahrenheit degree at 58° F and also at 149° F. The *mean Btu* corresponds to 1.0008 (I.T.) Btu.

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

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

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

The kilocalorie = 1,000 cal

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

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

Comfort Air Conditioning: The process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be maintained within required limits. (See *Air Conditioning.*)

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

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

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

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

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

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

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

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

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

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

Decibel: A unit used to express the relation between two amounts of power. By definition the difference in decibels between two powers P_1 and P_2, P_2 being the larger, is

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

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

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

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

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

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

Dew Point: See *Temperature, Dew Point.*

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

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

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

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

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

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

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

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

Dust: An air suspension (aerosol) of solid particles of any material.

Enthalpy: A term applied to the Gibbs function, $h = u + pv$ and superseding ***total heat, heat content,*** and ***heat of formation.*** The increase of enthalpy of a system is equal to the heat absorbed by the system during a process occurring at constant pressure when the only work done is that of compression or expansion.

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

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

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

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

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

where T is the absolute temperature of the system.

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

The term *Specific Entropy* is sometmes applied to entropy per unit weight, the English unit being Btu per (Fahrenheit degree) (pound).

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

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

Fog: Suspended liquid droplets generated by condensaton from the gaseous to the liquid state or by breaking up a liquid into a dispersed state, such as by splashing, foaming, and atomizing.

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

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

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

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

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

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

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

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

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

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

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

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

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

Heat, Total: See Enthalpy.

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

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

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

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

Humidity: Water vapor within a given space.

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

Humidity, Relative: The ratio of the actual partial pressure of the water vapor in a space to the saturation pressure of pure water at the same temperature.

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

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

Humidity, Specific: See Humidity Ratio.

Hygrostat: Same as Humidistat.

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

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

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

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

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

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

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

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

Mechanical Equivalent of Heat: The quantity of mechanical energy equal to one unit of heat. $J = 778.3$ ft-lb per Btu $= 4.187 \times 10^7$ ergs per gram-calorie.

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

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

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

Mol: A weight of a substance numerically equal to its molecular weight. If the weight is in pounds the unit is a *Pound Mol,* in grams the unit is a *Gram Mol.* For perfect gases the volume of 1 mol is constant for all gases at the same temperature and pressure. For real gases this is approximately true at moderate pressures. At 32° F and zero pressure the value of the product, pressure times specific volume, is 359.045 ± 0.006 atmospheric cubic feet (atm ft^3), *for 1 mol of any gas.* For dry air at 32° F and standard atmospheric pressure, the specific volume is 358.83 cu ft per mol (ft^3 per mol).

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

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

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

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

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

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

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

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

Pressure: Force per unit area. Common units are pounds per square inch, grams per square centimeter, inches of water, millimeters of mercury.

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

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

Pressure, Dynamic: Same as Total Pressure.

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

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

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

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

Pressure, Velocity: In a moving fluid, the difference, due to velocity, between total pressure and static pressure.

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

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

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

Pyrometer: An instrument for measuring high temperatures.

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

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

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

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

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

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

Radiator, Direct: Same as Radiator.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

Sorption: Adsorption or absorption.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

ABBREVIATIONS AND SYMBOLS

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

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

Abbreviations

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

Absolute	abs
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr
Atmosphere	atm
Average	avg
Avoirdupois	avdp
Barometer	bar.
Boiling point	bp

Brake horsepower.............................bhp
Brake horsepower-hour ....bhp-hr
British thermal unit ....Btu
British thermal units per hour......................Btuh
Calorie ....cal

Centigram ....cg
Centimeter ....cm
Centimeter-gram-second (system).......................cgs
Cubic ....cu
Cubic centimeter...........................cu cm, or cc

Cubic foot ....cu ft
Cubic feet per minute ....cfm
Cubic feet per second ....cfs
Decibel ....db
Degree ....deg or °

Degree, Centigrade C
Degree, Fahrenheit ....F
Degree, Kelvin ....K
Degree, Réaumur ....R
Diameter ....diam

Direct-current (as adjective) ....d-c
Electromotive force ....emf
Feet per minute ....fpm
Feet per second ....fps
Foot ....ft

Foot-pound ....ft-lb
Foot-pound-second (system) ....fps
Freezing point ....fp
Gallon ....gal
Gallons per minute ....gpm

Gallons per second ....gps
Gram ....g
Gram-calorie ....g-cal
Horsepower ....hp
Horsepower-hour ....hp-hr

Hour .... hr
Inch .... in.
Inch-pound .... in.-lb
Indicated horsepower .... ihp
Indicated horsepower-hour .... ihp-hr

Kilogram .... kg
Kilowatt .... kw
Kilowatthour .... kwhr
Mass .... mass
Melting pont .... mp

Meter .... m
Micron .... μ (mu)
Miles per hour .... mph
Millimeter .... mm
Minute .... min

Molecular weight .... mol. wt
Mol .... mol
Ounce .... oz
Pound .... lb
Pounds per square inch .... psi

Pounds per square inch, gage .... psig
Pounds per square inch, absolute .... psia
Revolutions per minute .... rpm
Revolutions per second .... rps
Second .... sec

Specific gravity .... sp gr
Specific heat .... sp ht
Square foot .... sq ft
Square inch .... sq in.
Watt .... w
Watthour .... whr

Symbols

A letter symbol is a single character, with subscript or superscript if required, used to designate a physical magnitude in mathematical equations and expressions. Two or more symbols together always represent a product. The following have been compiled from a selected list of approved standards.

Acceleration due to gravity . . . . g
Acceleration, linear . . . . a
Area . . . . A
Change in specific volume during vaporization . . . . v_{fg}
Density, Weight per unit volume, Specific weight. . . . *d or* ρ (rho)

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

Distance, linear . . . . s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, vapor in contact with liquid. . *Subscript* g
Efficiency . . . . η
Elevation above some datum . . . . z, Z
Emissivity . . . . ε
Energy in general; work, total; work, molal. . . . . E
Entropy. (The capital should be used for any weight, and the small letter for unit weight) . . . . S or s
Force, total load . . . . F
Gas Constant, in equation $pV = nRT$ . . . . R
Head . . . . H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight) . . H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid . . . . h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor . . . . h_g
Heat of vaporization at constant pressure . . . . L or h_{fg}
Hydraulic radius . . . . R_H
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight) . . U or u
Length of path of heat flow, thickness L
Load, total . . . . W
Mechanical efficiency . . . . e_m
Mechanical equivalent of heat . . . . J
Power, Horsepower, Work per unit time . . . . P

Pressure, Absolute pressure, Gage pressure, Force per unit area. . p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred .... Q
Quality of steam, Pounds of dry steam per pound of mixture. . x
Reynolds number .... N_{Re}
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor .... *Subscript* f
Specific heat c
Specific heat at constant pressure .... c_p
Specific heat at constant volume .... c_v
Specific volume, Volume per unit weight, Volume per unit mass. . v
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion) .... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature) .... T or Θ (*capital theta*)
Thermal conductance[2]: heat transferred per (unit time) (degree) C

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

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

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

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

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

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

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

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

(In general f is not equal to k/L, where L is the actual thickness of the fluid film.)

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

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

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

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

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

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

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

Conversion Equations[3]

Heat, Power and Work

1 ton refrigeration $= \begin{cases} \text{12,000 Btu per hour} \\ \text{200 Btu per minute} \end{cases}$

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

Latent heat of ice	= 143.4 Btu per pound
1 Btu	= 778.3 ft-lb 0.2930 Int. whr 252.0 I.T. calorie
1 Int. watthour	= 2656 ft-lb 3.413 Btu 3600 Int. joules 860 I.T. calories
1 Int. kilowatthour	= 3,413 Btu 3.517 lb water evaporated from and at 212° F
1 Int. kilowatt (1000 watts)	= 1.341 hp 56.88 Btu per minute 44,267 ft-lb per minute
1000 I.T. calories 1 I.T. Kilocalorie	= 3.968 Btu 3088 ft-lb 1.1628 Int. whr
1 horsepower	= 0.7455 Int. kw 42.40 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,475 Btu per hour 9.809 Int. kw

Weight and Volume

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

Pressure

1 lb per square inch	= 144 lb per square foot 2.0360 in. mercury at 32° F 2.0422 in. mercury at 62° F 2.309 ft water at 62° F 27.71 in. water at 62° F
1 oz per square inch	= 0.1276 in. mercury at 62° F 1.732 in. water at 62° F

1 atmosphere	=	14.696 lb per square inch 2116 lb per square foot 33.94 ft water at 62° F 30.01 in. mercury at 62° F 29.921 in. mercury at 32° F
1 in. water at 62° F (in vacuo)	=	0.03609 lb per square inch 0.5774 oz per square inch 5.197 lb per square foot
1 ft water at 62° F (in vacuo)	=	0.4330 lb per square inch 62.37 lb per square foot
1 in. mercury at 62° F (in vacuo)	=	0.4897 lb per square inch 7.835 oz per square inch 1.131 ft water at 62° F 13.57 in. water at 62° F
1 in. mercury at 32° F (in vacuo)	=	0.49115 lb per square inch

Metric Units

1 cm	=	0.3937 in. = 0.0328 ft
1 in.	=	2.540 cm
1 m	=	3.281 ft
1 ft	=	0.3048 m
1 sq cm	=	0.1550 sq in.
1 sq in.	=	6.452 sq cm
1 sq m	=	10.76 sq ft
1 sq ft	=	0.09290 sq m
1 cu cm	=	0.06102 cu in.
1 cu in.	=	16.39 cu cm
1 cu m	=	35.31 cu ft
1 cu ft	=	0.02832 cu m
1 liter	=	1000 cu cm = 0.2642 gal
1 kg	=	2.205 lb (avdp)
1 lb	=	0.4536 kg
1 metric ton	=	2205 lb (avdp)
1 gram	=	0.002205 lb (avdp)
1 kilometer per hour	=	0.6214 mph
1 gram per square centimeter	=	0.02905 in. mercury at 62° F 0.3942 in. water at 62° F
1 kg per sq cm (metric atmosphere)	=	14.22 lb per square inch
1 gram per cubic centimeter	=	0.03613 lb per cubic inch 62.43 lb per cubic foot
1 dyne	=	0.00007233 poundals
1 absolute joule	=	10,000,000 ergs 0.7376 ft-lb
1 Int. joule	=	0.7378 ft-lb
1 metric horsepower	=	75 kg-m per second 0.986 hp (U. S.)

1 I.T. kilocalorie per kilogram	= 1.8 Btu per pound
1 I.T. calorie per square centimeter	= 3.687 Btu per square foot
1 I.T. calorie per (second) (square centimeter) for a temperature gradient of 1° C per centimeter	= 2903 Btu per (hour) (square foot) for a temperature gradient of 1° F per inch of thickness.

The Greek Alphabet

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

GRAPHICAL SYMBOLS FOR DRAWINGS[6]

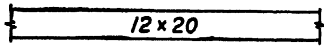
Graphical Symbols for Drawings — ***Piping***

Item	Symbol
Heating	
1. High Pressure Steam	
2. Medium Pressure Steam	
3. Low Pressure Steam	
4. High Pressure Return	
5. Medium Pressure Return	
6. Low Pressure Return	
7. Boiler Blow Off	
8. Condensate or Vacuum Pump Discharge	
9. Feedwater Pump Discharge	
10. Make Up Water	
11. Air Relief Line	
12. Fuel Oil Flow	FOF
13. Fuel Oil Return	FOR
14. Fuel Oil Tank Vent	FOV
15. Compressed Air	A
16. Hot Water Heating Supply	
17. Hot Water Heating Return	
Air Conditioning	
18. Refrigerant Discharge	RD
19. Refrigerant Suction	RS
20. Condenser Water Flow	C
21. Condenser Water Return	CR
22. Circulating Chilled or Hot Water Flow	CH
23. Circulating Chilled or Hot Water Return	CHR
24. Make Up Water	
25. Humidification Line	H
26. Drain	D
27. Brine Supply	B
28. Brine Return	BR
Plumbing	
29. Soil, Waste or Leader (Above Grade)	
30. Soil, Waste or Leader (Below Grade)	
31. Vent	
32. Cold Water	
33. Hot Water	
34. Hot Water Return	
35. Fire Line	F F
36. Gas	G G
37. Acid Waste	ACID
38. Drinking Water Flow	
39. Drinking Water Return	
40. Vacuum Cleaning	V V
41. Compressed Air	A
Sprinklers	
42. Main Supplies	S
43. Branch and Head	
44. Drain	S S

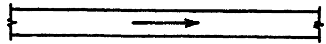
[6] Graphical Symbols for Use on Drawings in Mechanical Engineering, Z32.2-1941 (*American Standards Association*).

GRAPHICAL SYMBOLS FOR DRAWINGS ***Ductwork***

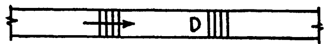
45. Duct (1st Figure, Width; 2nd, Depth)



46. Direction of Flow



47. Inclined Drop in Respect to Air Flow



48. Inclined Rise in Respect to Air Flow

49. Supply Duct Section

50. Exhaust Duct Section

51. Recirculation Duct Section

52. Fresh Air Duct Section

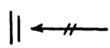
53. Other Duct Sections

54. Register R

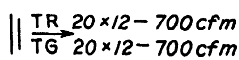
55. Grille G

56. Supply Outlet

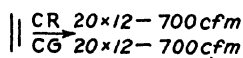
57. Exhaust Inlet



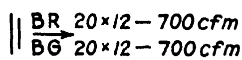
58. Top Register or Grille



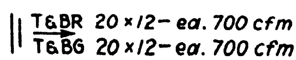
59. Center Register or Grille



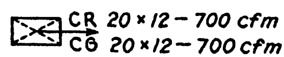
60. Bottom Register or Grille



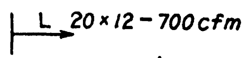
61. Top and Bottom Register or Grille



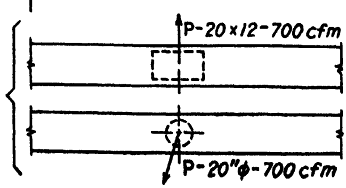
62. Ceiling Register or Grille



63. Louver Opening

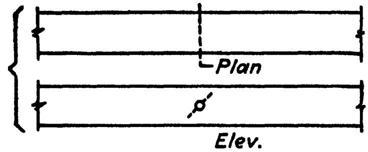


64. Adjustable Plaque

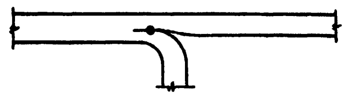


Graphical Symbols for Drawings **Ductwork**

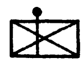
65. Volume Damper



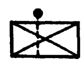
66. Deflecting Damper



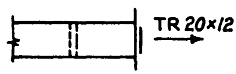
67. Deflecting Damper, Up



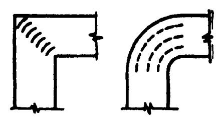
68. Deflecting Damper, Down



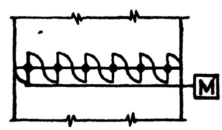
69. Adjustable Blank Off



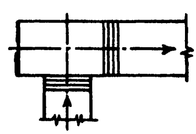
70. Turning Vanes



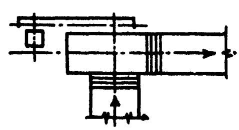
71. Automatic Dampers



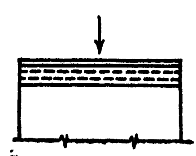
72. Canvas Connections



73. Fan and Motor With Guard



74. Intake Louvers and Screen

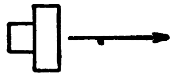


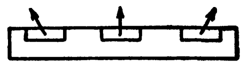
Graphical Symbols for Drawings ***Heating and Ventilating***

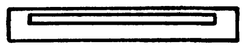
75. Heat Transfer Surface, Plan 

76. Wall Radiator, Plan 

77. Wall Radiator on Ceiling, Plan 

78. Unit Heater (Propeller), Plan 

79. Unit Heater (Centrifugal Fan), Plan 

80. Unit Ventilator, Plan 

Traps

81. Thermostatic 

82. Blast Thermostatic 

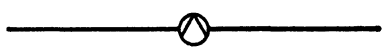
83. Float and Thermostatic 

84. Float 

85. Boiler Return 

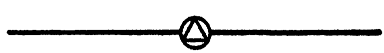
Valves

86. Reducing Pressure 

87. Air Line 

88. Lock and Shield 

89. Diaphragm 

90. Air Eliminator 

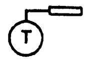
91. Strainer 

92. Thermometer 

93. Thermostat 

Graphical Symbols for Drawings *Refrigerating*

94. Thermostat (Self Contained)

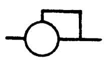

95. Thermostat (Remote Bulb)


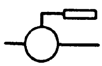
96. Pressurestat

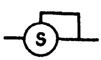

97. Hand Expansion Valve

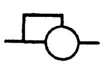

98. Automatic Expansion Valve


99. Thermostatic Expansion Valve


100. Evaporator Press. Regulating Valve, Throttling Type


101. Evaporator Press. Regulating Valve, Thermostatic Throttling Type


102. Evaporator Press. Regulating Valve, Snap-Action Valve


103. Compressor Suction Pressure Limiting Valve, Throttling Type


104. Hand Shut Off Valve


105. Thermal Bulb

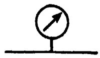

106. Scale Trap

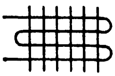

107. Dryer

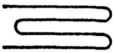

108. Strainer

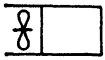

109. High Side Float


110. Low Side Float


111. Gage


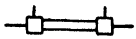
112. Finned Type Cooling Unit, Natural Convection


113. Pipe Coil


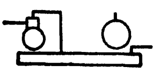
114. Forced Convection Cooling Unit


115. Immersion Cooling Unit


116. Ice Making Unit

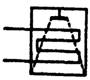

117. Heat Interchanger


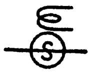
118. Condensing Unit, Air Cooled


119. Condensing Unit, Water Cooled


120. Compressor


121. Cooling Tower


122. Evaporative Condenser


123. Solenoid Valve


124. Pressurestat With High Pressure Cut-Out


IDENTIFICATION OF PIPING SYSTEMS BY COLOR

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

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

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

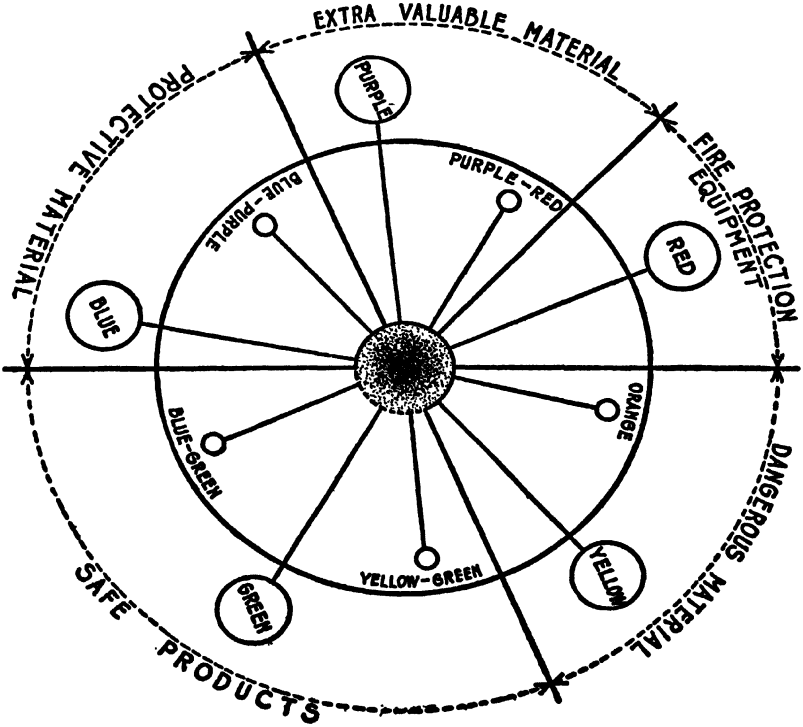


FIG. 1. MAIN CLASSIFICATION BY COLOR[a]

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

INDEX

A

F

G

Y

Z